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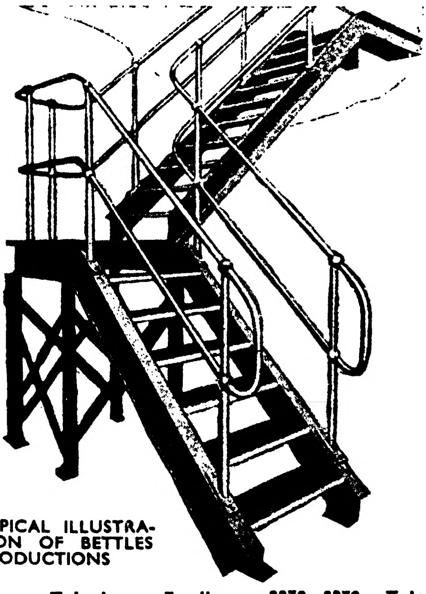

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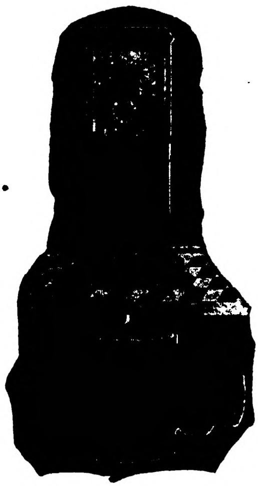

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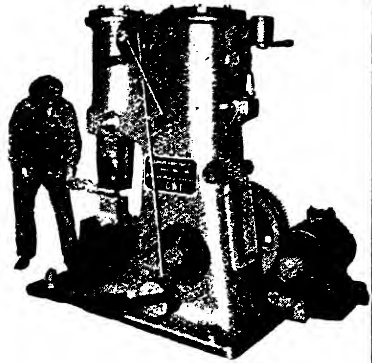

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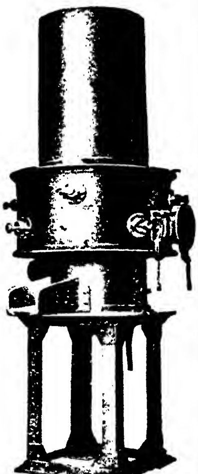

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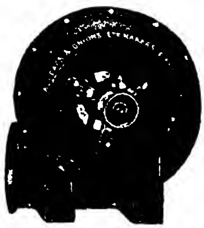

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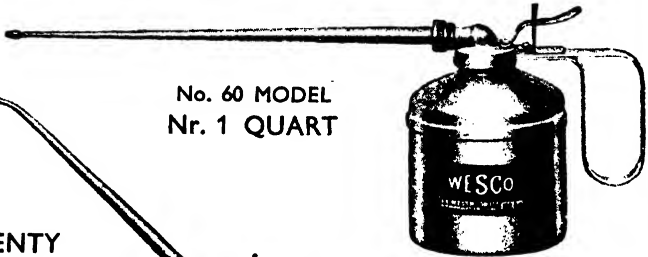

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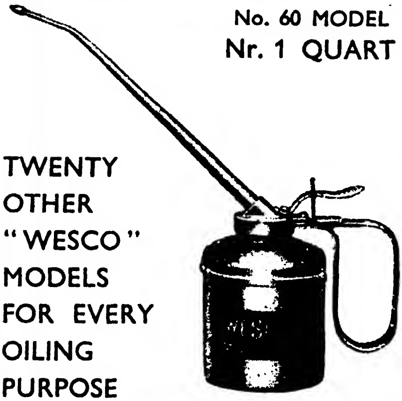

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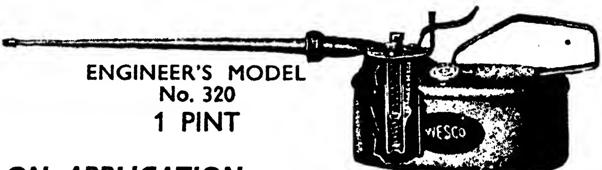

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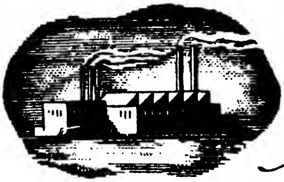

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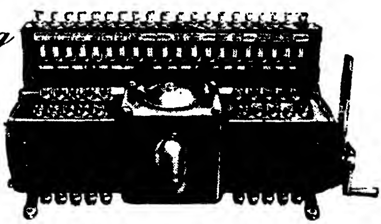

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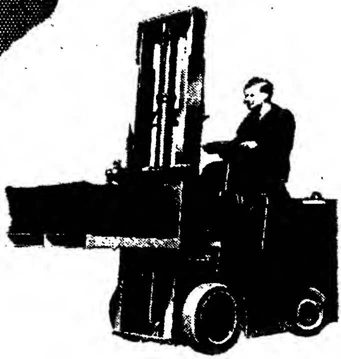

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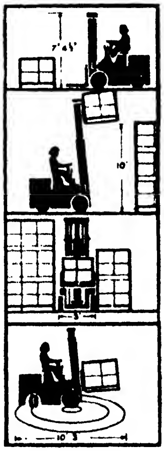

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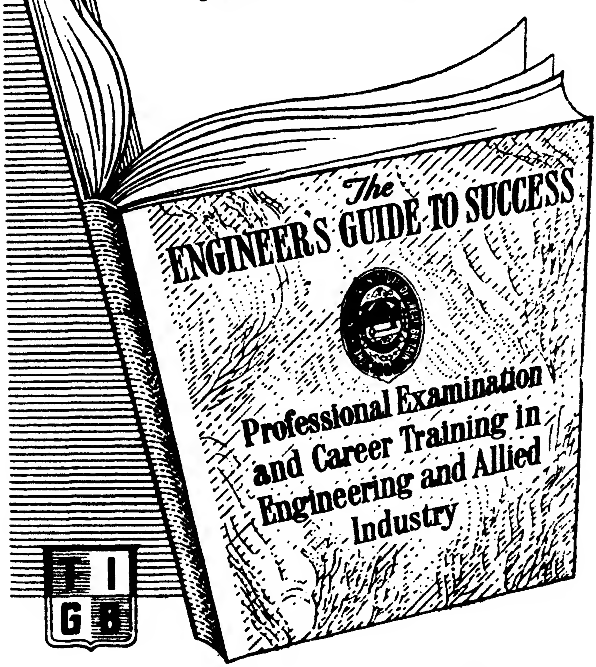

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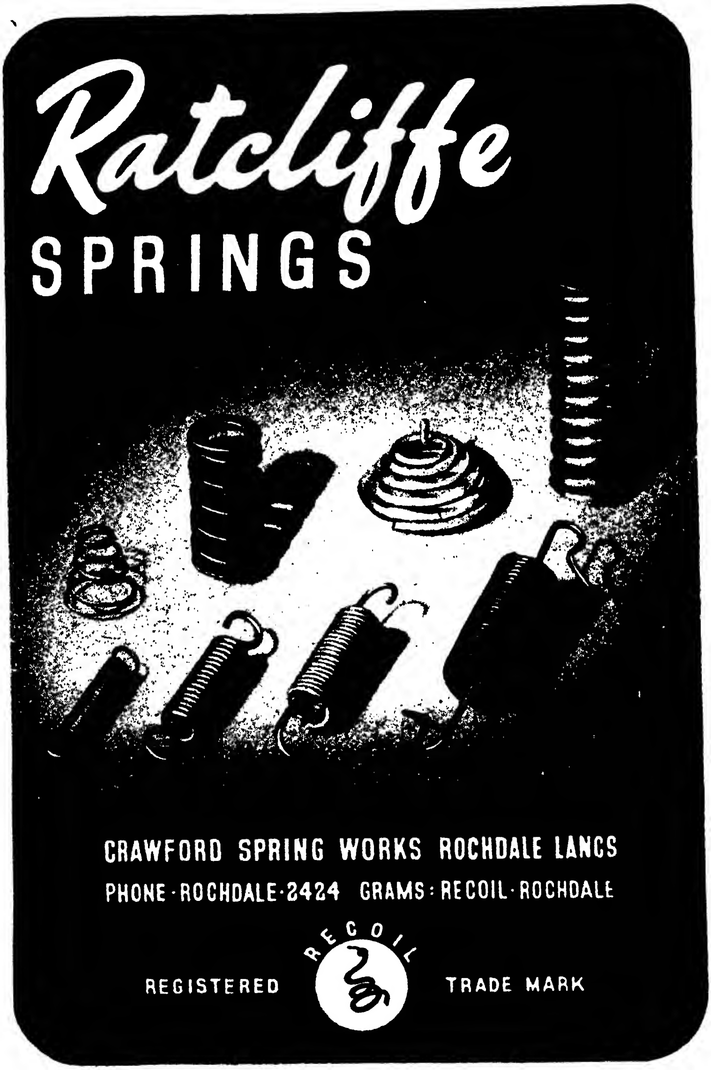

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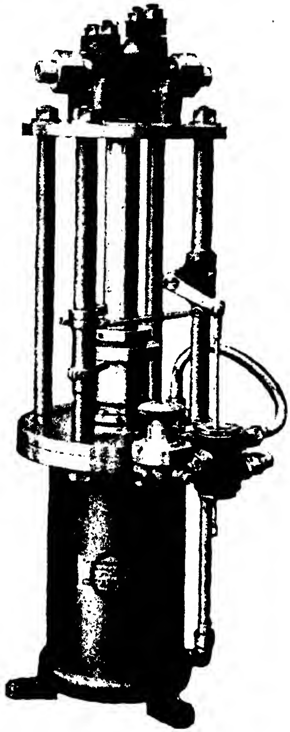

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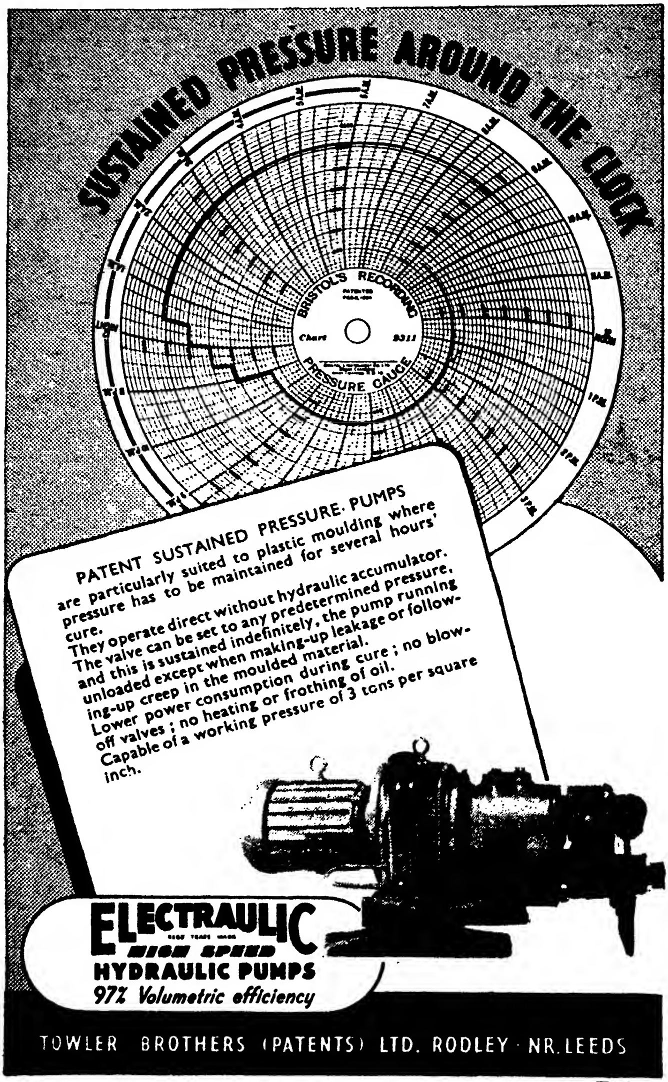
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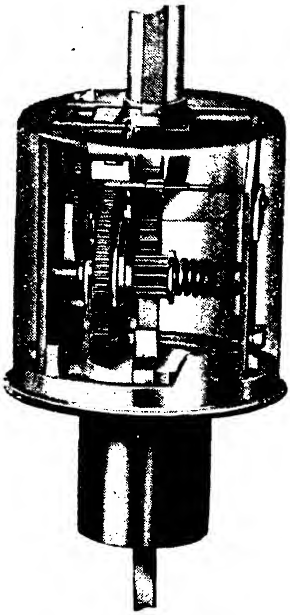

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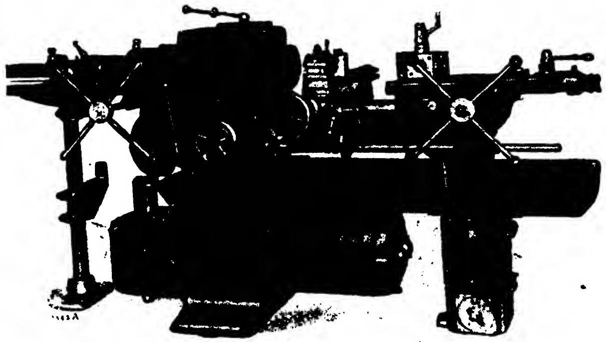

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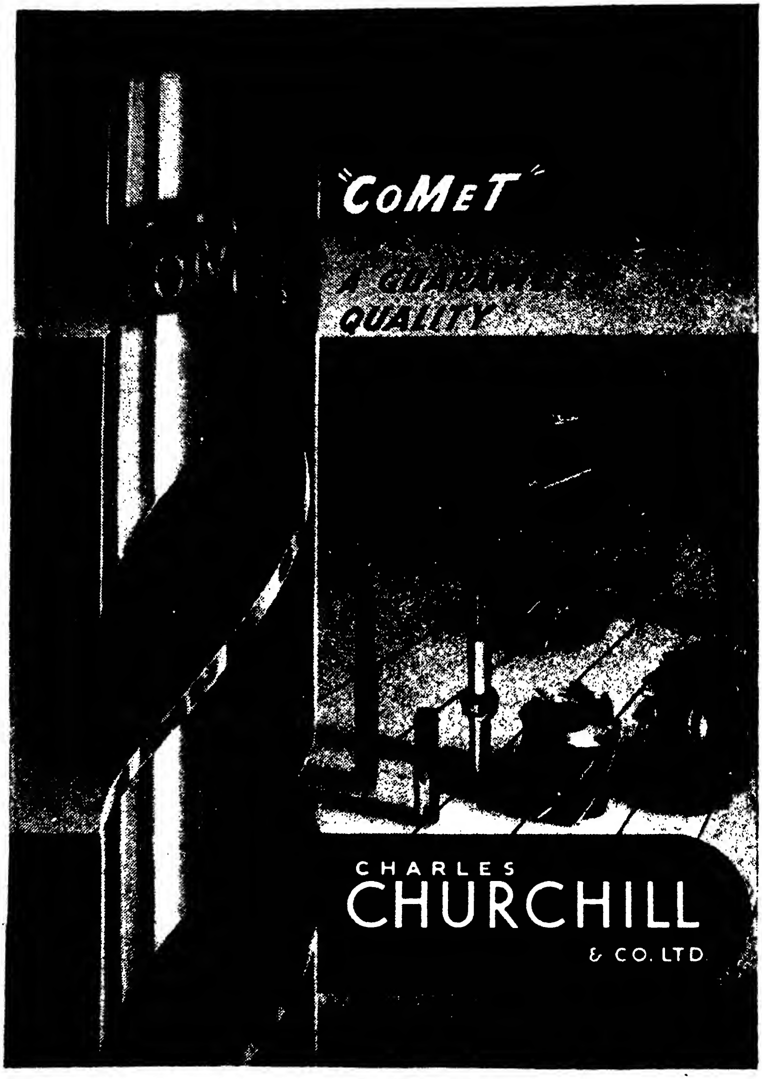
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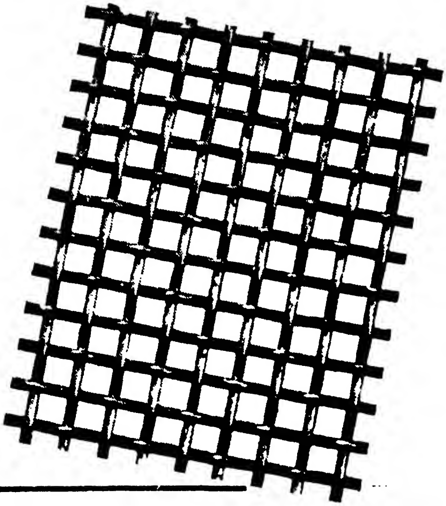

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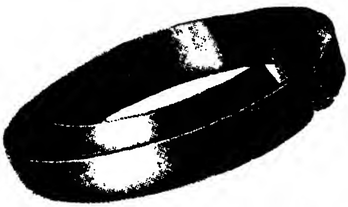

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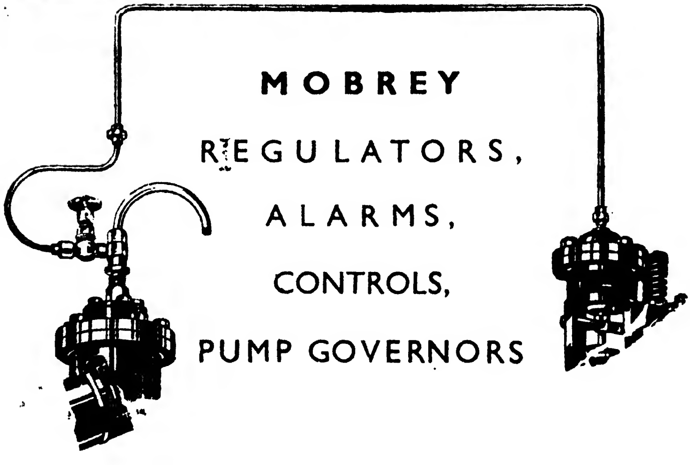

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(*Originally compiled by H. R. KEMPE, M.Inst.C.E., M.I.Mech.E., and W. HANNEFORD-SMITH, F.R.S.E., Assoc. Inst.C.E.*)

55th ANNUAL ISSUE

(IN TWO VOLUMES)

Revised under the direction

of

B. W. PENDRED, M.I.Mech.E., M.I.S.I.

EDITOR-IN-CHIEF of 'THE ENGINEER'

Volume One

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1949

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PREFACE TO FIFTY-FIFTH EDITION, 1949.

SLOWLY the business of publishing 'Kempe' is becoming less troublesome. But normality in the publishing trade is still far off. If we had been able, we should have liked this year to have subjected the whole of 'Kempe' to complete revision. But fate, the printers and the binders forbade it. Were we to have made so wholehearted an attempt, we were assured that 'Kempe' was unlikely to appear at all in 1949, so much delay must full revision have imposed.

But do not let it be thought that we have been idle! Last year, 'Kempe' appeared in a new guise, divided into two volumes with a single index. This year we have found it possible, without creating too great a delay in publication, to make a very complete revision of the second of the two volumes. Moreover, following the modern fashion, we have drawn up a 'plan' for 'Kempe.' This year, as we have just noted, Volume Two has been fully revised and Volume One has received only minor attention. Next year, it will be Volume One that will receive full revision whilst Volume Two gets only the slight revision that a year's developments in engineering may demand. In 1951 we shall rest content only with minor revision of the whole. But in 1952 the cycle will begin again with a full revision of Volume Two.

In this way we plan to ensure that every part of the book will receive proper full scale revision regularly at three year intervals. But there is always, of course, Burns' oft quoted remark about the plans of mice and men. We have not forgotten it! If engineering developments require it —and at times developments have been very rapid—any section that may be concerned will receive attention whether or not the 'plan' requires it to be done that year or not. We intend to keep 'Kempe' up to date so that any engineer consulting it, whether he lives in a Western community with every engineering facility in easy reach, or far from civilisation where he must depend upon the few tools he has available, will have available to guide him information based upon the very latest practice.

B. W. PENDRED.

PREFACE TO FIRST EDITION (1894).

THE importance and value of such a work as THE ENGINEER'S YEAR BOOK need hardly be insisted on. Perhaps there is no professional man who *needs* a YEAR-BOOK more than the Engineer of the present day, seeing that his duties are so multifarious and the requirements of his position so constantly varying. Yet, while other professions have their Year-Books, until now none has been provided for the Engineer; it is confidently believed that the present volume will meet a want that is constantly experienced by Practical Men in every branch of Engineering.

The aim of the compiler has been to produce a work which, being carefully brought up to date, shall take its place as the standard Book of Reference in the profession, and include only such Modern Formulæ, Rules, Tables, and Data as are required by the Engineer from day to day in the practical work of his calling, including Civil, Mechanical, Marine. Electrical, and Mine Engineering. The work is accordingly divided into Thirty-one Sections, each Section forming practically a complete treatise on its particular subject, and the whole embracing the present practice of Engineering.

The great advances made in Science, as applied to Engineering, during the last quarter of a century have fixed immutably many new principles, laws, and practical data. These, as well as the like particulars of older practice, have been carefully compiled from the best authorities, and find their place in THE ENGINEER'S YEAR-BOOK, whilst a large amount of original matter, presented in a handy and concise form, is also included in the volume.

In the work of compilation, most valuable assistance has been rendered by numerous practising Engineers, including those of the principal English Railway Companies, and of the Queensland Government, as well as by

Girder Manufacturers, Boiler and Engine Makers, and many other persons of experience. Every available work has been searched through and consulted for information, but in making use of particulars so obtained the compiler has been at pains to trace the matter to its original source, so as to determine its authoritative value, and in every case where it appeared necessary he has redrawn existing figures and verified current formulæ, so that clearness and accuracy might be ensured.

Obviously a work of this nature, to fulfil its purpose of being kept up to date, will require periodical revision, and it is accordingly intended to REISSUE IT FROM YEAR TO YEAR. The facilities thus afforded for thorough revision throughout the volume will make it practicable to *insert every year new matter of importance in any Section*, and render the book at all times a safe and reliable DAILY REFERENCE BOOK for the PRACTICAL ENGINEER.

Kindly criticism and help towards making the work as complete and as useful as possible are earnestly invited, and any suggestions, addressed to the care of the Publishers, with this object, will be gladly received, and, as far as possible, acted upon.

I have to thank Mr. H. HARTNELL Technical Officer of the Engineer-in-Chief's Office, for great assistance rendered in the work of compilation.

Finally, I think more than a word of thanks is due to the printers, Messrs. SPOTTISWOODE & Co., for the able manner in which they have executed the difficult task of concisely and clearly arranging the letter-press.

H. R. KEMPE.

ENGINEER-IN-CHIEF'S OFFICE,
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January 1894

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A. B. SEARLE . . .	Mortars, Concretes, Cements; Clays and Allied Products; Grinding, Abrasives, and Polishing; and Depreciation of Plant and Machinery.
P. W. SEEWER, D.E. (Zurich), M.I.C.E. .	Water Turbines.
E. STOKOE, A.M.Inst.Pet. .	Lubricating Oils; and Fuels.
H. W. SWIFT, D.Sc., M.I.Mech.E.	Bearing Pressures; Shafting and Shafts; and Belting, Rope Driving and Clutches.
W. J. A. Sykes, A.M.I.Mech.E. .	Electric Rail Traction.
G. W. TOOKEY . . .	Patents, Designs and Trade Marks.
L. C. WATTS, A.C.G.I., M.I.Mech.E. . .	Heating; Ventilation.
NORMAN WHINCUP . . .	Wire Ropes.
B. WOOD, M.A. (Cantab.) .	Gas Turbines.
S. A. WOOD, M.Sc., M.I.Mech.E.	Units of Measurement; Mechanics; Measurement of Power; Heat, Optics; Properties of Engineering Materials: and Metering and Mechanical Testing Appliances.

CONTENTS.

VOLUME ONE

Index to both volumes, Buyers' Guide, and Index to Advertisers are included at the end of Volume Two.

SECTION I

UNITS OF MEASUREMENT (pp. 1–13)

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

SECTION I

UNITS OF MEASUREMENT

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

LENGTH.

English.

Legal Definition of Standard.

The length of the standard yard of 36 inches is such that a pendulum vibrating seconds of mean time, temperature 62° F., at the level of the sea and latitude of Greenwich, has a length of 39·1393 inches.

Equivalent Measures of Length.

Miles.	Furlongs.	Chains.	Rods.	Yards.	Feet.	Inches.
1	8	80	320	1,760	5,280	63,360
·125	1	10	40	220	660	7,920
·0125	·1	1	4	22	66	792
·003125	·025	·25	1	5·5	16·5	198
·00056818	·0045454	·045454	.181818	1	3	36
·00018939	·00151515	·01515151	·060606	·33333	1	12
·000015783	·000126262	·001262626	·0050505	·0277777	·083333	1

1 micro-inch = 1 millionth of an inch.
1 mil = 1 thousandth of an inch.

Gunter's Surveying Chain.

7·92 inches = 1 link. 100 links = 1 chain, 4 rods or 22 yards. 80 chains = 1 mile.

Ropes and Cables.

1 cable-length = 120 fathoms = 720 feet. 1 fathom = 6 feet.

Geographical and Nautical Measures.

1 nautical mile = 6,080 feet = 1·1515 statute miles. 1 degree of a great circle around the equator = 69·17 statute miles = 60·07 nautical miles. 1 point = 11¼ degrees of arc.

Log-line.

The log-line should be about 150 fathoms long and 10 fathoms from the log to the first knot on the line. If half-minute glass is used, it will be 50·66 feet between each succeeding knot. For 28-seconds glass it will be 47·29 feet = 7·88 fathoms per knot. This is the length of knot by calculation but practically it is shortened to 7·5 fathoms per knot for a 28-seconds glass.

1 knot = 1 nautical mile per hour.

Miscellaneous.

10 penny diameters = 1 foot halfpenny = 1 inch ; 5-shilling piece = 1½ inch

INCHES REDUCED TO DECIMALS OF A FOOT.

Ins.	Foot.	Ins.	Foot.	Ins.	Foot.	Ins.	Foot.	Ins.	Foot.	Ins.	Foot.
0	·0000	2	·1667	4	·3333	6	·5000	8	·6667	10	·8333
1/32	·0026	1/32	·1693	1/32	·3359	1/32	·5026	1/32	·6693	1/32	·8359
1/16	·0052	1/16	·1719	1/16	·3385	1/16	·5052	1/16	·6719	1/16	·8385
3/32	·0078	3/32	·1745	3/32	·3411	3/32	·5078	3/32	·6745	3/32	·8411
1/8	·0104	1/8	·1771	1/8	·3438	1/8	·5104	1/8	·6771	1/8	·8438
5/32	·0130	5/32	·1797	5/32	·3464	5/32	·5130	5/32	·6797	5/32	·8464
3/16	·0156	3/16	·1823	3/16	·3490	3/16	·5156	3/16	·6823	3/16	·8490
7/32	·0182	7/32	·1849	7/32	·3516	7/32	·5182	7/32	·6849	7/32	·8516
1/4	·0208	1/4	·1875	1/4	·3542	1/4	·5208	1/4	·6875	1/4	·8542
9/32	·0234	9/32	·1901	9/32	·3568	9/32	·5234	9/32	·6901	9/32	·8568
5/16	·0260	5/16	·1927	5/16	·3594	5/16	·5260	5/16	·6927	5/16	·8594
11/32	·0286	11/32	·1953	11/32	·3620	11/32	·5286	11/32	·6953	11/32	·8620
3/8	·0313	3/8	·1979	3/8	·3646	3/8	·5313	3/8	·6979	3/8	·8646
13/32	·0339	13/32	·2005	13/32	·3672	13/32	·5339	13/32	·7005	13/32	·8672
7/16	·0365	7/16	·2031	7/16	·3698	7/16	·5365	7/16	·7031	7/16	·8698
15/32	·0391	15/32	·2057	15/32	·3724	15/32	·5391	15/32	·7057	15/32	·8724
1/2	·0417	1/2	·2083	1/2	·3750	1/2	·5417	1/2	·7083	1/2	·8750
17/32	·0443	17/32	·2109	17/32	·3776	17/32	·5443	17/32	·7109	17/32	·8776
9/16	·0469	9/16	·2135	9/16	·3802	9/16	·5469	9/16	·7135	9/16	·8802
19/32	·0495	19/32	·2161	19/32	·3828	19/32	·5495	19/32	·7161	19/32	·8828
5/8	·0521	5/8	·2188	5/8	·3854	5/8	·5521	5/8	·7188	5/8	·8854
21/32	·0547	21/32	·2214	21/32	·3880	21/32	·5547	21/32	·7214	21/32	·8880
11/16	·0573	11/16	·2240	11/16	·3906	11/16	·5573	11/16	·7240	11/16	·8906
23/32	·0599	23/32	·2266	23/32	·3932	23/32	·5599	23/32	·7266	23/32	·8932
3/4	·0625	3/4	·2292	3/4	·3958	3/4	·5625	3/4	·7292	3/4	·8958
25/32	·0651	25/32	·2318	25/32	·3984	25/32	·5651	25/32	·7318	25/32	·8984
13/16	·0677	13/16	·2344	13/16	·4010	13/16	·5677	13/16	·7344	13/16	·9010
27/32	·0703	27/32	·2370	27/32	·4036	27/32	·5703	27/32	·7370	27/32	·9036
7/8	·0729	7/8	·2396	7/8	·4063	7/8	·5729	7/8	·7396	7/8	·9063
29/32	·0755	29/32	·2422	29/32	·4089	29/32	·5755	29/32	·7422	29/32	·9089
15/16	·0781	15/16	·2448	15/16	·4115	15/16	·5781	15/16	·7448	15/16	·9115
31/32	·0807	31/32	·2474	31/32	·4141	31/32	·5807	31/32	·7474	31/32	·9141
1	·0833	3	·2500	5	·4167	7	·5833	9	·7500	11	·9167
1/32	·0859	1/32	·2526	1/32	·4193	1/32	·5859	1/32	·7526	1/32	·9193
1/16	·0885	1/16	·2552	1/16	·4219	1/16	·5885	1/16	·7552	1/16	·9219
3/32	·0911	3/32	·2578	3/32	·4245	3/32	·5911	3/32	·7578	3/32	·9245
1/8	·0938	1/8	·2604	1/8	·4271	1/8	·5938	1/8	·7604	1/8	·9271
5/32	·0964	5/32	·2630	5/32	·4297	5/32	·5964	5/32	·7630	5/32	·9297
3/16	·0990	3/16	·2656	3/16	·4323	3/16	·5990	3/16	·7656	3/16	·9323
7/32	·1016	7/32	·2682	7/32	·4349	7/32	·6016	7/32	·7682	7/32	·9349
1/4	·1042	1/4	·2708	1/4	·4375	1/4	·6042	1/4	·7708	1/4	·9375
9/32	·1068	9/32	·2734	9/32	·4401	9/32	·6068	9/32	·7734	9/32	·9401
5/16	·1094	5/16	·2760	5/16	·4427	5/16	·6094	5/16	·7760	5/16	·9427
11/32	·1120	11/32	·2786	11/32	·4453	11/32	·6120	11/32	·7786	11/32	·9453
3/8	·1146	3/8	·2813	3/8	·4479	3/8	·6146	3/8	·7813	3/8	·9479
13/32	·1172	13/32	·2839	13/32	·4505	13/32	·6172	13/32	·7839	13/32	·9505
7/16	·1198	7/16	·2865	7/16	·4531	7/16	·6198	7/16	·7865	7/16	·9531
15/32	·1224	15/32	·2891	15/32	·4557	15/32	·6224	15/32	·7891	15/32	·9557
1/2	·1250	1/2	·2917	1/2	·4583	1/2	·6250	1/2	·7917	1/2	·9583
17/32	·1276	17/32	·2943	17/32	·4609	17/32	·6276	17/32	·7943	17/32	·9609
9/16	·1302	9/16	·2969	9/16	·4635	9/16	·6302	9/16	·7969	9/16	·9635
19/32	·1328	19/32	·2995	19/32	·4661	19/32	·6328	19/32	·7995	19/32	·9661
5/8	·1354	5/8	·3021	5/8	·4688	5/8	·6354	5/8	·8021	5/8	·9688
21/32	·1380	21/32	·3047	21/32	·4714	21/32	·6380	21/32	·8047	21/32	·9714
11/16	·1406	11/16	·3073	11/16	·4740	11/16	·6406	11/16	·8073	11/16	·9740
23/32	·1432	23/32	·3099	23/32	·4766	23/32	·6432	23/32	·8099	23/32	·9766
3/4	·1458	3/4	·3125	3/4	·4792	3/4	·6458	3/4	·8125	3/4	·9792
25/32	·1484	25/32	·3151	25/32	·4818	25/32	·6484	25/32	·8151	25/32	·9818
13/16	·1510	13/16	·3177	13/16	·4844	13/16	·6510	13/16	·8177	13/16	·9844
27/32	·1536	27/32	·3203	27/32	·4870	27/32	·6536	27/32	·8203	27/32	·9870
7/8	·1563	7/8	·3229	7/8	·4896	7/8	·6563	7/8	·8229	7/8	·9896
29/32	·1589	29/32	·3255	29/32	·4922	29/32	·6589	29/32	·8255	29/32	·9922
15/16	·1615	15/16	·3281	15/16	·4948	15/16	·6615	15/16	·8281	15/16	·9948
31/32	·1641	31/32	·3307	31/32	·4974	31/32	·6641	31/32	·8307	31/32	·9974

Metric System.

Metric Unit.	Ins.	Feet.	Yards.	Miles.
Millimetre*	·03937	·003281		
Centimetre†	·39370113	·032808		
Decimetre	3·9370113	·3280843	·10936	
Metre ‡	39·370113	3·280843	1·0936143	
Decametre	393·70113	32·80843	10·936143	
Hectometre	Road	328·0843	109·36143	·06213718
Kilometre	measures.	3280·843	1093·6143	·6213718
Myriametre		32808·43	10936·143	6·213718

* Nearly the $\frac{1}{25}$th part of an inch. † Full $\frac{3}{8}$ inch.

‡ Very nearly 3 ft. $3\frac{3}{8}$ ins., which is too long by only 1 part in 8,056.

1 micron (otherwise 'μ') = 0·001 millimetre.

Relation between English and Metric Units.

The legal relations of English Imperial to International Metric Measures are thus given by the Standards Office of the Board of Trade:

1 inch = 25·399978 mm.; 1 metre = 39·370113 inches; 1 gallon = 4·5459631 litres; 1 litre = 1·75980 pint; 1 pound = 0·45359243 kilogram; 1 kilogram = 2·2046223 pounds.

In 1893 the United States of America recognised the metre as the fundamental standard of length, and adopted the conversion value of 1 metre = 39·37 inches.

Conversion of English Inches into Centimetres.

Ins.	0	1	2	3	4	5	6	7	8	9
	Cm.	Cm.	Cm.	Cm.	Cm.	Cm.	Cm.	Cm.	Cm	Cm.
0	·00	2·54	5·08	7·62	10·16	12·70	15·24	17·78	20·32	22·86
10	25·40	27·94	30·48	33·02	35·56	38·10	40·64	43·18	45·72	48·26
20	50·80	53·34	55·88	58·42	60·96	63·50	66·04	68·58	71·12	73·66
30	76·20	78·74	81·28	83·82	86·36	88·90	91·44	93·98	96·52	99·06
40	101·60	104·14	106·68	109·22	111·76	114·30	116·84	119·38	121·92	124·46
50	127·00	129·54	132·08	134·62	137·16	139·70	142·24	144·78	147·32	149·86
60	152·40	154·94	157·48	160·02	162·56	165·10	167·64	170·18	172·72	175·26
70	177·80	180·34	182·88	185·42	187·96	190·50	193·04	195·58	198·12	200·66
80	203·20	205·74	208·28	210·82	213·36	215·90	218·44	220·98	223·52	226·06
90	228·60	231·14	233·68	236·22	238·76	241·30	243·84	246·38	248·92	251·46
100	254·00	256·54	259·08	261·62	264·16	266·70	269·24	271·78	274·32	276·86

The above table is correct throughout to within 1 part in 1,000,000.

Conversion of Centimetres into English Inches.

Cm.	0	1	2	3	4	5	6	7	8	9
	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
0	·0000	·3937	·7874	1·1811	1·5748	1·9685	2·3622	2·7559	3·1496	3·5433
10	3·9370	4·3307	4·7244	5·1181	5·5118	5·9055	6·2992	6·6929	7·0866	7·4803
20	7·8740	8·2677	8·6614	9·0551	9·4488	9·8425	10·236	10·630	11·024	11·417
30	11·811	12·205	12·598	12·992	13·386	13·780	14·173	14·567	14·961	15·354
40	15·748	16·142	16·535	16·929	17·323	17·717	18·110	18·504	18·898	19·291
50	19·685	20·079	20·472	20·866	21·260	21·654	22·047	22·441	22·835	23·228
60	23·622	24·016	24·409	24·803	25·197	25·591	25·984	26·378	26·772	27·165
70	27·559	27·953	28·346	28·740	29·134	29·528	29·921	30·315	30·709	31·102
80	31·496	31·890	32·283	32·677	33·071	33·465	33·858	34·252	34·646	35·039
90	35·433	35·827	36·221	36·614	37·008	37·402	37·795	38·189	38·583	38·976
100	39·370	39·764	40·158	40·551	40·945	41·339	41·732	42·126	42·520	42·912

INCHES
1/64
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5/32
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15/64
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55/64
7/8
57/64
29/32
59/64
15/16
61/64
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63/64
1

MILLIMETRES
1/2
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1 1/2
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2 1/2
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INCHES
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1 1/32
1 3/64
1 1/16
1 5/64
1 3/32
1 7/64
1 1/8
1 9/64
1 5/32
1 11/64
1 3/16
1 13/64
1 7/32
1 15/64
1 1/4
1 17/64
1 9/32
1 19/64
1 5/16
1 21/64
1 11/32
1 23/64
1 3/8
1 25/64
1 13/32
1 27/64
1 7/16
1 29/64
1 15/32
1 31/64
1 1/2
1 33/64
1 17/32
1 35/64
1 9/16
1 37/64
1 19/32
1 39/64
1 5/8
1 41/64
1 21/32
1 43/64
1 11/16
1 45/64
1 23/32
1 47/64
1 3/4
1 49/64
1 25/32
1 51/64
1 13/16
1 53/64
1 27/32
1 55/64
1 7/8
1 57/64
1 29/32
1 59/64
1 15/16
1 61/64
1 31/32
1 63/64
2

MILLIMETRES
26
26 1/2
27
27 1/2
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28 1/2
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29 1/2
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30 1/2
31
31 1/2
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32 1/2
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49 1/2
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50 1/2
51

29/32 inch is shown to be equal to 23 millimetres; carried to three decimal places the correct value would be 23·019 millimetres.
(*O. H. W. 'Machinery,' June 26, 1919.*)

Conversion of English Feet into Metres.

Ft.	0	1	2	3	4	5	6	7	8	9
	Metres.	Metres.	Metres.	Metres.	Metres.	Metres.	Metres.	Metres.	Metres.	Metres.
0	·0000	·3048	·6096	·9144	1·2192	1·5240	1·8288	2·1336	2·4384	2·7432
10	3·0480	3·3528	3·6576	3·9624	4·2672	4·5720	4·8768	5·1816	5·4864	5·7912
20	6·0960	6·4008	6·7056	7·0104	7·3152	7·6200	7·9248	8·2296	8·5344	8·8392
30	9·1440	9·4488	9·7536	10·058	10·363	10·668	10·973	11·278	11·582	11·887
40	12·192	12·497	12·802	13·106	13·411	13·716	14·021	14·326	14·630	14·935
50	15·240	15·545	15·850	16·154	16·459	16·764	17·069	17·374	17·678	17·983
60	18·288	18·593	18·898	19·202	19·507	19·812	20·117	20·422	20·726	21·031
70	21·336	21·641	21·946	22·250	22·555	22·860	23·165	23·470	23·774	24·079
80	24·384	24·689	24·994	25·298	25·603	25·908	26·213	26·518	26·822	27·127
90	27·432	27·737	28·042	28·346	28·651	28·956	29·261	29·566	29·870	30·175
100	30·480	30·785	31·090	31·394	31·699	32·004	32·309	32·614	32·918	33·223

Conversion of Metres into English Feet.

Metres.	0	1	2	3	4	5	6	7	8	9
	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.
0	·0000	3·2808	6·5617	9·8425	13·123	16·404	19·685	22·966	26·247	29·528
10	32·808	36·089	39·370	42·651	45·932	49·213	52·493	55·774	59·055	62·336
20	65·617	68·898	72·179	75·459	78·740	82·021	85·302	88·583	91·864	95·144
30	98·425	101·71	104·99	108·27	111·55	114·83	118·11	121·39	124·67	127·95
40	131·23	134·51	137·80	141·08	144·36	147·64	150·92	154·20	157·48	160·76
50	164·04	167·32	170·60	173·88	177·17	180·45	183·73	187·01	190·29	193·57
60	196·85	200·13	203·41	206·69	209·97	213·25	216·54	219·82	223·10	226·38
70	229·66	232·94	236·22	239·50	242·78	246·06	249·34	252·62	255·91	259·19
80	262·47	265·75	269·03	272·31	275·59	278·87	282·15	285·43	288·71	292·00
90	295·28	298·56	301·84	305·12	308·40	311·68	314·96	318·24	321·52	324·80
100	328·08	331·37	334·65	337·93	341·21	344·49	347·77	351·05	354·33	357·61

Conversion of English Statute Miles into Kilometres.

Miles.	0	1	2	3	4	5	6	7	8	9
	Kiloms.	Kiloms.	Kiloms.	Kiloms.	Kiloms.	Kiloms.	Kiloms.	Kiloms.	Kilom.	Kilom.
0	·0000	1·6093	3·2187	4·8280	6·4374	8·0467	9·6561	11·265	12·875	14·484
10	16·093	17·703	19 312	20·921	22·531	24·140	25·749	27 359	28·968	30·578
20	32·187	33·796	35·406	37·015	38.624	40·234	41·843	43 452	45·062	46·671
30	48·280	49·890	51·499	53·108	54·718	56·327	57·936	59·546	61·155	62·764
40	64·374	65·983	67·592	69·202	70·811	72·420	74·030	75·639	77·248	78·858
50	80·467	82·076	83·686	85·295	86·905	88·514	90·123	91·733	93·342	94.951
60	96·561	98·170	99·779	101·39	103·00	104·61	106·22	107·83	109·44	111.04
70	112·65	114·26	115·87	117·48	119·09	120·70	122·31	123 92	125·53	127.14
80	128·75	130.36	131·97	135 58	135·18	136·79	138·40	140.01	141·62	143 23
90	144·84	146·45	148·06	149·67	151·28	152·89	154·50	156.11	157·72	159 32
100	160·93	162·54	164·15	165·76	167·37	168·98	170·59	172.20	173·81	175·42

Conversion of Kilometres into English Statute Miles.

Kiloms.	0	1	2	3	4	5	6	7	8	9
	Miles.	Miles.	Miles.	Miles.	Miles.	Miles.	Miles.	Miles.	Miles.	Miles
0	·0000	·6214	1·2427	1·8641	2·4855	3·1069	3·7282	4·3496	4·9710	5·5923
10	6·2137	6·8351	7·4565	8·0778	8·6992	9·3206	9·9419	10·563	11·185	11·806
20	12·427	13·049	13·670	14·292	14·913	15·534	16·156	16·777	17·398	18·020
30	18·641	19·263	19·884	20·505	21·127	21·748	22·369	22·991	23·612	24·234
40	24·855	25·476	26·098	26·719	27·340	27·962	28·583	29·204	29·826	30·447
50	31·069	31·690	32·311	32·933	33·554	34·175	34·797	35·418	36·040	36·661
60	37·282	37·904	38·525	39·146	39·768	40·389	41·011	41·632	42·253	42·875
70	43·496	44·117	44·739	45·360	45·982	46·603	47·224	47·846	48·467	49·088
80	49·710	50·331	50·952	51·574	52·195	52·817	53·438	54·059	54·681	55·302
90	55·923	56·545	57·166	57·788	58·409	59·030	59·652	60·273	60·894	61·516
100	62·137	62·759	63·380	64·001	64·623	65·244	65·865	66·487	67·108	67·730

AREA.

English.

Sq. Miles.	Acres.	Sq. Chains.	Sq. Rods.	Sq. Yards.	Sq. Feet.	Sq. Inches.
1	640	6,400	102,400	3,097,600	27,878,400	4,014,489,600
001562	1	10	160	4,840	43,560	6,272,640
·0001562	·1	1	16	484	4,356	627,264
·000009766	·00625	·0625	1	30·25	272·25	39,204
·000000323	·0002066	·002066	·033	1	9	1,296
·0000000359	·00002296	·0002296	·00367	·1111111	1	144
·00000000025	·000000159	·00000159	·00002551	·0007716	·006945	1

Metric.

Metric Unit.	Sq. Ins.	Sq. Ft.	Sq. Yards.	Acres.
Sq. Millimetre .	·00155	·00001076	·0000012	
Sq. Centimetre .	·1550	·0010764	·0001196	
Sq. Decimetre .	15·50006	·1076393	·0119599	
Sq. Metre, or Centiare	1550·006	10·76393	1·195992	·000247
Sq. Decametre, or Are	1550006	1076·393	119·5992	·024711
Sq. Hectometre or Hectare . .	—	107639·3	11959·92	2·471058
Sq. Kilometre .	·3861029 sq.	—	1195992	247·1058
Sq. Myriametre .	38·61029 miles	—	—	24710·58

WEIGHT.

English.

Legal Definition of Standard.—The *pound* is the weight in a vacuum of a platinum cylinder called the *Imperial Standard Pound.*

One cubic inch of distilled water (weighed in air against brass weights) at temperature 62° F., the barometer being at 30 ins., weighs 252·326 grains, of which 7,000 go to the pound avoirdupois, and 5,760 to the pound troy.

AVOIRDUPOIS.

Ton.	Cwts.	Pounds.	Ounces.	Drams.
1	20	2240	35840	573440
·05	1	112	1792	28672
·00044643	·0089286	1	16	256
·0000279	·000558	·0625	1	16
·000001744	·00003488	·003906	·0625	1

TROY.

Pound.	Ounces.	Dwts.	Grains.	Pound Avoir.
1	12	240	5760	·822857
·083333	1	20	480	·068571
·004166	·05	1	24	·00342857
·0001736	·002083333	·0416666	1	·000142857
1·215278	14·58333	291·6666	7000	1

APOTHECARIES'.

Pound.	Ounces.	Drams.	Scruples.	Grains.
1	12	96	288	5760
·08333	1	8	24	480
·01041666	·125	1	3	60
·0034722	·0416666	·3333	1	20
·00017361	·0020833	·016666	·05	1

Weight of 3 pennies, 1 crown, or 5 shillings each = 1 oz.

N.B.—The avoirdupois, troy, and apothecaries' *grain* are of the same weight.

Conversion of Pounds to Tons.
Tons = Pounds × 0·00044643.

Pounds.	Tons.	Pounds.	Tons.	Pounds.	Tons.	Pounds.	Tons.
1,000	0·446	24,000	10·714	47,000	20·982	88,000	39·286
2,000	0·893	25,000	11·161	48,000	21·429	90,000	40·179
3,000	1·339	26,000	11·607	49,000	21·875	92,000	41·072
4,000	1·786	27,000	12·053	50,000	22·321	94,000	41·964
5,000	2·232	28,000	12·500	51,000	22·768	96,000	42·857
6,000	2·679	29,000	12·946	52,000	23·214	98,000	43·750
7,000	3·125	30,000	13·393	54,000	24·107	100,000	44·643
8,000	3·571	31,000	13·839	56,000	25·000	110,000	49·107
9,000	4·018	32,000	14·286	58,000	25·893	120,000	53·572
10,000	4·464	33,000	14·732	60,000	26·786	130,000	58·036
11,000	4·911	34,000	15·179	62,000	27·679	140,000	62·500
12,000	5·357	35,000	15·625	64,000	28·572	150,000	66·965
13,000	5·804	36,000	16·071	66,000	29·464	160,000	71·429
14,000	6·250	37,000	16·518	68,000	30·357	170,000	75·893
15,000	6·696	38,000	16·964	70,000	31·250	180,000	80·357
16,000	7·143	39,000	17·411	72,000	32·143	190,000	84·822
17,000	7·589	40,000	17·857	74,000	33·036	200,000	89·286
18,000	8·036	41,000	18·304	76,000	33·929	210,000	93·750
19,000	8·482	42,000	18·750	78,000	34·822	220,000	98·215
20,000	8·929	43,000	19·196	80,000	35·714	230,000	102·660
21,000	9·375	44,000	19·643	82,000	36·607	240,000	107·140
22,000	9·821	45,000	20·089	84,000	37·500	250,000	111·610
23,000	10·268	46,000	20·536	86,000	38·393	260,000	116·072

Weight—Metric.

Legal Definition of Standard.—The *gramme* is $\frac{1}{1000}$ of the *International Prototype Kilogramme* which is the mass of a cylinder of platinum-iridium kept at Sèvres.

1 gramme = 15·432356 grains, and is the weight of a millilitre of distilled water at 4° Centigrade, or 39·2° Fahr., and 760 mm. of mercury pressure.

Weight.	Grammes.	Avoirdupois Ounces.	Avoirdupois Lbs.	Cwts.	Ton.	Grains.
Milligramme .	·001	—	—	—	—	·015
Centigramme .	.01	—	—	—	—	·154
Decigramme .	·1	—	—	—	—	1·543
Gramme . .	1	·035	·0022	—	—	15·432356
Decagramme .	10	·35	·022			
Hectogramme	100	3·527	·22046			
Kilogramme .	1000	35·2739	2·204622	·019	·00098	
Myriagramme	10000	—	22·04622	·1968	·00984	
Quintal . .	100000	—	220·4622	1·9684	·0984	
Millier or Bar.	1000000	—	2204·622	19·68413	·984206	

Conversion of Metric to English Weights.

Grammes to Grains.		Kilogrammes to Ounces. (Avoir.)		Kilogrammes to Pounds. (Avoir.)	
1=15·432356	6=92·594138	1=35·27396	6=211·64374	1=2·204622	6=13·227734
2=30·864713	7=108·02649	2=70·54791	7=246·01770	2=4·409245	7=15·432356
3=46·297069	8=123·45885	3=105·82187	8=282·19166	3=6·613867	8=17·636978
4=61·729426	9=138·89121	4=141·09583	9=317·46561	4=8·818489	9=19·841601
5=77·161782		5=176·36978		5=11·023111	

A milligramme is about $\frac{1}{65}$ grain.

Conversion of English to Metric Weights.

Grains to Grammes.	Ounces (Avoir.) to Grammes.	Pounds (Avoir.) to Kilogrammes.	Hundredweights to Kilogrammes.
1 = ·0647989	1 = 28·34953	1 = ·4535924	1 = 50·80235
2 = ·1295978	2 = 56·69905	2 = ·9071849	2 = 101·6047
3 = ·1943968	3 = 85·04858	3 = 1·360777	3 = 152·4071
4 = ·2591957	4 = 113·3981	4 = 1·814370	4 = 203·2094
5 = ·3239946	5 = 141·7476	5 = 2·267962	5 = 254·0118
6 = ·3887935	6 = 170·0972	6 = 2·721555	6 = 304·8141
7 = ·4535924	7 = 198·4467	7 = 3·175147	7 = 355·6165
8 = ·5183914	8 = 226·7962	8 = 3·628739	8 = 406·4188
9 = ·5831903	9 = 255·1457	9 = 4·082332	9 = 457·2212

The Metric Carat.

By decision of the Standards Department of the Board of Trade, the metric carat of 200 milligrams, and its necessary multiples and sub-multiples, are standard weights in the United Kingdom.

CAPACITY.

English.

Legal Definition of Standard.

Gallon.- The standard Imperial gallon (*International Bureau of Standards, Paris,* 1921) contains 10 lbs. (avoir.) weight of distilled water (weighed in air against brass weights) at the temperature of 62° F., the barometer being at 30 ins.

1 Imperial gallon = 277·420 cubic inches. 1 U.S. gallon = 231·0 cubic inches.

Dry Measure.

Cu. Yard.	Bushels.	Cu. Feet.	Pecks.	Gallons.	Cu. Ins.
1	21·0223	27	84·089	168·179	46656
·047568	1	1·28435	4	8	2219·35
·037037	·7786	1	3·1144	6·22884	1728
·011892	·25	·32109	1	2	554·838
	·125	·16054	·5	1	277·420
		·000579	·001802	·0036047	1

Liquid Measure.

Gallon.	Quarts.	Pints.	Gills.	Cub. Ins.
1	4	8	32	277·420
·25	1	2	8	69·355
·125	·5	1	4	34·677
·03125	·125	·25	1	8·6693
·0036047	·014419	·028837	·11535	1

Apothecaries' Measures.

Gallon (C) = 8 pints (O); 1 pint = 20 fluid ounces (oz. weight of water).
Ounce (f℥) = 8 drachms (fʒ) = 480 minims (♏) = 720 drops (gtt).
One wine-glass = 4 tablespoonfuls = 16 teaspoonfuls = 2 ounces.
Symbols.—f. or fl., fluid; ss., one half; aa., for each. Thus, f℥ss. means ½ a fluid ounce.

Metric.

Definition of Standard.—The litre, the metric standard of volume, is the volume occupied by 1 kg. of distilled water at 4° C. and 760 mm. of mercury pressure.

1 litre (1,000 millilitres) = 1,000·028 cubic centimetres.

The original standard kilogramme was intended to be equal to the mass of one cubic decimetre of water under certain standard conditions. It was later found that the standard differed slightly from this original definition, but it was decided to adhere to the standard as the unit of mass, and to define the litre afresh as above.

The difference between the millilitre and the cubic centimetre is thus negligible for ordinary purposes.

CONVERSION OF METRIC TO ENGLISH CAPACITIES.

Cub. Centimetres to Cub. Ins.	Litres to Fluid Ounces.	Litres to Pints.	Litres to Gallons.
1 = ·061024	1 = 35·1961	1 = 1·75980	1 = ·219975
2 = ·122048	2 = 70·3921	2 = 3·51961	2 = ·439951
3 = ·183072	3 = 105·5882	3 = 5·27941	3 = ·659926
4 = ·244096	4 = 140·7842	4 = 7·03921	4 = ·879902
5 = ·305120	5 = 175·9803	5 = 8·79902	5 = 1·099877
6 = ·366143	6 = 211·1764	6 = 10·55882	6 = 1·319852
7 = ·427167	7 = 246·3724	7 = 12·31862	7 = 1·539828
8 = ·488191	8 = 281·5685	8 = 14·07842	8 = 1·759803
9 = ·549215	9 = 316·7646	9 = 15·83823	9 = 1·979779

CONVERSION OF ENGLISH TO METRIC CAPACITIES.

Cub. Ins. to Cub. Centimetres.	Fluid Ounces to Cubic Centimetres.	Pints to Litres.	Gallons to Litres.
1 = 16·3870	1 = 28·413	1 = ·568245	1 = 4·54595
2 = 32·7740	2 = 56·826	2 = 1·136491	2 = 9·09193
3 = 49·1611	3 = 85·239	3 = 1·704736	3 = 13·63789
4 = 65·5481	4 = 113·652	4 = 2·272982	4 = 18·18385
5 = 81·9351	5 = 142·065	5 = 2·841227	5 = 22·72982
6 = 98·3221	6 = 170·478	6 = 3·409472	6 = 27·27578
7 = 114·7091	7 = 198·891	7 = 3·977718	7 = 31·82174
8 = 131·0962	8 = 227·304	8 = 4·545963	8 = 36·36770
9 = 147·4832	9 = 255·717	9 = 5·114209	9 = 40·91367

COMPARISON OF VARIOUS FOREIGN AND ENGLISH MEASURES.

Country.	Measure.	English Value.	Country.	Measure.	English Value.
America	Bushel	2150·4 cu. ins.	Germany	Morgen	27,484 sq. feet
"	Gallon (dry)	268·80 cu. ins.	"	Pfund	1·01 to 1·23 lbs.
"	" (fluid)	231 cu. ins.	India	Seer	32·914 oz.
"	Peck	537·605 cu. ins.	"	Tola	180 grains
"	Pint	28·875 cu. ins.	Netherlands	Myle	3,281 feet
"	Quart	57·750 cu. ins.	Norway	Centner	110·231 lbs.
Austria	Joch	1·422 acres	"	Mile	4·6803 miles
"	Klafter	6·2213 feet	"	Morgen	27,484 sq. feet
Bengal	Maund	82·2 lbs.	Russia	Archine	28 inches
Canada	Mino	2,339 cu. ins.	"	Dessatine	2·69977 acres
China	Catty or Gin	1·3333 lbs.	"	Dola	·68573 grains
"	Li	608·5 yds.	"	Pud or Pood	36·112 lbs.
"	Picul	133·33 lbs.	"	Sagene	7 feet
"	Tael	583 grains	"	Verst	3,500 feet
Denmark	Centner	110·231 lbs.	Spain	Arroba	25·35 lbs.
"	Mile	4·6803 miles	"	"	2·77 gallons
"	Morgen	27,484 sq. ft.	Sweden	Mile	6·64 miles
France	Aune	47·245 inches	"	Tunnland	1·2198 acres
"	Brasse	5·328 feet	Turkey	Arhoin	26·9 inches
"	Lieu	6,077 yds.	"	Oke	2·829 lbs.
Germany	Centner	110·231 lbs.	"	Pic	25·69 inches
"	Mile	4·6803 miles	Vienna	Centne	123·46 lbs.

VARIOUS FOREIGN WEIGHTS AND MEASURES WITH THEIR APPROXIMATE BRITISH EQUIVALENTS.

(Compiled by the Board of Trade (July 1923) from information published by the respective Governments.)

[NOTE.—In the case of fractional equivalents the last figure shown is uncertain.]

Country.	Length.	Area.	Weight.	Capacity.	Approximate British Equivalent.
1. United States of America	—	—	Barrel (flour)	—	196 lbs. avoir.
	—	—	Short Ton	—	2,000 lbs. avoir.
	—	—	Long ,,	—	2,240 ,, ,,
	—	—	—	Bushel (Winchester)—dry	0·96897 imperial bushel.
	—	—	—	Gallon (Old English)—wet	0·83268 ,, gallon.
	Metre	—	—	—	1·09361 yards.
	Kilometre	—	—	—	0·62137 mile.
2. Austria	—	Sq. Kilometre = 100 Hectares	—	—	0·38610 square mile.
	—	Are	—	—	0·024711 acre.
	—	Hectare	—	—	2·4711 acres.
	—	—	Kilogramme	—	2·20462 lbs. avoir.
	—	—	Quintal métrique (Centner) = 100 kilogs	—	220·462 lbs. avoir.
	—	—	Tonne	—	2204·62 lbs. avoir.
	—	—	—	Cubic metre	1·30795 cubic yards.
	—	—	—	Hectolitre (liquid)	21·9975 imperial gallons.
	—	—	—	,, (cereals)	2·74969 ,, bushels.
3. Belgium	—	As No. 2	—	—	—
4. Bulgaria	—	,, ,,	—	—	—
5. China	Ch'ih (= 10 Ts'un)	—	—	—	*14·1 inches.
	Chang (= 10 Ch'ih)	—	—	—	*11·75 feet.
	—	—	Tael (Liang)	—	*1⅓ ozs. avoir.
	—	—	Catty	—	*1⅓ lbs. avoir.
	—	—	Picul	—	*133⅓ lbs. avoir.
6. Czechoslovakia	—	As No. 2	—	—	—
7. Denmark	Weights and measures as No. 2 have been compulsory since April 1922.				
8. Egypt	Pic for textiles (0·58 metres) (Diraa Baladi)	—	—	—	22·83466 inches
	—	Feddan (land)	—	—	1·038 acres.
	—	—	Oke (400 Dirhems) = 1·248 kilogs.	—	2·751367 lbs. avoir.
	—	—	Kantar (36 Oke) = 44·928 kilogs.	—	99·049272 ,,
	—	—	—	Ardeb (198 litres)—dry	5·444391 bushels.
9. Esthonia	—	As No. 2	Russian weights and measures as No. 24 also in use.		
10. Finland	—	,, ,,	—	—	—
11. France	—	,, ,,	—	—	—
12. Germany	—	,, ,,	—	—	—
13. Greece	Metric system (as No. 2) compulsory since 1922.		Ocque = 1·280 kilogs.	—	2·82 lbs. avoir.
			Stater (Quintal = 56·320 kilogs.	—	124·16 ,,

13. Greece—*cont.*	—	—	Livre (Great Venetian)	—	1·06 lb. avoir.
	—	—	Drachme (Dram)	—	⅛ oz. avoir.
14. Holland	—	As No. 2	—	—	—
15. Hungary	—	,, ,,	—	—	—
16. Italy	—	,, ,,	—	—	—
	Ri	—	—	—	2·4403 miles.
	Cho	—	—	—	5·423 chains.
	Ken	—	—	—	1·9834 yards.
	—	Square Ri	—	—	5·95505 square miles.
	—	Cho (land)	—	—	2·4507 acres.
	—	Tsubo	—	—	3·9538 square yards.
17. Japan	—	—	Kwan	—	8·26733 lbs. avoir.
	—	—	Kin	—	1·32277 ,,
	—	—	—	Koku (wet)	39·7033 gallons.
	—	—	—	,, (dry)	4·9629 bushels.
	—	—	—	,, (capacity of vessel)	$\frac{1}{10}$ of a measurement ton.
	—	—	—	Sho (wet)	1·5881 quarts.
	—	—	—	,, (dry)	0·1985 peck.
18. Latvia	As No. 2.	—			
19. Lithuania	Russian system as No. 24, but at same time metric system is in use. Arrangements are being made for making latter compulsory.				
20. Norway	—	As No. 2	—	—	—
	Lokieć	—	—	—	0·62992 yard.
	—	Square lokieć (łokieć kwadretowe)	—	—	0·39680 square yard.
	—	Morg	—	—	1·3835 acres.
21. Poland	—	—	Funt	—	0·893983 lb. avoir.
	—	—	Center = 100 funt	—	89·3983 lbs.
	—	—	—	Cubic lokieć (łokieć szescienne)	0·24995 cubic yard.
	—	—	—	Kwarta (wet and dry) = 1 litre	0·21998 gallon.
22. Portugal	—	As No. 2	—	—	—
23. Roumania	—	,, ,,	—	—	—
	Verst	—	—	—	0·6629 mile.
	—	Square verst	—	—	0·4394 square mile.
	—	Dessiatine	—	—	2·6997 acres.
24. Russia	—	—	Pood	—	36·113 lbs. avoir.
	—	—	Berkovets (= 10 pood)	—	361·13 ,,
	—	—	—	Tchetverik (dry)	5·7717 imperial bushels.
	—	—	—	Vedro (wet)	2·705 ,, gallons
25. Jugoslavia	As No. 2	—	—	—	—
26. Spain	,, ,,	—	—	—	—
27. Sweden	,, ,,	—	—	—	—
28. Switzerland	,, ,,	—	—	—	—

* As fixed for foreign trade by treaty between Great Britain and China in 1902. Customary weights and measures of the same name vary in every province and trade in China.

PRESSURE.

Conversion of Kilogrammes per Square Centimetre to Pounds per Square Inch.

Kilos. per Sq. Cm.	0	·1	·2	·3	·4	·5	·6	·7	·8	·9
	Pounds per Square Inch.									
0	0·000	1·4223	2·8447	4·2670	5·6893	7·1117	8 5340	9·9563	11·379	12·801
1	14·223	15·646	17·068	18·490	19·913	21·335	22·757	24·180	25·602	27·024
2	28·447	29·869	31·291	32·714	34·136	35·558	36·981	38·403	39·825	41·248
3	42·670	44·092	45·515	46·937	48·359	49·782	51·204	52·626	54·049	55·471
4	56·893	58·316	59·738	61·160	62·583	64·005	65·427	66·850	68·272	69·694
5	71·117	72·539	73·961	75·384	76·806	78·228	79 651	81·073	82·495	83·918
6	85·340	86·762	88·185	89·607	91·029	92·452	93·874	95·296	96·719	98 141
7	99·563	100·99	102·41	103·83	105·25	106·67	108·10	109·52	110·94	112·36
8	113·79	115·21	116·63	118·05	119·48	120·90	122 32	123·74	125·17	126·59
9	128·01	129·43	130·85	132·28	133·70	135·12	136·54	137·97	139·39	140·81
10	142·23	143·66	145·08	146·50	147·92	149·34	150·77	152·19	153·61	155·03

Conversion of Pounds per Square Inch to Kilogrammes per Square Centimetre.

Lbs. per Sq. In.	0	1	2	3	4	5	6	7	8	9
	Kilogrammes per Square Centimetre.									
0	0·0000	0·07031	0·14061	0·21092	0·28123	0·35154	0·42184	0 49215	0·56246	0·63276
10	0·70307	0·77338	0·84369	0·91399	0·98430	1·0546	1·1249	1·1952	1·2655	1·3358
20	1·4061	1·4764	1·5468	1·6171	1·6874	1·7577	1·8280	1·8983	1·9686	2·0389
30	2·1092	2·1795	2·2498	2·3201	2·3904	2·4607	2·5311	2·6014	2·6717	2·7420
40	2·8123	2·8826	2·9529	3·0232	3·0935	3·1638	3·2341	3 3044	3·3747	3·4450
50	3·5154	3·5857	3·6560	3·7263	3·7966	3·8669	3·9372	4·0075	4·0778	4·1481
60	4·2184	4·2887	4·3590	4·4293	4·4997	4·5700	4·6403	4·7106	4·7809	4·8512
70	4·9215	4·9918	5·0621	5·1324	5·2027	5·2730	5·3433	5·4136	5·4840	5·5543
80	5·6246	5·6949	5·7652	5 8355	5·9058	5·9761	6·0464	6·1167	6·1870	6·2573
90	6·3276	6·3979	6·4683	6·5386	6·6089	6·6792	6·7495	6·8198	6·8901	6·9604
100	7·0307	7·1010	7·1713	7·2416	7·3119	7·3822	7·4526	7·5229	7·5932	7·6635

1 kilogramme per sq. centimetre = 0·91 tons per sq. ft.
1 ton per sq. ft. = 1·09 kilogrammes per sq. centimetre.
1 ton per sq. ft. = 15·55 lbs. per sq. in.

CORRESPONDING HEIGHTS OF BAROMETER IN MILLIMETRES AND ENGLISH INCHES.

Millimetres.	English Inches.	Millimetres.	English Inches.	Millimetres.	English Inches.
720	28·347	739	29·095	758	29·843
721	28·386	740	29·134	759	29·882
722	28·425	741	29·174	760	29·922
723	28·465	742	29·213	761	29·961
724	28·504	743	29·252	762	30·000
725	28·543	744	29·292	763	30·039
726	28·583	745	29·331	764	30·079
727	28·622	746	29·370	765	30·118
728	28·662	747	29·410	766	30·158
729	28·701	748	29·449	767	30·197
730	28·740	749	29·488	768	30·236
731	28·780	750	29·528	769	30·276
732	28·819	751	29·567	770	30·315
733	28·858	752	29·606	771	30·355
734	28·898	753	29·645	772	30·394
735	28·937	754	29·685	773	30·433
736	28·976	755	29·724	774	30·473
737	29·016	756	29·764	775	30·512
738	29·055	757	29·803		

SECTION II

PART I

MECHANICS – STATICS – COEFFICIENT OF FRICTION – DYNAMICS – THE PENDULUM – THE CENTRIFUGAL GOVERNOR–IMPACT OF BODIES–WORK AND ENERGY RADIUS OF GYRATION–VECTORS–GEOMETRICAL AND INERTIAL PROPERTIES OF SQUARES, ETC. (pp. 17-33)

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

PART II

MEASUREMENT OF POWER (pp. 35–45)

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.

SECTION II

PART I

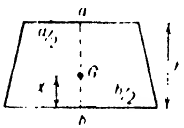
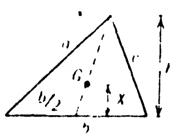
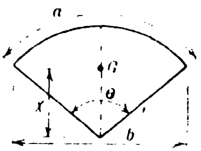
MECHANICS — STATICS — COEFFICIENT OF FRICTION — DYNAMICS — THE PENDULUM — THE CENTRIFUGAL GOVERNOR — IMPACT OF BODIES — WORK AND ENERGY RADIUS OF GYRATION — VECTORS — GEOMETRICAL AND INERTIAL PROPERTIES OF SQUARES, ETC.

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

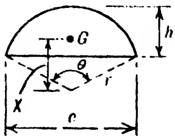
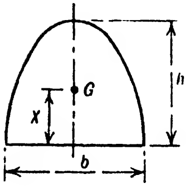
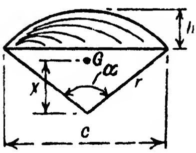
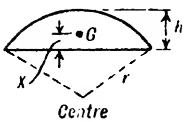
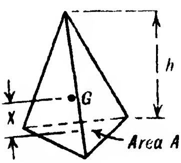
MECHANICS.

Definition.—Centre of Gravity: the point of application of the resultant gravitational force acting on a body; its position is independent of the attitude of the body.

Areas, Volumes and Positions of Centre of Gravity for Plane Figures and Solids.
(Centre of Gravity (C.G.) denoted by G.)

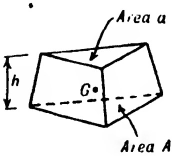
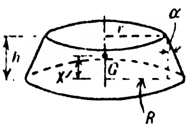
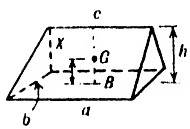
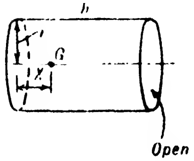
Figure.	Description.	Area or Volume.	Distance X.
	Trapezium	$\frac{h}{2}(a+b)$	$\frac{h}{3}\left(\frac{2a+b}{a+b}\right)$
	Triangle (Area)	$\frac{hb}{2}$	$\frac{h}{3}$
	Triangle (Perimeter)	—	$\frac{h}{2}\left(\frac{a+c}{a+b+c}\right)$
	Circular sector	$\frac{ar}{2}$ $\left[\frac{r^2\theta}{2}\right]$	$\frac{2}{3}\frac{br}{a}$ $\left[\frac{2}{3}r\frac{\sin\frac{\theta}{2}}{\frac{\theta}{2}}\right]$
$\angle\ \theta$ is in radians $\left(1 \text{ radian} = \frac{180^\circ}{\pi}\right)$	Circular arc	—	$\frac{br}{a}$

Areas, Volumes and Positions of Centre of Gravity for Plane Figures and Solids.
(Centre of Gravity denoted by G).—*cont.*

Figure.	Description.	Area or Volume.	Distance X.
 ∠ θ is in radians	Circular Segment	$\frac{r}{2}\times \text{arc} - \frac{c}{2}\times(r-h)$ $\left[\frac{r^2}{2}(\theta-\sin\theta)\right]$	$\frac{c^3}{12\times \text{area}}$ $\left[\frac{4r}{3}\cdot\frac{\sin^3\theta/2}{\theta-\sin\theta}\right]$
	Parabolic Segment	$\frac{2}{3}bh$	$\frac{2}{5}h$
	*Spherical Sector—* Surface Solid	$2\pi rh+\pi r\sqrt{r^2-(r-h)^2}$ $\frac{2}{3}\pi r^2h$ $h = r\left(1-\cos\frac{\alpha}{2}\right)$	$\frac{h^2+(r-h)\left(2h+\frac{c}{3}\right)}{2h+\frac{c}{2}}$ $\frac{3}{4}\left(r-\frac{h}{2}\right)$ $\left[\frac{3}{8}r\left(1+\cos\frac{\alpha}{2}\right)\right]$
	*Spherical Segment—* Convex surface Solid	$2\pi rh$ $\frac{1}{3}\pi h^2(3r-h)$	$\frac{h}{2}$ $\frac{h}{4}\left(\frac{4r-h}{3r-h}\right)$
 X = height of G above base	*Pyramid (or Cone)—* Slant surface Volume	$\frac{1}{2}$ slant height × perimeter of base $\frac{1}{3}Ah$	$\frac{h}{3}$ $\frac{h}{4}$ (G is on line joining apex to CG of base)

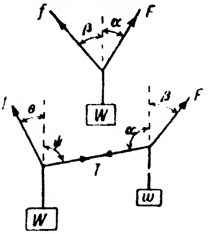
AREAS, VOLUMES AND POSITIONS OF CENTRE OF GRAVITY FOR PLANE FIGURES AND SOLIDS.

(Centre of Gravity denoted by G.)—*cont.*

Figure.	Description.	Area or Volume.	Distance X.
 X = Height of G above base	*Pyramidal Frustum*— Solid	$\frac{h}{3}(A + a + \sqrt{Aa})$	$\frac{h}{4}\left[\frac{A + 3a + 2\sqrt{Aa}}{A + a + \sqrt{Aa}}\right]$
	*Conic Frustum*— Convex Surface	$\pi(R+r)\sqrt{h^2 + (R-r)^2}$ $[\pi(R^2 - r^2)\operatorname{cosec}\alpha]$	$\frac{h}{2} - \frac{h}{6}\left[\frac{R-r}{R+r}\right]$
	Solid	$\frac{\pi h}{3}(R^2 + r^2 + Rr)$ $\left[\frac{\pi\cot\alpha}{3}(R^3 - r^3)\right]$	$\frac{h}{4}\left[\frac{R^2 + 3r^2 + 2Rr}{R^2 + r^2 + Rr}\right]$
 (Point B is centre of gravity of base)	*Wedge*— Solid	$\frac{(2a + c)bh}{6}$	$\frac{h}{2}\left(\frac{a + c}{2a + c}\right)$
	*Cylinder open at one end*— Surface	$\pi r(2h + r)$	$\frac{h^2}{2h + r}$

STATICS.

Weights suspended by inclined strings.



Case I

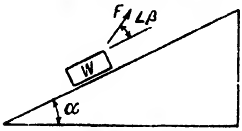
$$f = \frac{W \sin \alpha}{\sin(\alpha + \beta)}, \quad F = \frac{W \sin \beta}{\sin(\alpha + \beta)}.$$

$$W^2 = F^2 + f^2 + 2Ff \cos(\alpha + \beta)$$

Case II

$$T = \frac{W \sin \theta}{\sin(\theta + \psi)} = \frac{w \sin \beta}{\sin(\alpha + \beta)}$$

The Inclined Plane.



Coefficient of friction $= \mu$.
Limiting angle of friction $= \phi$.

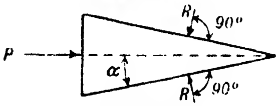
$$F = \frac{W(\sin \alpha + \mu \cos \alpha)}{\mu \sin \beta + \cos \beta} = \frac{W \sin(\phi + \alpha)}{\cos(\beta - \phi)},$$

$(\mu = \tan \phi)$.

If $\mu = 0$, $F = \dfrac{W \sin \alpha}{\cos \beta}$.

If $\beta = 0$ also, $F = W \sin \alpha$.

The Wedge.

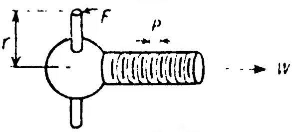


Coefficient of friction $= \mu$.

$$P = 2R(\mu \cos \alpha + \sin \alpha).$$

If $\mu = 0$, $P = 2R \sin \alpha$.

The Screw.

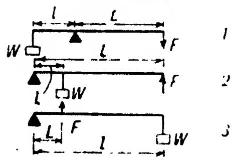


F = force applied to arm.

$$W = \frac{2\pi r F}{p + 2\pi \mu r_s},$$

where μ = coefficient of friction of screw thread, and r_s = radius of thread.

The Lever.

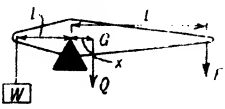


LIGHT LEVER.

In cases 1, 2 and 3,

$$F = \frac{Wl}{L}.$$

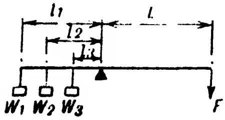
$$W = \frac{FL}{l}.$$



HEAVY LEVER.

Centre of gravity of lever at G.

$$F = \frac{Wl - Qx}{L}, \quad W = \frac{FL + Qx}{l}$$

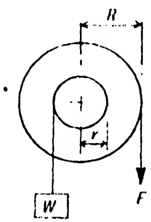


LEVER WITH SEVERAL WEIGHTS.

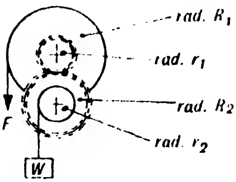
$$L = \frac{W_1 l_1 + W_2 l_2 + W_3 l_3 + \ldots}{F}$$

The Pulley.

(v_1 = velocity of point of application of force F, v_2 = velocity of weight W.)



Case I



Case II

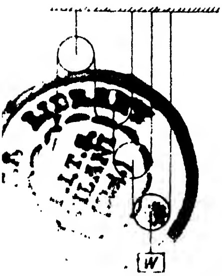
Case I.—2-diameter pulley.

$$F = \frac{Wr}{R}.\quad v_2 = \frac{v_1 r}{R}.$$

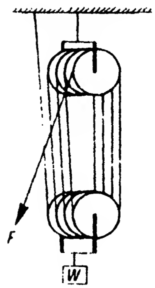
Case II.—2-diameter geared pulleys.

$$F = \frac{Wr_1r_2}{R_1R_2}.$$

$$v_2 = v_1\frac{r_1r_2}{R_1R_2}.$$



Case III



Case IV

Case III.—Compound single pulleys.

n = no. of movable pulleys.

$$F = \frac{W}{2^n}.\quad v_2 = \frac{v_1}{2^n}.$$

Case IV.—Movable multiple pulley.

n = no. of pulleys in movable block.

$$F = \frac{W}{2n}.\quad v_2 = \frac{v_1}{2n}.$$

Coefficient of Friction.

Definition.—The coefficient of static friction between two solid surfaces is the ratio of the tangential force required to produce relative sliding movement to the reaction between the surfaces

Owing to the variation in friction with the condition of the rubbing surfaces, and the contact pressure, it is impossible to specify accurate values without individual experiments. The following approximate values are those generally accepted:

Surfaces.	Coefficient of Friction.
Dry masonry on brickwork	0·6–0·7
Masonry on dry clay	0·5
Masonry on wet clay	0·3
Earth on earth	0·25–1·0
Timber on stone	0·4
Timber on timber	0·5–0·2
Timber on metals	0·6–0·2
Iron on stone	0·7–0·3
Metals on metals (dry)	0·25–0·15
Oiled metal surfaces	0·2–0·1

Once movement has begun, the coefficient of friction usually falls slightly. The following addition values are given for sliding friction:

Surfaces.	Coefficient of Friction.
Rubber on asphalt (wet)	0·25–0·75
" " " (dry)	0·5–0·8
Rubber on concrete (wet)	0·45–0·75
" " " (dry)	0·6–0·85
Brake lining on metal	0·3–0·7

These values fall with increase of rubbing speed. The value of the coefficient of friction may exceed unity.

DYNAMICS.

Definitions.

Mass is the amount of matter in a body.

Weight is the force exerted on the mass of a body by gravity.

Velocity is the distance travelled per unit time.

Acceleration is the increase of velocity per unit time.

Momentum is the product of mass and velocity.

Force is the product of mass and acceleration, thus force is the time-rate of increase or loss of momentum.

If the equation 'force = mass × acceleration' is to hold when force is expressed in pounds weight, mass must be expressed in *slugs*, or $\frac{\text{pounds mass.}}{g}$

Moment of Inertia or *Second Moment.*—If m_1, m_2, etc., are the masses of the particles composing the body, and r_1, r_2, etc., their perpendicular distances from the axis, the moment of inertia about the axis is

$$(m_1r_1^2 + m_2r_2^2 + \ldots .)$$

Radius of Gyration or *Swing Radius.*—The distance from the axis of rotation at which the whole mass of a body must be concentrated so that the moment of inertia will be unchanged.

Thus $$(\text{Radius of gyration})^2 = \frac{\text{moment of inertia}}{\text{mass}}.$$

Motion under Gravity.

Type of Motion.	Velocity (v).	v^2 (at distance s)	Distance Travelled (s).
Motion under gravity from rest	gt	$2gs$	$\frac{1}{2}gt^2$
Motion under gravity, with initial *downward* velocity V.	$V + gt$	$V^2 + 2gs$	$Vt + \frac{1}{2}gt^2$
Motion under gravity, with initial *upward* velocity V.	$V - gt$	$V^2 - 2gs$	$Vt - \frac{1}{2}gt^2$

g = acceleration due to gravity (= 32·2 feet per second per second).
t = time from beginning of motion.

THE PENDULUM.

(1) Simple Pendulum.

Definition.—A weight concentrated at one end of a weightless string, the other end of which is fixed.

If l = length of string in ft.

g = acceleration due to gravity (ft. per sec. per sec.).

Time for a complete oscillation (to and fro)

$$= 2\pi\sqrt{\frac{l}{g}} \text{ sec.}$$

$$= \text{period of oscillation} = \frac{1}{\text{frequency of oscillation}}$$

If L is the latitude of the point of observation, the length of the simple pendulum with a period of 1 sec. (in ins.) is

$$= 39{\cdot}0265 + 0{\cdot}1608 \sin^2 \text{L}.$$

Values of g and l in Various Localities.

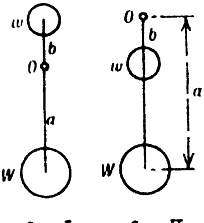
g in cm. per sec. per sec.; l in cm.

Place.	Latitude.	g.	l.
Equator	0	978·1	99·103
Greenwich	51° 29′	981·17*	99·413†
Paris	48° 50′	980·94	99·39
Berlin	52° 30′	981·25	99·422
Dublin	53° 21′	981·32	99·429
Glasgow	55° 52′	981·44	99·442
Edinburgh	55° 57′	981·54	99·451
North Pole	90°	983·11	99·61

* Equals 32·1902 ft. per sec. † Equals 3·2615 ft.

(2) Compound Pendulum.

Definition.—A rigid body suspended at a horizontal axis under the influence of gravity.



Case I *Case II*

If O is the centre of suspension, length of equivalent simple pendulum

$$= \frac{Wa^2 + wb^2}{Wa - wb}. \quad \text{(Case I.)}$$

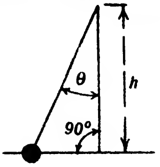
$$= \frac{Wa^2 + wb^2}{Wa + wb}. \quad \text{(Case II.)}$$

$$= \frac{\text{moment of inertia about centre of suspension}}{\text{mass} \times \text{distance of C.G. from centre of suspension}}.$$

The *Centre of Oscillation* is the point at which if all the mass were concentrated, the period of oscillation would be unchanged. Thus the length of the equivalent simple pendulum is the distance between the centres of suspension and oscillation, which are interchangeable.

(3) Conical Pendulum.

Definition.—A heavy particle suspended from a fixed point and made to move in a horizontal circle at constant speed.



If h = height in feet,
and g = gravity in feet per sec. per sec.

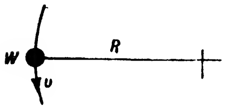
Period of revolution $= 2\pi \sqrt{\frac{h}{g}}$ sec.

i.e. period is independent of length of suspending rod or string.

CENTRIFUGAL FORCE.

Definition.—The radial force acting on a body travelling on a curve, due to the tendency of the body to continue travelling in a straight line.

Centrifugal Force acting on a Concentrated Mass.

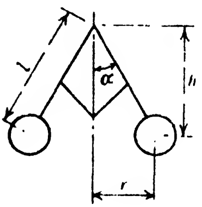


Velocity on curve $= v$; radius of curvature $=$ R.

Centrifugal force $= \mathrm{F} = \frac{\mathrm{W}v^2}{g\mathrm{R}}$ lb.

If n = revolutions per minute,

$$\mathrm{F} = \frac{\mathrm{WR}n^2}{2933} \text{ lb}$$



The Centrifugal Governor.

$$\text{R.P.M.} = n = \frac{60}{2\pi}\sqrt{\frac{g}{h}} = \frac{54{\cdot}16}{\sqrt{l \cos \alpha}}.$$

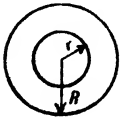
$$h = \frac{2933}{n^2}.$$

Centrifugal Tension.

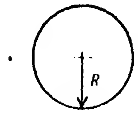
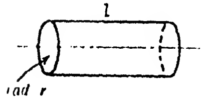
n = R.P.M.

Circumferential tension in a hoop $= mv^2 = m\omega^2 r^2$,

where m is the mass of unit length; v its linear velocity;
ω its angular velocity (same for all particles);
r its radius.

Body.		Centrifugal Tension (in Lb.). W = mass in lb.
	Ring about centre.	$\frac{\mathrm{W}n^2 \ (\mathrm{R}^2 + r^2)}{4150}$

CENTRIFUGAL TENSION—*cont.*

Body.		Centrifugal Tension (in Lb.). W = mass in lb.
	Disc about centre.	$\frac{Wn^2R}{4150}$
	Cylinder about diameter of one end.	$\frac{Wn^2\sqrt{4l^2+3r^2}}{10260}$

IMPACT OF BODIES.

Mass of the two bodies = M and m.

Velocities before and after impact = V and v, V′ and v' respectively.

1. *Inelastic Bodies.*—(v' = V′, no separation after impact).

$$v' = \frac{MV + mv}{M + m}.$$

2. *Elastic Bodies.*—(K = coefficient of restitution between the bodies, i.e. $v' - V' = K(V - v)$)

$$V' = \frac{V(M - Km) + vm(1 + K)}{M + m},$$

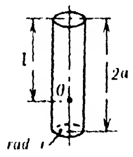
$$v' = \frac{v(m - KM) + VM(1 + K)}{M + m}.$$

(N.B.—The velocities must be taken as positive in one direction and negative in the opposite direction. The above formulæ cover all cases if this rule is observed.)

CENTRE OF PERCUSSION.

Definition.—When a body receives a blow on a line passing through the centre of percussion, the resulting motion is purely one of translation and not rotation.

Cylinder suspended at one end.



Centre of percussion is at O, where

$$l = \frac{4a}{3} + \frac{r^2}{4a}.$$

If r is small, $l = \frac{4a}{3}$.

O is also the centre of oscillation (see p. 23).

WORK AND ENERGY.

Definitions.—1. The *work* done by a force is equal to the product of the force and the distance travelled by its point of application in the direction of the force.

2. (*a*) *Potential energy* is the energy possessed by a body in virtue of its position relative to some zero position.

(*b*) *Kinetic energy* is the energy possessed in virtue of its motion (either rotatory or translatory). Potential energy of a body of mass W lb. at height h feet above the zero position = Wh ft.-lb.

Translation.—If linear velocity is V ft. per sec., kinetic energy of body = $\frac{\frac{1}{2}WV^2}{g}$ ft.-lb.

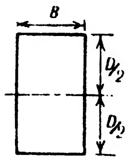
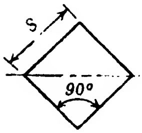
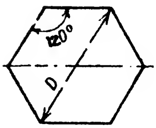
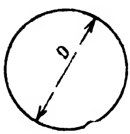
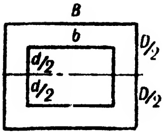
Rotation.—If angular velocity is ω radians per sec., kinetic energy of body = $\frac{\frac{1}{2}I\omega^2}{g}$ ft.-lb.

(I = moment of inertia in lb. × ft.²)

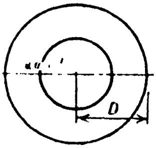
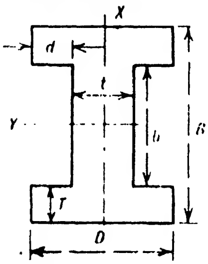
RADIUS OF GYRATION AND MOMENT OF INERTIA FOR PLANE SECTIONS AND SOLID BODIES.

Moment of Inertia of plane section = area × (radius of gyration)².
Moment of Inertia of solid body = mass × (radius of gyration)².

PLANE SECTIONS.

Figure.		Radius of Gyration.	Moment of Inertia.
	Axis parallel to side B	$0{\cdot}289D$	$\frac{1}{12}BD^3$
	Axis perpendicular to plane of figure through centre	$0{\cdot}289\sqrt{B^2+D^2}$	$\frac{1}{12}(BD^3+DB^3)$
	Axis a diagonal	$0{\cdot}289S$	$\frac{1}{12}S^4$
	Axis a diagonal	$0{\cdot}228D$	$0{\cdot}0338D^4$
	Axis perpendicular to plane of figure	$0{\cdot}323D$	$0{\cdot}0676D^4$
	Axis a diameter.	$0{\cdot}25D$	$\frac{\pi D^4}{64} = 0{\cdot}0491D^4$
	Axis perpendicular to plane of figure	$0{\cdot}354D$	$\frac{\pi D^4}{32} = 0{\cdot}0982D^4$
	Axis	$0{\cdot}289\sqrt{\frac{BD^3-bd^3}{BD-bd}}$	$\frac{1}{12}(BD^3-bd^3)$

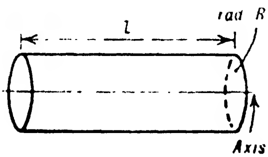
PLANE SECTIONS—*cont.*

Figure.		Radius of Gyration.	Moment of Inertia.
	Axis a diameter	$0.25\sqrt{D^2+d^2}$	$\frac{\pi}{64}(D^4-d^4)=0.0491(D^4-d^4)$
	Axis perpendicular to plane of figure	$0.354\sqrt{D^2+d^2}$	$\frac{\pi}{32}(D^4-d^4)=0.0982(D^4-d^4)$
Narrow ring (Mass = M) **Axis a diameter.**		$0.354\,D$	$\frac{MD^2}{8}$
		*Axis X.* $0.289\sqrt{\frac{2TD^3+bt^3}{2TD+bt}}$	$\frac{1}{12}(2TD^3+bt^3)$
		Axis Y. $0.289\sqrt{\frac{DB^3-2db^3}{DB-2db}}$	$\frac{1}{12}(DB^3-2db^3)$

General formula : For plane figures, the sum of the M.I. about two perpendicular axes through the centre of gravity, in the plane of the figure, is equal to the M.I. about an axis through the C.G., perpendicular to this plane (the polar M.I.).

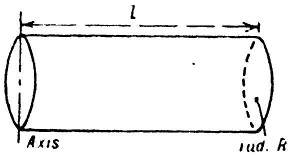
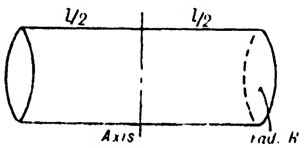
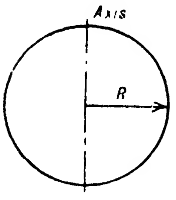
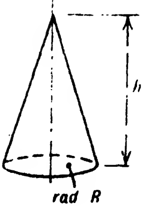
SOLID BODIES.

(Specific gravity of material = S.)

Body.	Radius of Gyration.	Moment of Inertia
	$\frac{R}{1.414}$	$\frac{\pi l R^4 S}{2}$
Hollow cylinder, inner radius = r	$\sqrt{\frac{R^2+r^2}{2}}$	$\frac{\pi l(R^4-r^4)S}{2}$

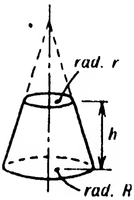
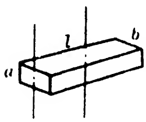
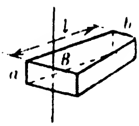
SOLID BODIES—*cont.*

(Specific gravity of material = S.)

Body.		Radius of Gyration.	Moment of Inertia.
		$\sqrt{\frac{4l^2+3R^2}{12}}$	$\frac{\pi R^2 l(4l^2+3R^2)}{12}.S$
When R is small (thin rod)		$0{\cdot}5775\,l$	$\frac{\pi R^2 l^3}{3}.S$
		$\sqrt{\frac{l^2+3R^2}{12}}$	$\frac{\pi R^2 l(l^2+3R^2)}{12}.S$
When R is small (thin rod):		$0{\cdot}2887\,l$	$\frac{\pi R^2 l^3}{12}.S$
	Spherical shell	$0{\cdot}8165\,R$	Mass $\times \frac{2R^2}{3}$
	Solid sphere	$0{\cdot}6324\,R$	$\frac{8\pi R^5}{15}.S$
Thick spherical shell Inner radius = r		$0{\cdot}6324\sqrt{\frac{R^5-r^5}{R^3-r^3}}$	$\frac{8}{15}\pi(R^5-r^5).S$
	About axis of cone	$0{\cdot}548\,R$	$\frac{\pi R^4 h}{10}.S$
	About a diameter of base	$\sqrt{\frac{2h^2+3R^2}{20}}$	$\frac{\pi R^2 h(2h^2+3R^2)}{60}.S$
	About an axis through vertex parallel to base	$\sqrt{\frac{12h^2+3R^2}{20}}$	$\frac{\pi R^2 h(12h^2+3R^2)}{60}.S$

SOLID BODIES—*cont.*

(Specific gravity of material = S.)

Body.		Radius of Gyration.	Moment of Inertia.
	About axis of conic frustum	$0\cdot348\sqrt{\frac{R^5-r^5}{R^3-r^3}}$	$\frac{\pi h}{10}\left(\frac{R^5-r^5}{R-r}\right).S$
	About a diameter of base	$k=\sqrt{\frac{h^2}{10}\left(\frac{R^2+3Rr+6r^2}{R^2+Rr+r^2}\right)+\frac{3}{20}\left(\frac{R^5-r^5}{R^3-r^3}\right)}$	$\frac{\pi h}{3}(R^2+r^2+Rr)\times k^2.S$
	Perpendicular axis through centre	$\sqrt{\frac{l^2+b^2}{12}}$	$abl\left(\frac{l^2+b^2}{12}\right).S$
	Perpendicular axis in plane of one end	$\sqrt{\frac{4l^2+b^2}{12}}$	$abl\left(\frac{4l^2+b^2}{12}\right).S$
	Perpendicular axis in plane of large end	$0\cdot204\sqrt{12l^2+B^2+b^2}$	$\frac{al(B+b)(12l^2+B^2+b^2)}{48}.S$

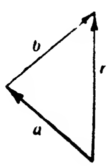
Rotation about an Axis at a Distance.

Moment of inertia = moment of inertia about a parallel axis through the centre of gravity + Mh^2, where M = mass and h = perpendicular distance from C.G. to axis of rotation.

VECTORS.

(*By J. D. W. Ball, A.M.I.C.E.*)

Vector quantities, as distinct from scalar quantities, have direction as well as magnitude, and are represented by straight lines of given length, in given directions, with arrow heads to denote their sense along the respective directions.



ADDITION OF VECTORS.

$$a + b = r$$

SUBTRACTION OF VECTORS.

A vector is said to be subtracted when it is added with its direction reversed.

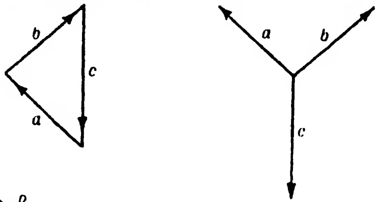
$$\text{If } c = -r$$

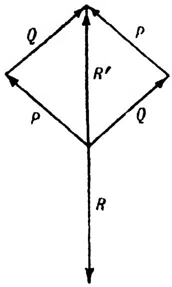
$$a + b - r = 0$$

$$a + b + c = 0$$

Triangle of Forces.

If three forces meeting at a point are in equilibrium, the forces can be represented in magnitude and direction by the sides of a triangle, the sides of which are parallel to the respective forces.



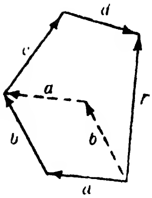


Parallelogram of Forces.

The vectors $P + Q + R = 0$

and $P + Q = - R = R'$

where R' is the resultant of P and Q. R' is the diagonal of the parallelogram of which P and Q are adjacent sides.



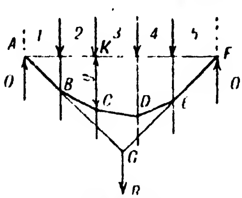
Polygon of Forces.

$$a + b + c + d = r$$

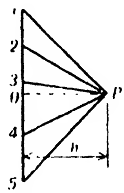
If $e = - r$

$$a + b + c + d + e = 0.$$

If a number of forces, acting at a point, be in equilibrium, the sum of their vectors = 0, or they may be represented in magnitude and direction by the sides of a closed polygon. It is immaterial in which order the vectors are added.



Link Polygon.

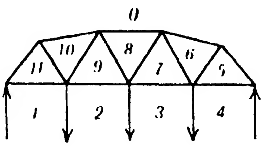


Vector Polygon.

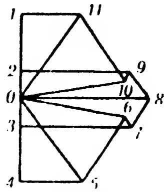
1, 2 ; 2, 3 ; 3, 4 and 4, 5 are a number of parallel forces, such as concentrated loads carried by a beam, the supports of which are 0, 1 and 5, 0.

The forces are represented to a convenient scale in the Vector Polygon, and a pole P selected at a convenient distance from the line of loads, representing, say 10 tons, to the same scale as that used in plotting the vertical loads. Join P1 and in the Link Polygon draw AB parallel to it. Join P2 and across the space marked 2 in the Link Polygon, draw BC parallel to it, and proceed in a similar way with P1, P4, and P5. Join AF and draw P0 parallel to it in the Vector

Polygon. Then 0, 1 and 5, 0 are the reactions at the supports. Produce AB and FE to meet at G, which marks the position of the resultant of the four forces considered. If GK measures *y* feet and the polar distance represents *h* tons, the bending moment at K is *hy* foot-tons.



Truss.



Force Diagram.

The above Force Diagram determines the forces in the members of the truss shown, due to loads applied at the lower chord. Set out the loads as in the previous example, and determine the reactions as there indicated, or by taking moments about one of the bearings. Draw 0, 11 parallel to the corresponding member of the truss, and 1, 11 parallel to the member thus designated in the lower chord. Then 0, 1, 11 is a triangle representing the three forces acting at the left abutment, i.e. the upward reaction 0, 1, the tension in the lower chord 1, 11, and the compression in the raker 11, 0. Draw 11, 10 parallel to the member marked 11, 10 and 0, 10 parallel to the upper chord member 0, 10. Then 0, 11, 10 is a triangle representing the three forces acting at the top of the raker. Draw 10, 9, parallel to the corresponding member, to meet 2, 9, drawn parallel to the lower chord. Then 9, 10, 11, 1, 2, is a polygon, the sides of which represent the four forces acting at the panel point next the abutment on the lower chord. By proceeding in a similar manner the diagram is completed as shown.

The Geometrical and Inertial Properties of Squares, Octagons, Hexagons and Circles.

(By C. L. T. Griffith, late Professor of Civil Engineering, Madras.)

The following tables contain a comprehensive set of factors relating to squares, octagons, hexagons and circles. Octagons and hexagons occur in some cases of engineering practice in steel and wooden shafts, concrete and brick chimneys, and in foundations.

The size of a square, an octagon or of a hexagon may be determined or described in four different ways, as :—

(1) The width across flats, F.
(2) The width across corners, C.
(3) The length of the side or edge, E.
(4) The area of the figure, A.

Table I gives each of these measurements in terms of the other three.

Table II gives the Moments of Inertia about an axis passing through the centre of the figure in terms of F, C, E and A; to which has been added the Moment of Inertia of a circle in terms of its diameter, D, and its area, A. For all these four figures the Moment of Inertia is invariable for any axis passing through the centre, in the plane of the figure.

Table III gives the Radii of Gyration, ρ, and their squares, ρ^2, in terms of the four systems of measurement. Here also the ρ's are invariable, or, in other words, the ellipse of gyration is a circle.

Table IV shows the 'neutral axis distance,' *n* (sometimes written *y*), and the same thing as 'the distance of the extreme fibre,' and is the maximum distance of the boundary of the figure measured perpendicularly from the neutral axis. These dimensions depend on the direction of the neutral axis, and are given for two cases, first, when the neutral axis is parallel to two of the sides of the figure, and second, when the neutral axis is perpendicular to the line joining two corners.

Table V gives certain 'core' or 'kern' distances, *q*. The core of a section is that small area in the centre of the section, hatched in the illustrations, within which the centre of pressure must pass to comply with the condition that no tension exists in any part of the cross section. As regards squares and rectangles this is often loosely referred to as 'the middle third'; but it is not always realised that the middle third is a diamond, and that the 'third' (of the side) is measured across corners. In the case of squares, a deviation of the centre of pressure towards the corner of the square must be 29 per cent. less than the permissible deviation towards the middle of a side if the condition of no tension is to be complied with.

Table VI gives factors for finding the dimensions of octagons and hexagons the areas of which equal the area of a circle of diameter D.

Hollow Squares, Octagons, Hexagons and Circles.

For these hollow figures the areas and moments of inertia are the *differences* between the I's and A's of the external and internal figures, the wall thicknesses being uniform all round.

The *squares* of the radii of gyration are the *sum* of the g^2's of the outer and inner figures. Hence, the radius of gyration of a hollow octagon

$$= 0\cdot257\sqrt{F^2 + f^2},$$

where F is the external, and f the internal, width across flats.

The neutral axis distances are, of course, those of the external figure.

The core distances for hollow figures, like the radii of gyration, are larger than those of solid figures of the same outside size, and can be obtained from the fact that the core distance equals g^2 divided by n, the neutral axis distance. Or we may use the factors given in Table V, and replace F, C, E, D and A by

$$\frac{F^2 + f^2}{F}, \frac{C^2 + c^2}{C}, \frac{E^2 + e^2}{E}, \frac{D^2 + d^2}{D} \text{ and } \frac{A + a}{\sqrt{A}} \text{ respectively}$$

Table I.—The Ratios of F, C, E and A to Each Other.

Squares.	Octagons.	Hexagons.
F = 0·7071C	F = 0·9239C	F = 0·8660C
= E	= 2·4142E	= 1·7321E
= $\sqrt{A}$	= 1·0987$\sqrt{A}$	= 1·0746$\sqrt{A}$
C = 1·4142F	C = 1·0824F	C = 1·1547F
= 1·4142E	= 2·6131E	= 2E
= 0·7071$\sqrt{A}$	= 1·1892$\sqrt{A}$	= 1·2408$\sqrt{A}$
E = F	E = 0·4142F	E = 0·5774F
= 0·7071C	= 0·3827C	= 0·5C
= $\sqrt{A}$	= 0·4551$\sqrt{A}$	= 0·6204$\sqrt{A}$
A = F²	A = 0·8284F²	A = 0·8660F²
= 0·5C²	= 0·7071C²	= 0·6525C²
= E²	= 4·8284E²	= 2·5981E²

Table II.—Moments of Inertia, I, about any Axis through the Centre in the Plane of the Figure.

Squares.	Octagons.	Hexagons.	Circles.
I = 0·0833F⁴	I = 0·0547F⁴	I = 0·0601F⁴	I = 0·0491D⁴
= 0·0208C⁴	= 0·0399C⁴	= 0·0338C⁴	= 0·0796A²
= 0·0833E⁴	= 1·8595E⁴	= 0·5413E⁴	
= 0·0833A²	= 0·0798A²	= 0·0802A²	

Table III.—Radii of Gyration, g, and their Squares, g^2, about any Axis through the Centre in the Plane of the Figure.

Squares.	Octagons.	Hexagons.	Circles.
g = 0·2887F	g = 0·2571F	g = 0·2635F	g = 0·25D
= 0·2041C	= 0·2375C	= 0·2282C	= 0·2821$\sqrt{A}$
= 0·2887E	= 0·6206E	= 0·4564E	
= 0·2887$\sqrt{A}$	= 0·2824$\sqrt{A}$	= 0·2832$\sqrt{A}$	
g^2 = 0·0833F²	g^2 = 0·0661F²	g^2 = 0·0694F²	g^2 = 0·0625D²
= 0·0417C²	= 0·0564C²	= 0·0521C²	= 0·0796A
= 0·0833E²	= 0·3851E²	= 0·2083E²	
= 0·0833A	= 0·0798A	= 0·0802A	

Table IV.—Neutral Axis Distance, n, or Distance to Extreme Fibre.

Case A.—Neutral Axis is parallel to two sides of the figure as in figs. IA, IIA and IIIA.

Squares.	Octagons.	Hexagons.
$n = 0·5F$	$n = 0·5F$	$n = 0·5F$
$= 0·3536C$	$= 0·4619C$	$= 0·4330C$
$= 0·5E$	$= 1·2071E$	$= 0·8660E$
$= 0·5\sqrt{A}$	$= 0·5493\sqrt{A}$	$= 0·5373\sqrt{A}$

Case B.—Neutral Axis is perpendicular to line joining two opposite corners as in figs. IB, IIB, IIIB.

Squares.	Octagons.	Hexagons.
$n = 0·7071F$	$n = 0·5412F$	$n = 0·5774F$
$= 0·5C$	$= 0·5C$	$= 0·5C$
$= 0·7071E$	$= 1·3066E$	$= 1·0E$
$= 0·7071\sqrt{A}$	$= 0·5946\sqrt{A}$	$= 0·6204\sqrt{A}$

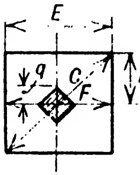


FIG. IA.

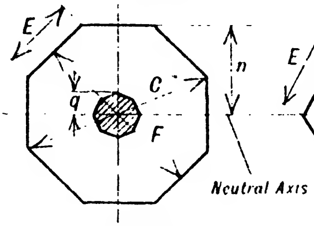


FIG. IIA.

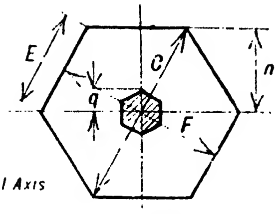


FIG. IIIA.

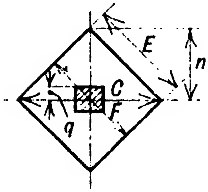


FIG. IB.

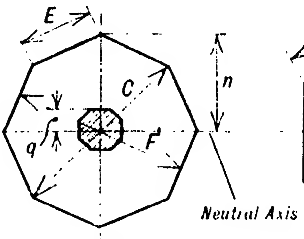


FIG IIB.

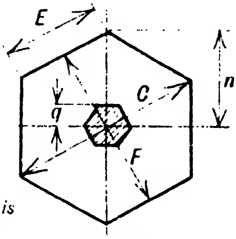


FIG. IIIB.

Table V.—Core Distances, q.

Case A.—Centre to corners of core, figures set as in figs. IA, IIA, IIIA.

Squares.	Octagons.	Hexagons.
$= 1·1667F$	$q = 0·1321F$	$q = 0·1389F$
$= 0·1179C$	$= 0·1221C$	$= 0·1203C$
$= 0·1667E$	$= 0·3190E$	$= 0·2406E$
$= 0·1667\sqrt{A}$	$= 0·1453\sqrt{A}$	$= 0·1492\sqrt{A}$

Case B.—Centre to side of core, figures set as in figs. IB, IIB, IIIB.

$q = 0·1179F$	$q = 0·1221F$	$q = 0·1203F$
$= 0·0833C$	$= 0·1128C$	$= 0·1042C$
$= 0·1179E$	$= 0·2948E$	$= 0·2083E$
$= 0·1179\sqrt{A}$	$= 0·1341\sqrt{A}$	$= 0·1293\sqrt{A}$

Table VI.—Factors for Dimensions of Octagons and Hexagons having Areas equal to the Area of a Circle of Diameter D.

Octagons.	Hexagons.
$F = 0·9737D$	$F = 0·9523D$
$C = 1·0539D$	$C = 1·0996D$
$E = 0·4033D$	$E = 0·5498D$

SECTION II

PART II

THE MEASUREMENT OF POWER.

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

1. ANIMAL POWER.

WORK DONE BY MEN AND ANIMALS.

Nature of Labour.	Daily Duration of Work in Hours.	No. of Units of Work per Day.	No. of Units of Work per Minute.	Weight raised or Mean Pressure.	Velocity. Feet per Min.	Velocity. Miles per Hour.
(1) *Raising weights vertically.*						
A man mounting a gentle incline or ladder without burden, i.e. raising his own weight . .	8·0	2,032,000	4,230	145	29	0·33
Labourer raising weights with rope and pulley, the rope returning without load . .	6·0	563,000	1,560	40	39	0·44
Labourer lifting weights by hand	6·0	531,000	1,480	44	34	0·38
Labourer carrying weights on his back up a gentle incline or up a ladder and returning unladen	6·0	406,000	1,130	145	8	0·09
Labourer wheeling materials in a barrow up an incline of 1 in 12 and returning with the empty barrow	10·0	313,000	520	130	4	0·045
Labourer lifting earth with a spade to a mean height of 5½ ft.	10·0	281,000	470	6	78	0·9
(2) *Action on Machines.*						
Labourer walking and pushing or pulling horizontally . .	8·0	1,500,000	3,130	27	116	1·32
Labourer turning a winch . .	8·0	1,250,000	2,600	18	144	1·64
Labourer pulling and pushing alternately in a vertical direction	8·0	1,146,000	2,390	11	216	2·70
Horse yoked to a cart and walking	10·0	15,688,000	26,150	150	175	2·00
Do. to a whim gin . . .	8·0	8,440,000	17,600	100	175	2·00
Do. do. trotting . . .	4·5	7,036,000	26,060	66⅔	391	4·44
Ox yoked to a whim gin and walking	8·0	8,127,000	16,930	145	117	1·33
Mule do. do.	8·0	5,627,000	11,720	66⅔	176	2·00
Ass do. do.	8·0	2,417,000	5,030	30	168	1·95

The following table gives the useful effect of men and animals employed in the horizontal transport of burdens. The second and third columns give the useful effect, viz. the product of the weight in lbs. and the distance in feet. This must not be mistaken for the units of work done by the agent, the agent being employed *not* in raising the weight, but in overcoming the passive resistances, friction, etc., which depend on the weight indeed, but are only a *fraction* of it.

USEFUL EFFECT OF AGENTS EMPLOYED IN THE HORIZONTAL TRANSPORT OF BURDENS.

Agent.	Daily Duration of Work in Hours.	Useful Effect Daily.	Useful Effect per Minute.	Weight transported.*	Velocity. Feet per Min.	Velocity. Miles. per Hr.
Man walking on a horizontal road without burden, i.e. transporting his own weight . .	10·0	25,398,000	42,330	145	292	3·32
Labourer transporting materials in a truck on two wheels, returning with it empty for a new load	10·0	13,025,000	21,710	220	99	1·12
Do. in a wheelbarrow . .	10·0	7,815,000	13,030	130	160	1·14
Labourer walking with a weight on his back	7·0	5,470,000	13,030	90	145	1·64
Labourer transporting materials on his back and returning unburdened for a new load .	6·0	5,087,000	14,110	145	97	1·10
Do. on a handbarrow . .	10·0	4,298,000	7,160	110	65	0·74
Horse transporting materials in a cart, walking, always laden	10·0	200,582,000	334,300	1,500	223	2·53
Do. trotting	4·5	90,262,000	334,300	750	44	5·06
Do. transporting materials in a cart returning with the cart empty for a new load . .	10·0	109,408,000	182,350	1,500	121	1·38
Horse walking with a weight on his back	10·0	34,385,000	57,310	270	212	2·41
Do. trotting	7·0	32,092,000	76,410	180	424	4·82

* Exclusive of the weight of the barrow, truck, cart, etc.

In advocating the employment of mechanical tractors on the farm, *Modern Farming* states that the horse eats annually the product of five acres. According to Edison, the efficiency of the horse is about 2 per cent.

DATA ON MAN-POWER.

The mean effect of the power of a man, unaided by a machine, working to the best practicable advantage, is the raising of 70 lbs. 1 foot in a second, for 10 hours in a day.

Two men, working at a windlass with the handles at right angles to each other, can raise 70 lbs. more easily than one man can 30 lbs.

A man of ordinary strength exerts a force of 30 lbs., for 10 hours in a day, with a velocity o 2½ feet per second = 4,500 lbs. raised 1 foot in a minute = 0·2 of the work of a horse.

A man can travel, without a load, on level ground, during 8½ hours a day, at the rate of 3·7 miles an hour, or 31½ miles a day. He can carry 111 lbs. 11 miles in a day.

A porter going short distances, and returning unloaded, can carry 135 lbs. 7 miles a day. He can transport, in a wheelbarrow, 150 lbs. 10 miles in a day.

WEIGHT OF CROWDS.

Experiments carried out by Professor L. J. Johnson, of Harvard University, show that it is possible for this to reach the high figure of 181·3 lbs. per square foot of floor surface. If the crowd jumps, the apparent load may be 50 per cent. more than the steady load, and running four abreast will show a similar result. A weight of 140 lbs. per square foot is quite possible where there are throngs of people; while a load of 80 lbs. per square foot is quite common in buildings and private houses where social gatherings are frequent, and a maximum of 40 lbs. or 45 lbs. is far short of what actually does prevail.

Some experiments carried out in America show that a crowd of men pushing against a balustrade may exert a horizontal pressure of 168 lbs. per foot run when they are three deep.*

COSTS AND CAPACITIES OF VARIOUS METHODS OF TRANSPORT.

Transport Unit.	Load.	Total Daily Mileage Capacity.	Total Daily Mileage Laden.	Capacity in Ton-Miles a Day.	Charges per Ton-Mile.	Gross Daily Takings.
Porters	60 lb.	15	Assumed always laden	0·4	5*s*. to 1*s*. 6*d*.	2*s*. to 7*d*.
Pack animals	200 to 400 lb.	24	12 to 18	1 to 3	5*s*. to 2*s*.	5*s*. to 6*s*.
Animal-drawn vehicles	1 to 5 tons	16	8 to 12	8 to 60	1*s*. 6*d*. to 9*d*.	12*s*. to 2*l*. 5*s*.
Mechanical pneumatic-tyred wheeled vehicles	½ to 1 ton	80	40 to 60	20 to 60	2*s*. 6*d*. to 1*s*. 6*d*.	2*l*. 10*s*. to 4*l*. 10*s*.
Roadless vehicles	1 to 2 tons	70	35 to 50	35 to 100	1*s*. 6*d*. to 1*s*.	2*l*. 12*s*. 6*d*. to 5*l*.
Roadless vehicles	2 to 5 tons	60	30 to 45	60 to 225	1*s*. to 8*d*.	3*l*. to 7*l*. 10*s*.
Roadless vehicles	5 to 20 tons	60 to 40	30	150 to 600	8*d*. to 4*d*.	5*l*. to 10*l*.
Roadless vehicles	20 to 200 tons	40 to 32	20 to 24	400 to 4,800	4*d*. to 2*d*.	6*l*. 13*s*. 4*d*. to 40*l*.
Arterial railway goods trains	200 to 500 tons	150 to 300	75 to 225	15,000 to 112,500	2*d*. to 1*d*.	125*l*. to 468*l*. 15*s*.

NOTE.—The column of total daily mileage is based on no-return loads as a minimum up to half-return loads as a maximum.

(*Philip Johnson; paper read at a Joint Meeting of Section G (Engineering) and Section F (Economics) at the Southampton Meeting of the British Association, August* 31, 1925.)

DATA ON HORSE-POWER.

A horse can travel 400 yards, at a walk, in 4½ minutes; at a trot, in 2 minutes; and at a gallop, in 1 minute. A draught horse can draw 1,600 lbs. 23 miles a day, weight of carriage included. The ordinary work of a horse may be stated at 32,500 lbs., raised 1 foot in a minute, for 8 hours a day. In a horse-mill a horse moves at the rate of 3 feet in a second; the diameter of the track should not be less than 25 feet. The strength of a horse is equivalent to that of 5 men.

TABLE OF THE AMOUNT OF LABOUR A HORSE OF AVERAGE STRENGTH IS CAPABLE OF PERFORMING, AT DIFFERENT VELOCITIES, ON CANALS, RAILROADS, AND ROADS.

Velocity per Hour.	Duration of Work.	Useful Effect for 1 Day, drawn 1 Mile.			Velocity per Hour.	Duration of Work.	Useful Effect for 1 Day, drawn 1 Mile.		
		On a Canal.	On a Railroad.	On a Road.			On a Canal.	On a Railroad.	Road.
Miles.	Hours.	Tons.	Tons.	Tons.	Miles.	Hours.	Tons.	Tons.	Tons.
2½	11·5	520	115	14·0	6	2·000	30·0	48·0	6·0
3	8·0	243	92	12·0	7	1·500	19·0	41·0	5·1
4	4·5	102	72	9·0	8	1·125	12·8	36·0	4·5
5	2·9	52	57	7·2	10	0·750	6·6	28·8	3·6

A horse in a mill can produce an effect of 106 lbs., at a velocity of 3 feet in a second, for 8 hours in a day. A mule can produce, under a like velocity and time, an effect of 71 lbs.; and an ass, 37 lbs.

An ox, walking at a velocity of 2 feet in a second (1·34 miles per hour), will exert a pull of 154 lbs. for 8 hours in a day.

The draught of man and animals by traces is as follows:

Man, 150 lbs. Horse, 600 lbs. Mule, 500 lbs. Ass, 360 lbs.

* *Engineering News-Record*, Vol. 94, No. 5 (1925), p. 200.

A man rowing a boat 1 mile in 7 minutes performs the labour, while rowing, of 6 fully-worked labourers at ordinary occupations of 10 hours.

Labour upon Embankments.

Single Horse and Cart.—A horse with a loaded dirt-cart, employed in excavation and embankment, will make 100 lineal feet of trip, or 200 feet in distance per minute, while moving. The time lost in loading, dumping, awaiting, etc. = 4 minutes per load.

A medium labourer will load with a cart in 10 hours, the following earths, measured in the bank:

Gravelly Earth, 10; *Loam*, 12; *Sandy Earth*, 14 cubic yards.

Carts are loaded as follows; *Descending Hauling*, ⅓ of a cubic yard in bank; *Level Hauling*, ¼ of a cubic yard in bank; *Ascending Hauling*, ⅕ of a cubic yard in bank.

Loosening, etc.—In *Loam*, a three-horse plough will loosen from 250 to 800 cubic yards per day of 10 hours.

Power Required for Ploughing.

$$\text{Horse-power} = \frac{rwdv}{375}$$

where,

r = soil resistance in lbs. per sq. in.; w = total width in ins. of all furrows ploughed at once d = depth of furrows in ins.; v = speed of travel in miles per hour.

The quantity rwd is the draught of the plough, and at steady speeds it equals the pull exerted at the draw-bar of the tractor of the haulage rope.

The soil resistance r varies according to the type and condition of the soil. It is measured in lbs. per sq. in. of furrow cross-section, and may vary as much as 40 per cent. in one field.

Values of r are as follows :—

Type of Soil.	Sandy.	Sandy Loam.	Light.	Dry Clay Loam.	Heath Land.	Heavy Clay Sod.	Heavy Loam.	Firm Clay.
r	3	3–6	5–7	7–8	8·8	10–11	12	16

The 'angle of hitch' is most important. When the hitch is not correct vertically, either the share digs into the ground too deeply or does not penetrate at all.

A re-pointed and re-sharpened share will show a less draught than a dull one, while a new one may reduce the draught by 36 per cent. over the re-sharpened one.

Experiments carried out at Rothamsted indicated that the speed of ploughing could be considerably increased without proportionately increasing the draught. In fact, an increase of speed of from 1 to 2½ m.p.h., or 150 per cent., increased the draught by 14·5 per cent.

It was found that the draught in heavy clay could be reduced some 15 per cent. by chalking the soil.

See also 'Electric Ploughing,' by Borlase Matthews, *Journal of the Institution of Electrical Engineers*, No. 383, Nov. 1928.

(*R. Borlase Matthews.*)

See also 'Electricity in Agriculture' Section XXVI.

2. MECHANICAL POWER.

Mechanical power may be determined by measuring either a *force* or a *torque*, the second being the more common method.

POWER EXPENDED BY A FORCE.

$$\text{Horse-power} = \frac{\text{force (in lbs.)} \times \text{distance travelled per minute (in feet)}}{33{,}000}.$$

POWER EXPENDED BY A TORQUE.

$$\text{Horse power} = \frac{\text{torque (in lb.-feet)} \times 2\pi \times \text{R.P.M.}}{33{,}000}.$$

I. *Dynamometers measuring a Force.* (Traction or drawbar dynamometers.)

(*a*) *Spring Type.*—A coil or laminated spring is incorporated so that its deformation is a measure of the force transmitted by the drawbar. Previous calibration is necessary.

The spring dynamometer has the disadvantage of needing heavy damping. Mechanical multiplication is used to give a more open scale on the indicator. The type is useful when there are no sudden variations of load.

(*b*) *Hydraulic Type.*—The force is transmitted through a piston to fluid in a cylinder and measured as a fluid pressure on a gauge. The force is deduced from the pressure either directly by multiplying it by the piston area, or by a calibration.

Dynamometers of this type include:

(i) The *Hyatt* dynamometer, which uses a rubber bag connected to a recording Bourdon gauge with a radial chart.

(ii) The *Amsler* traction dynamometer, which has a ground and polished piston and cylinder with a coil spring pressure indicator giving a cylindrical chart.

(iii) The *Watson* traction dynamometer, based on an experimental dynamometer designed at the National Physical Laboratory, which uses a steel cylinder and plunger and a remote steam engine type pressure indicator.

(*c*) *Pendulum Type.*—The force to be measured is balanced by the restoring force on a pendulum. This type is often used on dynamometer cars to measure the work done by the locomotive, when it is known as an ergometer. The tangent of the angle made by the pendulum with the vertical is proportional to the algebraic sum of the force of gravity and the force causing acceleration or retardation on the track.

(*d*) *Strain Gauge Type.*—A steel tension bar is fitted with electrical strain gauges, which may be of the induction (moving coil or moving iron) or the resistance type. The gauges are connected either to an indicating milli-ammeter or an oscillograph (Duddell or Cathode Ray). Thus inertia effects due to the dynamometer itself are avoided, since the natural frequency of the deformed element of the dynamometer is high.

Speed-Recording Device.

The ordinary type of centrifugal speedometer is not accurate enough for use with a traction dynamometer. The method used is to record the distance travelled in a known time, by connecting the recording drum to a roadwheel and causing a pen to draw a continuous line with sideways kicks at known time intervals, the pen being worked magnetically or by clockwork.

Rail Dynamometer Cars.

These are used for traction tests on locomotives and resistance tests on rolling stock. They are usually interposed between the locomotive and the train and are fitted with facilities for measuring drawbar pull, buffer thrust, speed, and work done by the prime mover, as well as boiler pressure, flue gas analysis, etc.

The dynamometer is usually of the spring type, but sometimes hydraulic. The recording drum usually integrates the pull with respect to distance, giving a measure of the work done. An 'ergometer' (see above) is also fitted to measure work done in acceleration and deceleration.

II. *Dynamometers measuring a torque.*

These consist of:

(*a*) The *transmission or shaft type*, interposed between the prime mover and the driven apparatus and (*b*) the *absorption type*, where all the power measured is converted to heat or electrical energy.

Transmission or *shaft dynamometers* are chiefly useful where the power to be measured is so large that an absorption dynamometer would be wasteful or inconveniently large. They may be classed either as *torsion meters*, measuring the torque by means of the elastic twist of the shaft, or as direct *torque meters*, measuring the driving force at a given radius from the shaft centre.

(*a*) *Torsion Meters.*—These consist essentially of two sleeves concentric with the shaft and attached to it at opposite ends of the length over which the twist is measured.

(i) The *Föttinger* torsionmeter multiplies the twist by levers and records it on a drum concentric with the shaft. This instrument shows variations in torque during each separate revolution.

(ii) The *Hopkinson-Thring* torsion meter (see fig. 1) uses two mirrors attached to the ends of the test length of shaft, which successively reflect a beam of light on to a fixed scale. The distance between the two spots illuminated on the scale is a measure of the twist. The instrument is callibrated by applying a known twist to the shaft, or by calculation from the elastic constants of the shaft.

(iii) The *Bevis-Gibson* torsionmeter passes a beam of light parallel to the shaft through radial slits in two discs fixed at the ends of the test length. The twist is read off by a 'finder,' consisting of an eye-piece which can be moved circumferentially round the shaft. The finder is rotated until the beam of light, interrupted owing to the twist, is again visible. The movement of the finder measures the twist.

(iv) The *Amsler* torsionmeter has two discs, one attached to each end of the shaft, which are illuminated by a spark synchronised with the rate of revolution, so that reference scales on the discs appear to stand still and can be read off with a telescope.

(v) *Electrical* torsion meters may be of the condenser, induction, or resistance strain gauge type. In the first type, twist of the shaft varies the capacity of a condenser; in the second, the inductance of a pair of coils is changed; the resistance type has resistance elements attached to the shaft so as to indicate torsional strain. The Siemens torquemeter is a good example of the electrical type.

(b) *Direct Torque Meters* work by measuring the driving force at a fixed radius either by means of springs, by hydraulic pressure, by a system of levers connected to a jockey weight or pendulum, or by the difference in tension between the tight and slack sides of a driving belt. A correction is usually needed to allow for the centrifugal force on the mechanism, and it is best to calibrate this type against an absorption dynamometer.

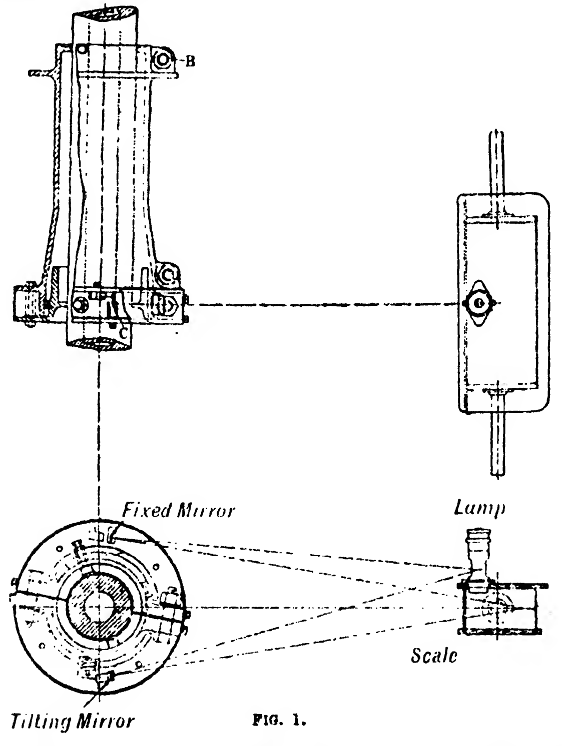


FIG. 1.

(i) The *Ayrton-Perry* torquemeter consists of two parallel discs on the ends of the driving and driven shafts, connected by three tangential coil springs, whose deformation under load causes relative twist between the discs. This is multiplied by a lever system so as to give a radial displacement of a bright bead observed against a graduated dark disc.

(ii) The *Amsler* hydraulic torquemeter works similarly to the above, the coil springs being replaced by oil cylinders connected by an axial hole in the shaft to an indicator of the steam engine type.

(iii) A simple type of torquemeter useful when the driven shaft is parallel to the driver but at a distance from it is one developed at the National Physical Laboratory for use with an abrasion testing machine. Here an idler gear is interposed between gears on the two shafts and the sideways thrust is measured by a weighted lever extension on the idler housing (see fig. 2). The thrust is approximately double the load at the pitch circles of the driving and following gears. (See N.P.L. Annual Report for 1932, p. 181.)

(iv) The *Thornycroft* or *Froude* belt dynamometer uses two pulleys interposed between the driving and driven pulleys and arranged on a swinging frame with a lever arm carrying weights which balance the belt tension.

(v) *Bailey and Bowers'* torquemeter is designed for light loads such as the drives of certain types of textile machinery. It operates by transmitting the torque through two concentric rings, relative motion of which is opposed by a pendulum connected through linkage.

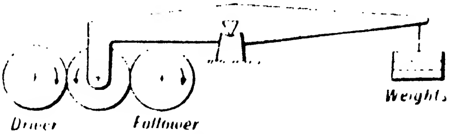


FIG. 2.

III. *Absorption Dynamometers (Brakes).*

Where the power to be measured is not inconveniently large, the absorption dynamometer is the most accurate means of measurement.

The power may be absorbed by solid friction, fluid (water or air) friction, or electrical resistance.

Solid Friction Brakes.

(i) *Hirn's or Kelvin's Rope Brake.* (For small powers.)—A band of rope encircles a pulley on the power shaft. One end of the rope holds a scale pan and the other is attached to a spring balance, as shown.

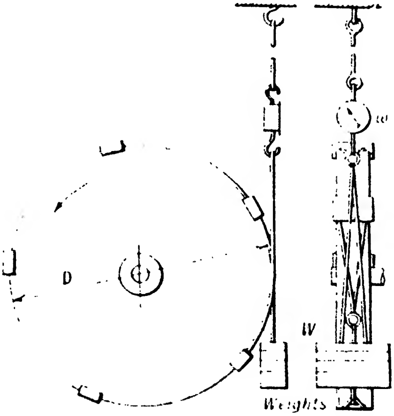


FIG. 3.

$$\text{B.H.P.} = \frac{(W - w) \times \pi DN}{33,000}$$

where W = weight on scale pan; D = diameter to centre of rope;
w = load on spring balance; N = R.P.M.

The arrangement shown has the advantage that an increase in friction tends to slacken the rope, giving stability.

(ii) *The Prony Brake.*—A couple of wooden blocks clipped upon the pulley or shaft transmitting the power to be indicated. The diameter D in the formula for Hirn's brake is, in the case of the Prony brake, double the radius R measured to the point of suspension of the pan.

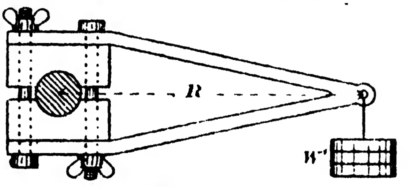


FIG. 4.

The difficulty with the Prony brake is the constant vibration of the weighted arm. The difficulty has been reduced at Purdue University by the addition of a pendulum lever weight, B, fig. 5, the quadrant of which may be divided by actual weight determination.

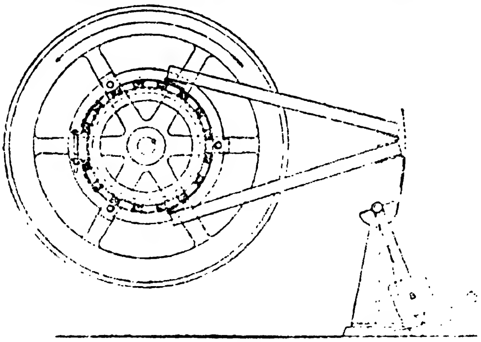


FIG. 5.

FIG. 6.

Fig. 6 shows a form of stabiliser which, it is stated, will damp down the vibrations to a very marked extent. Between the balance and the brake arm are placed in series a heavy weight A and a resilient coil spring B. The pointer of the balance is then set back to offset the initial force, or tare, exerted by the weight. When the motor being tested is in motion, the average torque is transmitted through the spring and suspended weight, and is accurately indicated on the dial of the balance. The momentary vibrations, however, are not of sufficient duration to move the weight, and their motion is dissipated by the auxiliary spring. By employing this type of stabiliser, an oil dashpot may be dispensed with entirely, and readings of a high degree of accuracy may be obtained.

(iii) *The Balk Brake.*—Many methods have been used to compensate for variations in the friction coefficient between the brake blocks and wheel. In the Balk brake (see fig. 7) the tightness of the band is controlled by a compensating lever A with a suspended weight, working between fixed stops X-X. An increase in friction causes the band to carry the lever downwards on to the lower stop; it then rotates clockwise and slackens the band by causing B to move upwards relative to C. A fall in friction causes the opposite effect.

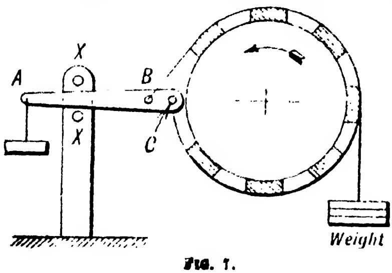


FIG. 7.

Cooling Brake Rims.

In many cases the inside of a brake rim is of trough form (fig. 8), which will hold water when revolving, by virtue of the centrifugal action. The trough being filled with water is kept to (at most) 212° F. by the evaporation of the water, which is kept constantly running into the rim in a small stream.

Another means of cooling is to encircle a sheet-steel brake band with iron wire netting, which holds by capillary action water poured on it; this evaporates and cools the strap. Surplus water runs off at the bottom of the band.

Pullen's brake uses a white-metal brake lining with interior water cooling.

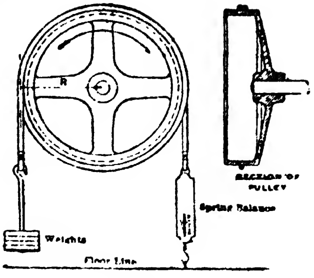


FIG. 8.

FLUID FRICTION BRAKES.

A simple fluid friction brake can be furnished by causing a series of metal discs to rotate in a water bath. The power absorbed at a given speed can be varied by altering the amount of water in the casing. Unsteady action due to splashing can be reduced by radial interior vanes on the casing; the reaction on the latter will then be equal to the resistance to rotation of the vanes, and can be measured by allowing the casing to swing about the shaft axis and balancing the torque by weights on a lever arm. This principle is used in the ' Brotherhood ' absorption dynamometer.

THE FROUDE HYDRAULIC DYNAMOMETER.

In the Froude Water Dynamometer the engine to be tested is directly coupled to the main shaft, transmitting the power to a rotor revolving inside a casing, through which water is circulated to provide hydraulic resistance and simultaneously to carry away the heat developed by absorption of power.

In each face of the rotor are formed pockets of semi-elliptical cross-section divided one from another by means of oblique vanes. The internal faces of the casing are also pocketed in the same way.

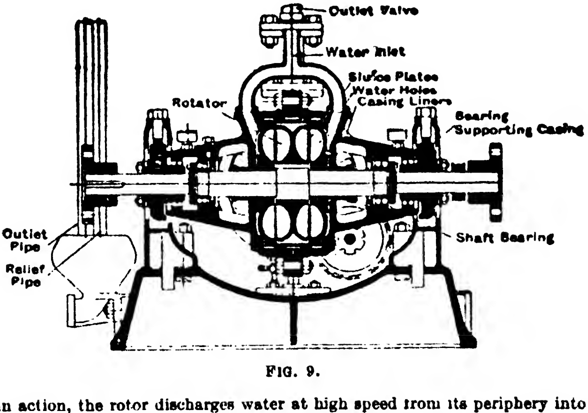


FIG. 9.

When in action, the rotor discharges water at high speed from its periphery into the casing pockets, by which it is then returned at diminished speed into the rotor pockets at a point near the shaft. Thus, the pockets in rotor and casing together form elliptical receptacles round which the water courses at high speed, creating vortices which dissipate the power of the engine as quickly as it is developed.

Hence the rotor causes a reaction on the casing, which endeavours to revolve, but is resisted by a weighing machine calibrated to read the power absorbed. Several types are made.

The Fan Dynamometer.

Owing to the important influence of atmospheric conditions and of environment, the fan brake is more frequently used for applying a load to a rotating shaft than for actually measuring the torque. The power unit is usually mounted on a swinging frame or 'reaction bench' and the torque is measured by a weighted lever on the frame.

The load is adjusted either by altering the radius of the vanes or by fitting vanes of a different size. Where the fan is to be used for measuring torque, the following constants and formulæ may be used:

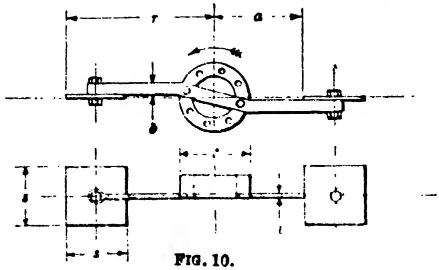


FIG. 10.

If

w = torque driving the fan in inch-lbs.; w_a = resisting torque due to the arms alone; w_b = resisting torque due to the blades alone; N = revs. per min. of the fan;

s = length of side of square blades or diameter of round blades in inches;

r, a, b, t, and c are other dimensions in inches;

W = density of the fluid in which the fan rotates, in lbs. per cubic foot,

$= 62\cdot33$ for water at 50° F. $= \dfrac{1\cdot317 \times \text{barometric pressure in in., for air}}{(460 + \text{temp. in °F.})}$

H.P. = horse-power absorbed by fan;

K = a coefficient depending upon the value of r/s and upon the shape of the blade; see values tabulated below for square and circular blades having square edges.

Values of r/s	2·5	3·0	3·5	4·0	4·5	5·0	5·5	6·0	6·5	9·0	12·0
$K \times 10^{-9}$ for circular blades	1·69	1·69	1·66	1·63	1·60	1·59	1·58	1·575	1·575	1·575	1·575
$K \times 10^{-9}$ for square blades	2·25	2·28	2·24	2·18	2·16	2·12	2·10	2·08	2·04	2·04	2·04

σ = a coefficient depending upon the ration b/t.

Values of b/t	2	3	6
Values of σ	1·3	0·94	0·62

then,

$$w_a = 0\cdot35 + 10^{-8}\sigma N^2 a^4 t W\,;\quad w_b = KN^2 r^3 s^2 W\,;\quad w = w_a + w_b{}^{*}\,;\quad \text{H.P.} = wN/63{,}000.$$

Notes on the Fan Dynamometer.

Care should be taken that the arms and fastenings are amply strong enough to withstand the centrifugal stresses. For reasonably accurate work, with fans rotating in air, one side of the fan should be completely free from obstruction for a distance of at least '3 r' inches from the

* For this relation to hold, t must be less than $s/7$.

plane of rotation, and on the other side 'O' should not exceed '*a*.' Also the floor should be distant at least '*r*' inches. The linear velocity of the centre of the blades must not exceed, say, 350 ft. per second. In order to compute W with accuracy, the barometric pressure, and the temperature of the air at the periphery of the fan should be taken. Fans rotating in water must have at least as much clearance around them as is specified above, and must not be rotated at such a speed as to produce cavitation. To facilitate work in the test-shop the '*H.P., Rev.' curves for each blade used should be plotted on logarithmic paper for some standard value of W. Since the H.P. varies approximately as N^3, N should be observed with great accuracy—a speedometer is usually not sufficiently accurate.

*See also Hodgson, *Proc. Inst. Automobile Engineers*, March 1916.

(*J. L. Hodgson.*)

ELECTRICAL DYNAMOMETERS.

1. *Generator Type.*—An electric generator may be adapted as a dynamometer, but this involves knowing its efficiency and electrical measuring instruments are needed. When high accuracy is not required, the method is sometimes economical, especially when the current generated can be used. Higher accuracy is obtained by mounting the generator field frame to swing freely about the shaft axis and measuring its reaction torque by a weighted lever. The electrical power is then absorbed by resistance mats. A suitable mounting may be obtained either by fitting ball-bearing trunnions to the bearing caps of the frame or by arranging the frame to swing in a cradle in which it rests on its base. The weight of the field frame must be balanced so as to be just stable when the torque is zero.

The range of power and speed available with the electric dynamometer is usually comparatively small, unless complicated auxiliary electrical equipment is provided, but this disadvantage is balanced in that the generator may be used as a motor to determine the power used in driving a machine.

2. *Eddy Current Type.*—A comparatively recent development (at any rate for high powers) is the eddy current or 'dynamatic' dynamometer, in which a magnetic field is set up by the action of a relatively small control current in the stator windings. A special rotor operating in this magnetic field provides the power-absorbing torque through the action of eddy currents, the energy being disposed of partly by air and partly by water cooling. This system has the advantage of providing high torque down to relatively low speeds.

REGENERATIVE OR POWER CIRCULATING SYSTEMS.

For testing gearboxes or transmissions it is economical to measure power losses by connecting output to input by a return circuit (either electrically or mechanically), and supplying only the power loss in the system from an outside source. The test load is varied either by inserting mechanical distortion in the circuit (*e.g.* belt slip or shaft torsion) or by Ward-Leonard control if the return circuit is electrical.

(For further information on power measurement, see Batson and Hyde, 'Mechanical Testing,' vol. ii, chaps. i-ix, also Drysdale, *Journ. Jun. Inst. Engineers.* (London), vol. 48, part ii, p. 499 (August, 1938), 'The Measurement of Mechanical Power.')

See also Descriptive Section II, Part II.
Crosby Valve & Engineering Co., Ltd.
Heenan & Froude, Ltd.

SECTION III

HEAT — OPTICS (pp. 49 65)

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

SECTION III

HEAT—OPTICS.

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

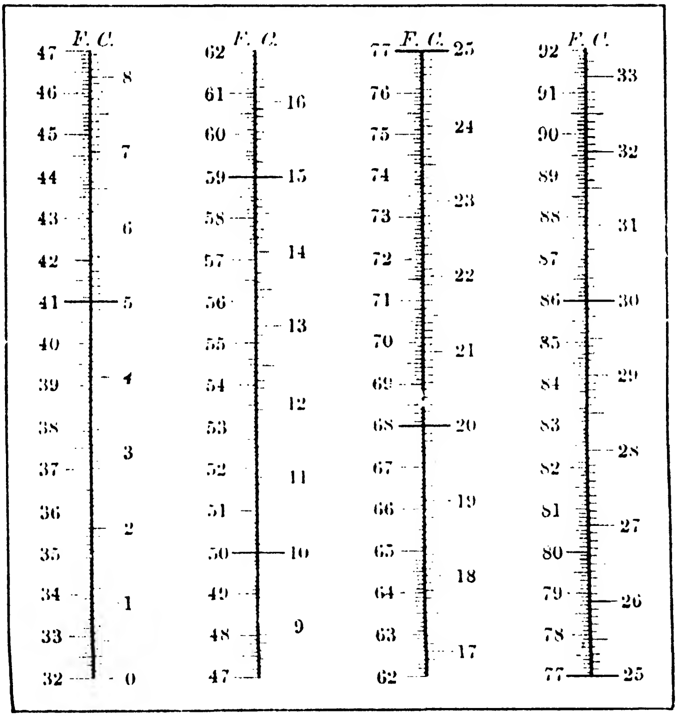
HEAT.

Temperature Scales.

The following three temperature scales are those most commonly in use :—

Scale.	Fahrenheit (F.).	Centigrade (C.).	Réaumur (R.).
Freezing point of water	32°	0°	0°
Boiling point of water.	212°	100°	80°

Thermometer Scale Conversion Chart.



(R. E. Gould American Bureau of Standards.)

Conversion Factors.

°C. contain °F.			°F. contain °C.		
1 = 1·8	4 = 7·2	7 = 12·6	1 = 0·56	4 = 2·22	7 = 3·89
2 = 3·6	5 = 9·0	8 = 14·4	2 = 1·11	5 = 2·78	8 = 4·44
3 = 5·4	6 = 10·8	9 = 16·2	3 = 1·67	6 = 3·33	9 = 5·00

$$°C = \frac{5}{9}(°F - 32°) = \frac{°F - 32°}{1·8} \qquad °F = \frac{9}{5}°C + 32° = 1·8° C + 32°.$$

Colour of Different Temperatures.

Colour.	°F.	°C.	Colour.	°F.	°C.	Colour.	°F.	°C.
Faint Red .	960	516	Bright Cherry Red	1830	999			
Dull Red .	1290	699	Orange . .	2010	1099	Bright White	2550	1399
Brilliant Red	1470	743	Bright Orange .	2190	1199			
Cherry Red.	1650	899	White . .	2370	1299	Brilliant White	2750	1510

(*Becquerel.*)

Absolute Zero of Temperature.

Definition.—The temperature at which the pressure of a given mass of an ideal gas, maintained at constant volume, becomes zero.

Absolute zero is − 460° F. or − 273° C.

Freezing Temperatures.

Sea water, − 2·5° C., 27·5° F.; Mercury, − 38·8° C., − 37·8° F.; Acetone, − 95° C., − 139° F.; No. 1 motor spirit, − 128° C., − 198° F.

Melting Points.

MELTING POINTS OF VARIOUS SUBSTANCES.

Solids.	°F.	°C.	Solids.	°F.	°C.
Litharge . . .	1612	877	Ice of milk . .	30°	− 1·1°
Slag of copper . .	2462	1350	„ sea-water. .	28	− 2·2
„ tin . .	2402	1318	„ vinegar . .	28	− 2·2
Puddle slag . .	2606	1430	„ strong wine .	20	− 6·6
Sodium nitrate . .	591	310·5	„ „ brandy .	7	− 13·9
Potassium nitrate .	612	339	1 snow, 1 salt . .	00·0	− 17·8
Beeswax (white) .	155	68	1 alcohol, 1 water .	− 7	− 21·6
„ (yellow) .	142	61	Cyanogen . .	− 31	− 35
Spermaceti . .	113	45	Sulphuric acid, density 1·65 . .	− 30	− 35
Paraffin wax . .	131	55	Sulphurous acid. .	− 105	− 76
Olive oil . . .	92	33	Nitrous oxide . .	− 150	− 101
Tallow . . .	111	44			
Ice of water . .	32	0·000			

MELTING POINTS OF REFRACTORIES.—See p. 371.

Melting Points of Chemical Elements.

Element.	°F.	°C.	Element.	°F	°C.
Helium	< −456	< −271	Barium	1299	704
Hydrogen	−434	−259	Praseodymium	1725	940
Neon	−416	−249	Germanium	1756	958
Fluorine	−369	−223	Silver	1761	960·5
Oxygen	−360	−218	Glucinum	—	> Ag
Nitrogen	−346	−210	Radium	1760	960?
Argon	−306	−188	Gold	1945·5	1063·0
Krypton	−272	−169	Copper	1981·5	1083·0
Xenon	−220	−140	Manganese	2268	1242
Chlorine	−150·5	−101·5	Yttrium	2720	1490
Mercury	− 37·7	− 38·7	Samarium	2370–2550	1300–1400
Bromine	+ 18·9	− 7·3	Scandium	2190	1200
Cæsium	79	26	Silicon	2588	1420
Gallium	86	30	Nickel	2646	1452
Rubidium	100	38	Cobalt	2697	1480
Phosphorus	111·4	44	Chromium	3326	1830
Potassium	144	62·3	Iron	2768	1520
Sodium	207·5	97·5	Palladium	2820	1549
Iodine	236·5	113·5	Zirconium	3100	1700?
Sulphur S_{I}	235·0	112·8	Thorium	3056	1690
Sulphur S_{II}	246·6	119·2	Vanadium	3110	1710
Sulphur S_{III}	224·2	106·8	Platinum	3191	1755
Indium	311	155	Beryllium	2337	1281
Lithium	367	186	Ytterbium	—	?
Selenium	422–428	217–220	Titanium	3272	1800
Tin	449·4	231·9	Rhodium	3552	1955
Bismuth	520	271	Ruthenium	3453	1900
Thallium	576	302	Columbium		
Cadmium	609·6	320·9	(Niobium)	3540	1950
Lead	621·1	327·4	Boron	4000–4500	2200–2500
Zinc	786·9	419·4	Iridium	4170	2300?
Tellurium	846	452	Uranium	—	?
Antimony	1166	630·0	Molybdenum	4447	2450
Cerium	1153	623	Osmium	4900	2700?
Magnesium	1217	659	Tantalum	5160	2850
Aluminium	1217·7	658·7	Tungsten	5972	3300
Calcium	1563	851	Carbon	> 6500 for p = 1 At.	> 3600 for p = 1 At.
Lanthanum	1520	826			
Strontium	1419	771			
Neodymium	1544	840			
Arsenic	1497	814?			

? Volatilise, but do not melt.

Melting Points of Low Melting-point Alloys.

Bismuth.	Lead.	Tin.	Melting Point. °F.	Melting Point. °C.	Bismuth.	Lead.	Tin.	Melting Point. °F.	Melting Point. °C.
50	31·2	18·8	201	94	16·6	33·2	50·2	316	158
47	35·5	17·7	208	98	16	36	48	311	155
42·1	42·1	15·8	226	108	15·3	38·8	45·9	309	154
40	40	20	235	113	14·8	40·2	45	307	153
36·5	36·5	27	243	117	14	43	43	309	154
33·3	33·3	33·3	253	123	13·7	44·8	41·5	320	160
30·8	38·4	30·8	266	130	13·3	46·6	40·1	329	165
28·5	43	28·5	270	132	12·8	49	38·2	342	172
25	50	25	300	149	12·5	50	37·5	352	178
23·5	47	29·5	304	151	11·7	46·8	41·5	333	167
22·2	44·4	33·4	289	143	11·4	45·6	43	329	165
21	42	37	289	143	11·2	44·4	44·4	320	160
20	40	40	293	145	10·8	43·2	46	318	159
19	38	43	298	148	10·5	42	47·5	320	160
18·1	36·2	45·7	304	151	10·2	41	49·8	322	161
17·3	34·6	48·1	311	155	10	40	50	324	162

The addition of cadmium gives alloys of still lower melting point. The following table includes the best known of these :—

Alloy.	Cadmium.	Lead.	Tin.	Bismuth.	Melting Point.	
					° F.	° C.
Fusible alloy . .	12·5	25	12·5	50	149	65
Lipowitz's alloy .	10	26·6	13·3	50·1	158	70
Woods' alloy . .	15·4	30·8	15·4	38·4	160	71
Fusible alloy . .	34·5	27·5	10	27·5	167	75
" " . .	6·2	34·5	9·3	50	171	77
" " . .	25	25	50	—	187	86
" " . .	16·6	—	33·3	50·1	203	95
" " . .	11·1	—	33·3	55·6	203	95
" " . .	25	—	25	50	203	95

(*Parkes and Martin.*)

Boiling Points.

Boiling Points of Saturated Solutions of Various Salts.

Salt.	C.°	F.°	Salt.	C.°	F.°
Sodium nitrate . .	120·7	249·5	Sodium di-phosphate .	106·6	224
Potassium nitrate . .	115·7	240	Sal-ammoniac . .	114·2	237·6
" chlorate .	104	219·2	Calcium chloride . .	179·4	355
Sodium chloride . .	108·2	227	Potassium acetate .	169	336
Potassium chloride . .	108	226·4	" carbonate .	134 7	275
Sodium carbonate . .	104·6	220	Calcium nitrate . .	151	304
" acetate . .	124·5	256	Strontium chloride .	118	244
Barium chloride . .	104·2	220	Potassium tartrate .	114·4	238

Boiling Points of Water at Various Pressures.

Barometer in Inches.

Boiling-point. F.°	Barometer. Inches.	Boiling-point. F.°	Barometer. Inches.	Boiling-point. F.°	Barometer. Inches.	Boiling-point. F.°	Barometer. Inches.
185	17·047	193	20·254	201	23·937	209	28·183
186	17·421	194	20·687	202	24·441	210	28·744
187	17·803	195	21·124	203	25·014	211	29·331
188	18·196	196	21·576	204	25·468	212	29·922
189	18·593	197	22·030	205	25·992	213	30·516
190	18·992	198	22·498	206	26·529	214	31·120
191	19·407	199	22·965	207	27·068	215	31·730
192	19·822	200	23·454	208	27·614	216	32·350

Barometer in Millimètres.

Boiling-point. C.°	Barometer. Millimètres.	Boiling-point. C.°	Barometer. Millimètres.	Boiling-point. C.°	Barometer. Millimètres.	Boiling-point. C.°	Barometer. Millimètres.
85·5	441·62	89·5	515·53	93·5	599·49	97·5	694·56
86	450·34	90	525·45	94	610·74	98	707·26
86·5	459·21	90·5	535·53	94·5	622·17	98·5	720·15
87	468·22	91	545·78	95	633·78	99	733·21
87·5	477·38	91·5	556·19	95·5	645·57	99·5	746·50
88	486·69	92	566·76	96	657·54	100	760·00
88·5	496·15	92·5	577·50	96·5	669·69	100·5	773·71
89	505·76	93	588·41	97	682·03	101	787·63

Ignition Temperatures.

Name of Substance.	° F.	° C.	Name of Substance.	° F.	° C.
Phosphorus, transparent .	112	44·4	Rifle-powder . . .	550	287·8
Bisulphide of carbon . .	300	143·3	Forced gunpowder . .	560	293·3
Fulminating powder . .	370	187·8	Picrate powder for torpedoes	570	299·0
Fulminate of mercury .	390	198·9	Charcoal from willow wood.	660	349·0
Gun-cotton	430	221·1	Picrate powder for cannon .	720	382·2
Nitro-glycerine . . .	490	254·4	Very dry pine wood . .	800	426·7
Phosphorus, amorphous .	500	260·0	Dry oak wood . . .	900	480·2

Flash Temperatures.

Dr. H. Holm finds that the temperature at which substances in contact with atmospheric air at ordinary pressures will catch fire are, among others:—Lighting gas, 600° C.; benzine, 415° C.; petroleum, 380° C.; gas oil, 350° C.; machine oil, 380° C.; coal tar, 500° C.; tar oil, 580° C.; and benzol, 520° C. These temperatures are greatly influenced by the presence of other substances, which act as catalysers. This is especially the case with hydrogen, methane, and ethane, not ethylene or acetylene. Among solids it is the substances with the biggest molecules which most readily burst into flame. As a broad rule, the nearer a substance is to a gaseous condition, the higher the temperature to which it must be exposed before it will kindle: but this depends also on the chemical constitution.

Freezing Mixtures.

Mixture.	Reduction in Temperature.	Mixture.	Reduction in Temperature.
1 part nitre 1 part sal ammoniac 2 parts water . .	40° F.; 22° C.	5 parts hydrochloric acid 8 parts crystallised sulphate of soda . .	50° F.; 28° C.
Equal parts crystallised nitrate of ammonia and water	46° F.; 25½° C.	2 parts powdered ice or fresh snow . . . . 1 part common salt .	a temperature of −4° F.; −20° C.
Equal parts crystallised nitrate of ammonia (powdered), crystallised carbonate of soda (powdered), and water	57° F.; 32° C.	3 parts crystallised chloride of calcium (cooled to 32°) . . . . 2 parts fresh snow . .	a temperature of −50° F.; −45·6° C.

Anti-freezing Mixtures.

Common Salt (Sodium Chloride).				Commercial Calcium Chloride.			
Lbs. per Gal.	Freezing Point, F.°	Lbs. per Gal.	Freezing Po F.°	Lbs. per Gal.	Freezing Point, F.°	Lbs. per Gal.	Freezing Point, F.°
½	24 above zero	2	6 above zero	½	29 above zero	3½	6 below zero
1	18 ,,	2¼	3 ,,	1	27 ,,	4	17 ,,
1¼	15 ,,	2½	1 ,,	1½	23 ,,	4½	27 ,,
1½	12 ,,	3	3 below zero	2	18 ,,	5	39 ,,
1¾	9 ,,	3½	8 ,,	2½	4 ,,	5½	54 ,,

The salt should be thoroughly dissolved, or the results will not be satisfactory. Calcium chloride is said to be superior to sodium chloride in that it does not corrode steel tanks and barrel hoops. Where calcium chloride is used, the wooden barrels should first be well coated inside with asphaltum, or a mixture of crude paraffin and resin, to prevent the shrinking of the staves and subsequent leakage.

Glycerine is recommended for use in chemical fire extinguishers. It has no effect upon metals, but has a tendency to disintegrate rubber. It has been stated, however, that the continued use of glycerine in water renders it liable to a decomposition that would develop compounds having a corrosive action on metals. The following proportions may be used:—

Glycerine, pounds per gallon . .	3½	5½	(One gall. of glycerine
Temp. solution will withstand . .	+ 10° F.	− 10° F.	weighs 10½ lbs.)

Denatured alcohol has advantages over both glycerine and calcium chloride, in water solutions, as it has no injurious effect on either metal or rubber. A solution of about 50 per cent. is inflammable, but it would rarely be necessary to have a solution of over 30 per cent.

A 20 per cent. solution freezes at	10° F.	A 40 per cent. solution freezes at	− 20° F.
30 " " "	− 5° F.	50 " " "	− 35° F.

Temperature of the Atmosphere

diminishes according to the height above the earth's surface. In the Temperate Zone a diminution of 1° C. corresponds in the mean to an ascent of 180 yards. As the mean of a series of observations made during balloon ascents, a diminution of 1° C. corresponded to an increase in height of 232 yards.

According to Kropotkin, the temperature above the earth's surface decreases 3° to 5° F. for each 1,000 ft. The average temperature at an altitude of 20,000 ft. is 13° below 0° F.; at 25,000 ft. it is 35° below 0° F.; these low temperatures prevail all the year round. The ratio of decrease in temperature, which is 3° F. for each 1,000 ft. at the lower strata, is twice as great at the highest levels, but varies from summer to winter to the extent of about 20° F.

The Mean Daily Temperature is that obtained by adding together twenty-four hourly observations and dividing by 24, the temperature being taken of the air, and not of the ground. The temperature of a month is the mean of those of thirty days, and the temperature of the year is the mean of those of twelve months. The mean temperature of London is 8·28° C., or 46·9° F.; the mean in summer is 62° F., and in winter 40° F.

Temperature of Seas and Lakes.

The temperature of the sea under the Torrid Zone is always about 26° to 27° C. at the surface; it diminishes as the depth increases, and in temperate as well as in tropical regions the temperature of the sea at great depths is between 2·5° and 3·5° C. The average winter temperature of the sea round the coast of England is higher than that of the land. The mean annual temperature of the surface of the sea round England is 49° F.; the mean surface temperature of the Indian Ocean is 89° F., and of the Red Sea 94° F. The temperature of lakes at their surface may be 20° to 25° C. in summer; the temperature of the bottom, 4° C., which is that of the maximum density of water. The Great Geyser, in Iceland, at a depth of 66 feet, has a temperature of 124° C.

Temperature of the Earth.

The average increase in temperature is 1° C. for every 27·4 metres of descent, which is at the rate of 100° F. per mile. Cordier says 1° C. for every 25 metres, or 1° F. in 45 feet.

Another statement is that the increase in temperature of the earth's strata shows a mean average for the entire earth surface of 1° F. for every 56 feet of depth.

Thermal and Other Properties of Gases, &c.

(*W. N. Booth.*)

In the tables, pages 56 to 59, are given many figures relative to the gases chiefly, which are of interest in questions of heat, combustion, and the gas engine. The figures are principally based on Berthelot's thermal determinations of the calorific capacity of the gramme molecule, the figure for which, if divided by the molecular weight, gives the heat generated per gramme of the substance. An attempt is made to give the correct values of the specific heat at high temperatures, with a view to the bearing of this property on the theory of the gas engine, and the possibly erroneous ideas as to the supposed effects of dissociation, which have been much overrated. The thermophysical properties of carbon are more fully tabulated than has hitherto been attempted, with a view to explain the variations due to allotropism.

Heat and Combustion.

There are in practice only two elementary fuels, carbon and hydrogen. They generate heat by union with oxygen, and their compounds are also available as fuels.

Heat is measured in units. The British thermal unit B.Th.U. is the amount of heat necessary to raise 1 pound of water from 60° F. to 61° F., and its mechanical equivalent is J = 778 pounds raised 1 foot in the latitude of Manchester.*

The metrical or French calorie is the amount of heat necessary to raise 1 gramme of water 1° C. This is the 'gramme calorie.' The kilogramme calorie is the heat necessary to raise 1 kilogram of water through 1° C. Its mechanical equivalent is 427 kilograms raised 1 metre at the

* The latitude need not be considered by engineers.

latitude of Paris; this latter refinement being unnecessary in engineering. The following table gives the ratios of English and French units:

1 Calorie = ·001 kilog. calorie.	1 B.Th.U. = 1055 joules = 778 foot-pounds (until recently the accepted value was 772).
1 kilog. calorie = 3·968 B.Th.U.	1 kilog. calorie = 3087·3 foot-pounds.
1 B.Th.U. = 0·252 kilog. calorie.	1 B.Th.U. = 107·6 kilog. metres.
1 kilog. calorie = 127 kilog. metres.	

Chemists use somewhat different terms and state calorific measurements in terms of the molecule. The gramme molecule of hydrogen is the unit employed. One gramme molecule of hydrogen, H_2, taken at 0° C. and 760 mm. of pressure of a mercury column, weighs 2 grammes and has a volume of 22·32 litres. The molecular weight of any other simple or compound gas is the weight which occupies the volume occupied by 2 grammes of hydrogen, that is 22·32 litres. The molecular specific heat of a gas is the heat of this volume in 'petit calories.' Then the molecular heat divided by the molecular weight will give the specific heat per unit weight or gramme; and the molecular heat divided by 22·32 will give the specific heat per unit of volume.

The specific heat of a gas is usually stated at constant pressure and at constant volume. In perfect gas the ratio of these two is about 1·4 at ordinary temperatures. This is an erroneous way of putting it. The difference is really due to work done on the atmosphere in expanding to constant pressure. This work is invariable at 1·91 calories. The true ratio of specific heat at constant pressure or Cp to that at constant volume Cv, is thus properly stated, $Cp = Cv + 1·94$, or usually $= Cv + 2$. Thus, if 22·32 litres of gas are heated 1° C. from 0° C. they will perform work against a column of mercury 760 mm. high to the extent of 1·94 calories at all temperatures. A gas expands when heated 1° C. to an additional bulk = $\frac{1}{273}$ of the bulk occupied at 0° C. Hence the invariable quantity 1·94.

The specific heat of a gas increases with temperature. The number of calories generated by the combustion of a unit weight of a substance if divided by the specific heat of the products of combustion and by the total weight of such products, will give the nominal temperature. The 'theoretical' temperature of combustion is the nominal temperature found by using the specific heat at ordinary temperatures. These nominal temperatures are never attained. The ordinary specific heat of carbon dioxide is quadrupled at 4000° C., and the temperature that might theoretically reach 18000° C., in oxygen in a certain case would not go beyond 4500° C. or thereabouts.

The calorific capacity of fuel may be approximated by calculating the capacity of its constituents of carbon and hydrogen. This is absolutely accurate, subject to allowance for the heat of formation. Thus in the formation of some hydrocarbons, heat is evolved. When these hydrocarbons are burned, the heat which was produced in their original formation cannot be again produced, and the combustion is productive of so much less heat. Others, such, for example, as acetylene, absorb heat when formed, and this substance when burned gives out more heat than corresponds with its carbon and hydrogen. This is one reason for its brilliance, and accounts for its explosive tendency.

HEAT OF COMBUSTION OF SUBSTANCES.

	Calories per gramme.	B.Th.U. per lb.	Authority.
Cellulose	4,200	7,560	Berthelot
Soft resinous wood	5,050	9,090	Gottlieb
Hard wood	4,750	8,550	,,
Peat	5,940	10,692	Bainbridge
Cane sugar	3,961	7,130	Berthelot
Asphalt	9,532	17,159	Slossom & Colburn
Pitch	8,400	15,120	Anon.
Naphthalene	9,690	16,842	Berthelot
Paraffin	9 800	17,600	Mahler
Tallow	9,500	17,100	Stohmann
Sulphur	2,500	4,500	Berthelot
Petroleum	9,600 to 11,000	17,280 to 19,800	Various
Schist oil	9,000 to 10,000	16,200 to 18,000	,,
Heavy coal gas oil	8,900	16,020	Ste-Claire Deville
Cotton oil	9,500	17,100	Anon.
Rape oil	9,489	17,080	Stohmann
Olive oil	9,473	17,051	,,
Sperm oil	10,000	18,000	Gibson

Name.	Symbol.	Specific gravity H = 1.	Molecular Weight.	Lbs. per cubic ft.	Cubic ft. per lb.	Grams per litre.	Litres per gram.	Required to burn one unit. Weight. Air.	Weight. Ox.	Volume Air.	Volume Ox.
Air	O_{21} N_{78} X_1	14·44	—	·08073	12·385	1·29318	·773	—	—	—	—
Ammonia	NH_3	8·5	17	·05324	18·783	·761	1·313	6·13	1·41	3·58	·750
Carbon C. Diamond	—	—	12	—	—	—	to CO / „ CO_2	5·797 / 11·594	1·334 / 2·667	—	—
Carbon C. Graphite	—	—	„	—	—	—	to CO_2	do.	do.	—	—
Carbon C. Amorphous	—	—	„	—	—	—	to CO / „ CO_2	„	„	—	—
Carbon C. Vapour	—	12	„	·06696	14·930	1·0727	·932	„	„	9·54	2·00
Carbon dioxide	CO_2	22	44	·12344	8·147	1·967	·508	—	—	—	—
Carbon monoxide	CO	14	28	·07817	12·80	1·2515	·800	2·484	·571	2·381	·500
Carbon bisulphide	CS_2	38	76	·21242	4·706	3·4058	·2947	5·478	1·26	14·30	3·00
Helium	He	1·98	4	·01115	89·8	·1786	5·59	—	—	—	—
Hydrogen	H_2	1	2	·00559	178·83	·08961	11·16	34·785	8·000	2·39	·500
Oxygen	O_2	16	32	·08926	11·203	1·4298	·699	—	—	—	—
Nitrogen	N_2	14	28	·07845	12·763	1·25616	·796	—	—	—	—
Steam	H_2O	9	18	·05022	19·912	·8047	1·242	—	—	—	—
Steam at 2,000°	H_2O	—	„	—	—	—	—	—	—	—	—
Steam at 4,000°	H_2O	—	„	—	—	—	—	—	—	—	—
Acetylene	C_2H_2	13	26	·07267	13·456	1·190	·840	13·378	3·077	11·93	2·500
Benzene	C_6H_6	39	78	·208	4·808	3·333	·303	13·378	3·077	35·80	7·500
Ethylene	C_2H_4	14	28	·07814	12·797	1·2519	·799	14·903	3·428	14·30	3·000
Ethane	C_2H_6	15	30	·08565	11·950	1·3415	·746	16·484	3·733	16·70	3·500
Methane	CH_4	8	16	·04466	22·391	·7155	1·397	17·392	4·000	9·54	2·000
Ethylene chloride	$C_2H_4Cl_2$	49·5	99	·2767	3·631	4·4357	·22626	3·855	·808	11·93	2·500
Ethyl Alcohol	C_2H_6O	23	46	·12857	7·775	2·061	·287	9·074	2·037	14·30	3·000
Methyl Alcohol	CH_4O	16	32	·08926	11·203	1·4298	·699	6·521	1·500	7·15	1·500
Cyanogen	C_2N_2	26	52	·1453	6·88	2·338	·427	5·348	1·23	9·54	2·000
Glycerine	$C_3H_8O_3$	—	92	—	—	—	—	18·148	4·174	16·70	3·500
Blast Furnace Gas. $[CO]_{27}$ $N_{55}(CO_2)_6$ H_2		14±	—	·079	12·65	1·2515	·800	1·00 / ·721	·22 / ·166	·82	·164
Producer Gas, $[Sundry]_{1\cdot5}$	$[CO_{25}]$ N_{56} $[CO_2]2\frac{1}{2}$ $[CH_4]_3$	14±	—	·079	12·65	1·2515	·800	·99 / ·721	·21 / ·166	—	—
Water Gas, $[CO]_{16}$ $[CH_4]_3$ $[Sundry]_{7\cdot5}$ $(CO_2)_{16}N2\frac{1}{2}$		8±	—	·045	22·5	·726	1·40	3·878	·788	—	—
Coal Gas, &c., 16 H_4 $[CH_4]_{37}[CO]_{15}$ N_4		4·7	—	·032	31·6	·516	1·975	13·89	2·81	6·16	1·23
Natural Gas $[CH_4]_{96}$ N_4 [Sundry]		8	—	·045	22·5	·725	1·40	15·00	3·06	—	—

Nominal temp. of combustion. Air. F.°	Air. C.°	Oxygen. F.°	Oxygen. C.°	Heat generated by combustion of one Lb. B.Th.U.	Cub. ft. B.Th.U.	Gram. Cal.	Litre. Cal.	Molecule. Cal.	Heat of formation at 15° per Molecule. Cal.
—	—	—	—	—	—	—	—	—	—
376	2090	8278	4590	9666	514·6	5·371	4·087	91·3	12·2 gas 16·6 liq.
2402	1317	6880	3804	3915	—	2·175	—	26·1	·50[1]
4786	2718	17780	9870	14146	—	7·859	—	94·31	3·34[2]
4812	2659	17875	9931	14222	—	7·901	—	94·81	2·84[3] 50[2]
2673	1485	7725	4293	4415	—	2·453	—	29·44	− 2·84[1]
4988	2753	18440	10226	14647	—	8·137	—	97·65	− 3·34[2]
6955	3846	25752	14290	20461	1410	11·367	12·193	136·41	−38·76[1] −42·1 [2]
—	—	—	—	—	—	—	—	—	68·20[4] 94·31[1] 97·65[2]
3494	1923	12892	7144	4383	342	2·436	3·047	68·2	26·1[1] 29·4[2]
6590	3661	24271	13484	9344	1985	5·197	17·66	394·5	−25·4 gas −19 liq.
—	—	—	—	—	—	—	—	—	—
—	(Water Vapour)			52290	293	29·15	2·612	58·3 gas 69 liq. 70·4 sol.	—
4812	2674	12108	6727	62100	347	34·50	3·091		—
—	—	—	—		Water Liquid.			—	—
—	—	—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—	Solid = 70·4 Liq. = 69·0 Gas = 58·3	per H_2
—	—	—	—	—	—	—	—	—	50·6
—	—	—	—	—	—	—	—	—	37·1
6120	3400	20340	11300	21856	1624	12·142	14·46	315·7	−58·1
5022	2790	16830	9350	18094 17930	3764	10·052 9·960	33·496	784·1 gas 776·9 liq.	− 1·8 sol. − 4·1 liq. −11·3 gas
5400	3000	16886	9381	21927	1744	12·182	15·250	341·1	−14·6
4354	2419	14848	8249	22338	1912	12·410	16·641	372·3	23·3
4036	2245	14348	7971	24017	1073	13·343	9·547	213·5	18·9
5144	2858	13179	7321	5490	1511·8	3·050	13·52	302·0	34·4 gas 41 liq.
4630	2573	12583	6690	12744	1639·1	7·080	14·54	325·7	59·8 gas 69·9 liq.
4186	2325	10216	5675	9596	856·5	5·331	7·627	170·6	53·3 gas 61·7 liq.
6099	3388	18222	10125	9086	1230·6	5·048	12·02	262·5	−73·9 gas −68·5 liq.
4000	2222	8078	4488	7770	—	4·317	—	397·2	161·7 liq. 165·6 sol.
2160	1200	4500	2500	1223 to 1237	96·7 97·8	·700	·900	—	—
3440 2160	1910 1200	4500	2500	1256 to 2530	100 to 200	·773 to 1·370	·9674 to 1·713	—	—
4850	2700	—	—	4230 to 5458	330 to 700	2·35 to 3·03	3·00 to 6·35	—	—
4500	2500	—	—	21400	685	11·9	6·099	—	—
4200	2333	—	—	24444	1100	13·58	10·0	—	—

[1] From Graphite. [2] From Amorphous Carbon. [3] From Diamond. [4] From Carbon Monoxide.

Name.	Critical T.° and Pressure (Atmospheres). Lower limit. T.° C. Absol.	Lower limit. Atmos.	Upper limit. T.° C.	Upper limit. Atmos.	Temp. Boil. C.°	Temp. Fusion. C.°	Density of liquid. Water = 1.
Air	133	39	−192·2	—	−192·2	—	·933
Ammonia	403	115	—	—	− 38·5	− 75	·636
Carbon C. Diamond	—	—	—	—	3600	—	—
Carbon C. Graphite	—	—	—	—	"	—	—
Carbon C. Amorphous	—	—	—	—	"	—	—
Carbon C. Vapour	—	—	—	—	"	—	—
Carbon dioxide	306	77	—	—	− 78·2	−115	·83
Carbon monoxide	137·5	35·5	—	—	−190	−207	—
Carbon bisulphide	546	77·9	277·55	95·86	46·2	−116	1·298
Helium	5	2·3	—	—	−269	—	·13
Hydrogen	39	20	—	—	−243	—	·0714
Oxygen	154·2	50·8	—	—	−181·4	−211+	1·124
Nitrogen	127	35	—	—	−194·4	−203	·855
Steam	—	—	370	195·5	100	0	1·00
Steam at 2,000°	—	—	—	—	—	—	—
Steam at 4,000°	—	—	—	—	—	—	—
Acetylene	310	68	—	—	−85	− 81	·451
Benzene	569·7	60·5	—	—	80·8	4·5	·899
Ethylene	283	51	—	—	−103	−169	—
Ethane	307	50·2	—	—	− 93	−151+	—
Methane	200	56·8	—	—	−164	−185·5	·415
Ethylene chloride	—	—	283	—	84·9	—	1·28
Ethyl Alcohol	235·39	66·78	—	—	78·3	− 90+	·809
Methyl Alcohol	202·76	72·85	233	69·7	66·8	—	·814
Cyanogen	7·2	3·6	124	61·7	− 20·7	− 40	·87
Glycerine	—	—	—	—	290·4	17	1·267

VISCOSITY OF GASES.

Sutherland's Formula.

For all gases,
$$\eta_\theta = \eta_{273} \left[\frac{273 + C}{\theta + C}\right] \left(\frac{\theta}{273}\right)^{3/2}$$

Where θ = absolute temperature (°C.).

η_θ = dynamic (not kinematic) viscosity.

c = a constant.

For air, η at 23° C. = 1830×10^{-7} C.G.S. units.

$c = 117$

Temperature coefficient of $\eta = 4\cdot9 \times 10^{-7}$ C.G.S. units per °C.

Kinematic viscosity $= \dfrac{\text{dynamic viscosity}}{\text{density}}$.

Specific heat. Water = 1. Liquid.	Constant Press.	Constant Vol.	Molecular. Constant Press.	Molecular. Constant Vol.	Elementary sp. heat at high temp.	Temp. of igni-tion.	Heat of solution.	a ☆
—	·2375	·1686	—	—	—	— —	—	·0010
Sol.	·508	·391	{9·64 gas 15·1 liq.}	7·6	8·51 + ·0053T	— —	4·4	·0071
{·1468 ·285}	—	—	6·85 at	200°	{11·3 at 1000° 14·2 „ 2000°}	— —	—	—
·2018	—	—	7·12 at	200°	18·8 „ 3600°	— —	—	—
·2415	—	—	4·6 at	100°	—	— —	—	—
—	·285	—	6·85	—	{8·42 + ·00114 T at 1000° to 3600°}	— —	—	—
—	·216	·171	{8·4 + ·0106T}	6·4 + ·0106T	{19·1 + ·0030 (T − 2000°) = 25·1 at 4000°}	— —	5·6	·0081
—	·245	·173	6·8	4·8	4·8 + ·0032(T − 1600°)	— —	—	·0010
·20884	·157	·130	{21·7 gas 18·2 liq.}	—	{10 + ·0746 T 80° to 230°}	— —	—	—
—	2·45	1·74	9·80	6·9	—	— —	—	·0010
—	3·410	2·4146	6·82	4·8	{4·8 + ·0032 (T − 1600°) = 14·1 at 4500°}	—	—	·0010
—	·217	·1548	6·95	4·95	4·95 + ·00324(T − 1600°)	— —	—	·0010
—	·244	·173	6·83	4·8	4·8 + ·0032(T − 1600°)	— —	—	·0010
{1·0 liq. ·504 solid}	·479	·370	8·65	6·65	16·2 + ·038(T − 2000°)	— —	—	—
—	—	·9	—	16·2	—	— —	—	—
—	—	1·32	—	23·8	-	— —	—	—
—	·373	—	9·7	—	—	580 —	5·3	—
·42602	·3754	·350	{34 liq. 29·2 gas (116° to 218°)}	—	17·45 + ·0798T	— —	—	·0510
—	·404	·332	{9·42 at 0° 14·0 at 200°}	—	9·42 + ·0231T	— —	—	·0137
—	—	—	—	—	—	616—	—	—
—	·593	·468	{9·5 100° to 200°}	—	—	667—	—	—
—	·23	{liq. = 28·9 + ·0044T gas = 22·7 at 111° to 211°}		—	—	— —	—	—
{·60 liq. ·50 gas}	·451	·320	{51·2 liq. 20·8° (110° to 220°)}		—	— —	2·5	—
{·66 liq. ·46 gas}	—	—	{21·4 liq. 14·7 gas (110° to 225°)}		—	— —	2·0	—
—	—	—	—	—	—	— —	{6·8 gas 1·4 liq.}	—
—	—	—	—	—	—	— —	5·4	—

[NOTE.—Gases expand by heat to the extent of $\frac{1}{273}$ of their bulk at 0° C. for each degree Centigrade, or $\frac{1}{491 \cdot 4}$ of their bulk at 32° F. for each degree Fahr.

☆ The specific heat of gases varies with the temperature, being more for higher temperatures. At the absolute zero the values of the molecular heat of all gases seem to converge at 6·5 for constant pressure values. Lechatelier therefore gives a formula for specific heat $Cp = 6 \cdot 5 + aT$, where T is the absolute temperature Centigrade and a is a coefficient greater according to the complexity of the molecule. For values of a see table. This has an important bearing on the theory of the gas engine.]

[T = Temperature Centigrade.]

Combustion of Carbon.

Carbon exists only as a solid and is found in three forms or allotropic modifications: (*a*) the diamond; (*b*) graphite; (*c*) amorphous, as charcoal. The heat generated by the perfect combustion of each of these forms is peculiar to that form. Carbon exists in liquid form only in combination with other substances and does not vaporise except at 3600° C. The total heat of combustion of carbon is 11·646 calories per gramme from the state (*a*). Of this amount ·2783 calories disappear in converting the diamond to the amorphous state (*c*); 3·231 calories are absorbed in vaporising the carbon; and 2·4533 calories are set free by its conversion into monoxide CO. 5·684 calories are then produced by the second oxidation of this CO into dioxide = CO_2. In practice the only available heat is that which is set free after vaporising the carbon, *i.e.* 8·137 calories. The following table gives these characteristics in brief, of converting 1 gramme of carbon.

			Calories.
1.	Converting diamond to graphite	=	·041
2.	„ graphite to amorphous	=	·237
3.	„ diamond to amorphous (1) + (2)	=	·278
4.	„ amorphous to monoxide CO	=	2·453
5.	„ monoxide to dioxide	=	5·684
6.	Vaporising amorphous carbon (5) − (4)	=	3·231
7.	Total energy generated converting carbon vapour to dioxide (1) + (2) + (4) + (5) + (6)	=	11·646
8.	Total available heat in practice (7) − (3) − (6)	=	8·137

The difference between (7) and (8) represents expenditure in unlocking the crystal form of the diamond (3) and in converting the carbon to gas (5). The latent heat of vaporisation of carbon is thus 3·231 or about half that of steam.

Hydrogen is known in three states, liquid, solid, and gaseous. The liquid and solid states are those of low temperature. All practical determinations start from the gaseous state. The following figures are found in the combustion of 1 gramme of hydrogen:

		Calories generated.
1.	Converting hydrogen to water gas, H_2O	29·15
2.	„ „ „ liquid	34·50
3.	„ „ „ solid	35·2

The ordinary available heat is that of (1) only; (2) being only partially secured when the products of combustion are cooled considerably below 100° C.

Stated per pound, carbon yields 14,500 British Thermal Units, and hydrogen 52,000 to 62,100 according to the final temperature. The first oxidation of 1 lb of carbon to monoxide yields barely 4,500 heat units, the further oxidation of the resulting carbonic oxide yielding fully 10,000 units.

The difference of the two oxidations is due to the rendering latent of about half the heat of the first oxidation in gasifying the solid carbon. The following equivalent units are useful:

1 Calorie per square metre	=	0·369 B.Th.U. per square foot.
1 B.Th.U. per square foot	=	2·713 calories per square metre.
1 Calorie per kilogram	=	1·800 B.Th.U. per pound.
1 B.Th.U. per pound	=	0·556 calorie per kilogramme.

These calories are kilogramme calories (p. 55).

For thermo-chemical tables of gases see pp. 56-57.

The temperature produced by combustion is largely dependent upon the weight of the resultant product. Thus 1 lb. of hydrogen when burned to water vapour will produce 52,380 heat units, which have to be divided among the 9 lbs. of water. The nominal temperature is 14,375° F., which, however, is based on the suppositious constancy of the specific heat; 1 lb. of carbon, however, which only generates about 14,500 heat units, produces only 3⅔ lbs. of gas, and the nominal resulting temperature is 18·457. These are for combustion in oxygen. With air the additional weight of nitrogen still further reduces the nominal temperatures to about 5,000° in each case.

The percentage of each constituent of a mixed product must be multiplied by the specific heat in order to find the true valency of that product in the calculation of temperature. To find the mean specific heat of a mixture of gases, the weight of each gas is to be multiplied by its specific heat. The products are to be added together and divided by the sum of the weights of all the gases. The result is the mean specific heat. Then the temperature of combustion is found by dividing the total heat of combustion of the combustible gases by the weight of products and by the mean specific heat.

Similarly the calorific value of a mixed gas is found by multiplying the weight of each gas by its calorific value per pound, adding the products together and dividing by the total weight. The result is the thermal value per pound of the mixed gas.

Specific Heat.

Definition.—The amount of heat required per unit mass to cause unit rise of temperature over a small temperature range.

Determination of Specific Heat.

M = weight of body. T = temperature of body. C = specific heat of body.
m = weight of cold water. t = temperature of cold water.
θ = temperature of water after body has been plunged into it.

$$C = \frac{m(\theta - t)}{M(T - \theta)}$$

Specific Heat of Substances at Ordinary Temperatures.

Element.	Specific Heat of Equal Weights.	Element.	Specific Heat of Equal Weights.	Element.	Specific Heat of Equal Weights.
Diamond	0·1468	Cast-iron (white)	0·1298	Palladium	0·0523
Graphite	0·2018	,, (grey)	0·1216	Rhodium	0·0580
Wood charcoal	0·165	Cast-steel (hard)	0·1185	Osmium	0·0306
Coke	0·2000	,, (soft)	0·1166	Iridium	0·0326
Average coal	0·2412	Mild steel	0·1158	Iodine	0·0541
Slag	0·2035	Wrought-iron	0·1146	Bromine (solid)	0·0843
Petroleum	0·511	Nickel	0·1086	,, (liquid)	0·1060
Silicon (fused)	0·1750	Cobalt	0·1070	Potassium	0·1696
,, (crystallised)	0·1767	Manganese	0·1217	Sodium	0·2934
Boron (crystallised)	0·307	Tin	0·0562	Lithium	0·9408
Firebrick	0·2000	Tungsten	0·0334	Phosphorus	0·1887
Sulphur (native)	0·1776	Molybdenum	0·0065	Arsenic	0·0814
,, (purified)	0·2026	Copper	0·0965	Antimony	0·0508
Selenium	0·077	Gunmetal	0·0952	Bismuth	0·0294
Tellurium	0·0474	Brass	0·0940	Thallium	0·0326
Magnesium	0·2499	Lead	0·0314	Silver	0·0558
Zinc	0·0956	Mercury (solid)	0·0319	Gold	0·0312
Cadmium	0·0567	,, (liquid	0·0333	Water	1·0000
Aluminium	0·2143	Platinum	0·0324	Ice (0°C)	0·492
Asbestos	0·20	Ebonite	0·33	Glass (crown)	0·16
Rubber	0·48	Wood	0·42		

Specific Heat of Gases at High Temperatures.

Calories per Gramme per Degree Centigrade, at a constant pressure of one atmosphere.

(*Based on Mallard and Le Chatelier's results.*)

Temperature. F.°	Temperature. C.°	CO_2.	CO & N.	O.	H.
200	93·3	0·198	0·241	0·211	3·37
400	204	0·210	0·245	0·214	3·41
600	316	0·222	0·247	0·216	3·45
800	427	0·235	0·250	0·218	3·49
1,000	538	0·247	0·252	0·220	3·53
1,200	649	0·260	0·255	0·223	3·57
1,400	760	0·272	0·257	0·225	3·60
1,600	871	0·285	0·260	0·227	3·64
1,800	981	0·297	0·261	0·230	3·67
2,000	1,093	0·310	0·265	0·231	3·70

(*G. H. Gill.*)

Latent Heat.

Definition.—The heat absorbed per unit mass in change of state from solid to liquid, or from liquid to gaseous.

Latent heat of fusion of ice at 0° C. = 80 gramme calories.
Latent heat of evaporation of water at 100° C. = 540 gramme calories.

Equal weights of ice and water at 0° C. and 100° C. respectively, have a temperature after mixing of 10·6° C.

Thermal Expansion.

DILATATION OR EXPANSION OF SUBSTANCES PER DEGREE OF FAHRENHEIT SCALE.

Temperatures.	Solids.	Linear. (*a.*)	Surface. (*b.*)	Volume. (*c.*)
32° to 212°	Glass	0·00000478	0·00000956	0·00001434
212 ,, 392	Glass	0·00000546	0·00001093	0·00001639
392 ,, 572	Glass	0·00000660	0·00001320	0·00001980
32 ,, 212	Wrought iron	0·00000656	0·00001312	0·00001968
212 ,, 572	Wrought iron	0·00000895	0·00001790	0·00002686
32 ,, 212	Soft, good iron	0·00000680	0·00001360	0·00002040
32 ,, 212	Cast iron	0·00000618	0·00001236	0·00001854
32 ,, 212	,, steel	0·00000600	0·00001200	0·00001800
32 ,, 212	Hardened steel	0·00000689	0·00001378	0·00002067
32 ,, 212	Copper	0·00000955	0·00001910	0·00002865
212 ,, 572	Copper	0·00001092	0·00002184	0·00003276
32 ,, 212	Lead	0·00001580	0·00003160	0·00004740
32 ,, 212	Gold, pure	0·00000815	0·00001630	0·00002445
32 ,, 212	,, hammered	0·00000830	0·00001660	0·00002490
32 ,, 212	Silver, pure	0·00001060	0·00002120	0·00003180
32 ,, 212	,, hammered	0·00001116	0·00002232	0·00003348
32 ,, 212	Brass, common cast	0·00001043	0·00002086	0·00003129
32 ,, 212	,, wire or sheet	0·00001075	0·00002150	0·00003225
32 ,, 212	Platinum, pure	0·00000491	0·00000982	0·00001473
212 ,, 572	Platinum, pure	0·00000520	0·00001040	0·00001560
32 ,, 212	Palladium	0·0000062	0·0000124	0·0000186
32 ,, 212	Roman cement	0·00000797	0·00001594	0·00002391
32 ,, 212	Platinum, hammered	0·00000530	0·00001060	0·00001590
32 ,, 212	Zinc, pure or cast	0·00001633	0·00003266	0·00004899
32 ,, 212	,, hammered	0·00001722	0·00003444	0·00005166
32 ,, 212	Tin, cast	0·00001207	0·00002414	0·00003621
32 ,, 212	,, hammered	0·00001500	0·00003000	0·00004500
32 ,, 212	Concrete	0·000006	0·000012	0·000018
32 ,, 212	Fire-brick	0·00000275	0·00000470	0·00000705
32 ,, 212	Good red brick	0·00000305	0·00000610	0·00000915
32 ,, 212	Marble	0·00000613	0·00001226	0·00001839
32 ,, 212	Granite	0·00000438	0·00000876	0·00001314
32 ,, 212	Bismuth	0·00000773	0·00001546	0·00002319
32 ,, 212	Antimony	0·00000602	0·00001204	0·00001806
32 ,, 212	Mercury	—	—	0·00010000
212 ,, 392	Mercury	—	—	0·00010250
392 ,, 572	Mercury	—	—	0·00010500
32 ,, 212	Water	—	—	0·00026420
212 ,, 392	Water	—	—	0·00051020
392 ,, 572	Water	—	—	0·00056713
32 ,, 212	Salt, 26 per cent. solution	—	—	0·000242
32 ,, 212	Sulphuric acid, 100 per cent.	—	—	0·00033333
32 ,, 212	Turpentine and ether	—	—	0·00038900
32 ,, 212	Oil, common	—	—	0·00044444
32 ,, 212	Alcohol and nitric acid	—	—	0·00055555
32 ,, 212	All permanent gases	—	—	0·00203250
32 ,, 212	Aluminium	0·0000142	0·0000284	0·0000426
32 ,, 212	Ebonite	0·000039	0·000078	0·000117
32 ,, 212	Rubber	0·000055	0·00011	0·000165

The superficial expansion is twice the linear, and the cubical expansion three times the linear. The coefficient of expansion of an alloy is equal to the mean coefficients of expansion of the volumes of the metals composing it.

If a bar is *increased* in length from l to L by an *increase* of temperature of $t°$, then

$$L = l(1 + at°).$$

If a bar is *decreased* in length from L to l by a *decrease* of temperature $t°$, then

$$l = \frac{L}{1 + at°}$$

Thermal Expansion of Air.

Dry air expands or contracts uniformly 0·002039 its volume per degree F. difference under *constant pressure*.

If the *volume is constant* and the pressure varies, then the coefficient per degree F. is 0·002036.

Thermal Conductivity.

Definition.—The quantity of heat per unit area conducted per second through a slab of the substance of unit thickness, per degree of temperature difference.

1 gram calorie per sec. per sq. cm. per degree C. per cm. = 0·0056 pound calories per sec. per sq. inch per degree C. per inch.

Values of Thermal Conductivity.

(Gram calories per sec. per cm.^{2} per degree C. per cm.)

Substance.	Conductivity.	Substance.	Conductivity.
Metals and Alloys (at about 100 ° C.)			
Aluminium	0·49	Magnesium	0·38
Antimony	0·04	Nickel	0·14
Bismuth	0·016	Platinum	0·17
Cadmium	0·22	Silver	0·99
Copper	0·91	Tin	0·15
Gold	0·70	Zinc	0·26
Iron, wrought	0·15	Duralumin	0·36
,, cast	0·11	Brass (70–30)	0·33
Steel (1 per cent. C.)	0·11	Bronze	0·18
Lead	0·08	German silver	0·09
Mercury	0·02	Manganin	0·06
Miscellaneous Solids and Liquids (at room temperatures).			
Glass, crown	0·0025	Porcelain	0·0025
,, flint	0·002	Quartz	0·023
,, soda	0·0016	Rubber	0·00045
Asbestos sheet	0·0006	Sawdust	0·00012
Brick (Fire)	0·0003	Silk	0·0001
Cork	0·0001	Slagwool	0·00010
Ebonite	0·0004	Timber (dry)	0·0005
Felt	0·0001	Alcohol	0·0004
Gas Carbon	0·010	Glycerine	0·0007
Ice	0·005	Transformer oil	0·0003
Mica	0·0018	Turpentine	0·0003
Wax, paraffin	0·0006	Water	0·0015
Paper	0·0003	Concrete	0·004
Gases (at room temperatures).			
Air	0·00006	Hydrogen	0·00035
Ammonia	0·00005	Nitrogen	0·00005
CO_2	0·00003	Oxygen	0·00006
CO	0·00005	SO_2	0·00002
Helium	0·00035	Water vapour	0·00004

Effect of Temperature on Thermal Conductivity.

For pure metals, conductivity falls with rise of temperature.
,, alloys, ,, rises ,, ,, ,,
,, liquids, ,, ,, ,, ,, ,,
,, gases ,, ,, ,, ,, ,,

OPTICS.

Decomposition of Light in the Spectrum.

Colours.	Maximum Ray.	Combination of Colours.		
		Primary.	*Secondary.*	*Tertiary.*
Violet	Chemical	Blue, Yellow	Green	Dark Green
Indigo		Blue, Red	Purple	Brown
Blue	Electrical	Yellow, Red	Orange	
Green				
Yellow	Light			
Orange				
Red	Heat			

All the colours of the spectrum mixed together make white.

Velocity of Light.

Velocity of light = $3 \cdot 004 \times 10^{10}$ cms. per sec. Mean wave length = $5 \cdot 3 \times 10^{-5}$ cms.

The velocity of light through the atmosphere is 186,330 miles per second (*Newcomb*). The velocity of light through transparent bodies is not known, but probably varies inversely as the square root of the specific gravity of the transparent substance.

Light passes from the sun to the earth ($1 \cdot 487 \times 10^{13}$ cms. = 92,390,000 miles [*Everett*]) in eight minutes, at which velocity light can pass around the earth in one-eighth of a second.

To find the Focal Length of a Convex Mirror.

Use as object an opaque screen, with a hole and pin-point, and painted white, or covered with white paper.

Set up on the bench, in line, say, with the left edge of the hole, the convex mirror and an auxiliary biconvex lens of short focal length (6 ins. or so), and adjust the lens so that the image of the hole and pin-point is formed side by side with the object. The centre of the mirror is now at the point at which the image would be formed by the lens alone; this position may either be calculated or found (after noting the position of the mirror and then removing it) by means of a screen. Thus the radius is easily measured.

If the focal length of the mirror be greater than f, that of the lens, the simplest way of adjusting is to put the lens as close as possible to the mirror, put the object at principal focus of lens, and move the object back until the image is formed as above.

If, however, the focal length be less, we can be sure of finding the position by putting the mirror at a distance of $4f$ from the object, and the lens at $2f$ and moving the lens back until the desired position is reached.

To find the Focal Length of a Concave Lens.

Use an object like the one mentioned above, an auxiliary convex lens (say, 6 ins. focal length) to produce a convergent beam, and an auxiliary plane mirror, placed beyond the concave lens.

Adjust until the image is formed side by side with the object, as before; then the rays must be emerging parallel to one another from the concave lens, and hence the convergent beam from the convex lens will (when the concave lens and mirror are removed) form an image at the principal focus of the concave lens. A direct measure can thus be made of the focal length.

(*E. Budden.*)

Relation between Positions of Object and Image.

Distance of object from lens or mirror = u.
" " image " " " " = v.
Focal length of lens or mirror = f.
Radius of curvature of mirror = r
∴ $2f$.

Distances measured on the same side of the lens or mirror as the object are taken as positive and those on the opposite side as negative.

Spherical mirror.— $$\frac{1}{v} + \frac{1}{u} = \frac{2}{r}$$

(For concave mirror, r is positive. For convex mirror, r is negative.)

Lens.— $$\frac{1}{v} = \frac{1}{u} + \frac{1}{f}$$

(For concave or divergent lens, f is positive. For convex or convergent lens, f is negative.)

Photographic Lens Apertures.

Stops or apertures are openings in the lens diaphragms which determine the amount of light admitted to the sensitive plate in a camera. They are usually expressed as fractions of the lens focus, *i.e.* with an 8-in. focus lens a stop or aperture 1 in. in diameter would be known as *f*8—*viz.* it would divide into the focal length eight times; *f*16 would be ½-in. diameter, and so on.

The exposure required varies inversely as the area of the opening. The apertures mostly in use are *f*5·6, *f*8, *f*11, *f*16, *f*22, *f*32, *f*45 and *f*64, and each will be half the area and require twice the exposure of the next larger aperture. *E.g.*, if *f*11 requires 1 second *f*16 would require 2 seconds and *f*8 only half a second.

If the diameter of any one aperture be known the remainder can be ascertained by the method shown in fig. 1.

Say the diameter of *f*11 is ¾ in., set off *f*11 as one side of a right-angled triangle ABC; then BD, at right angles to the hypotenuse, would give the diameter of *f*16, DE would give *f*22 FA the diameter of *f*8, and so on as indicated.

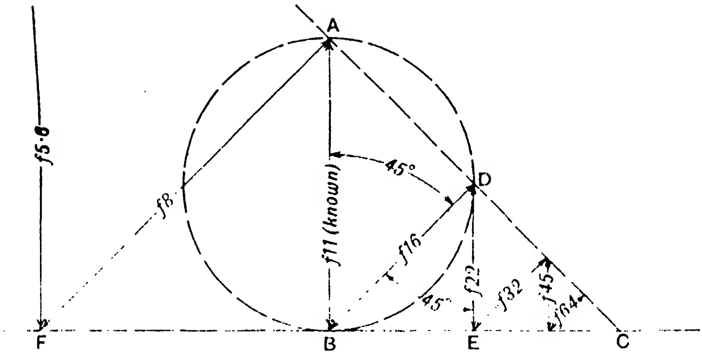


FIG. 1.

Theoretically *f*8 in one lens should be of the same exposure value as *f*8 in any other lens of whatever make or focus, but loss due to refraction, etc., must vary according to the number of component parts, and a compound lens containing many surfaces would obviously be slightly slower than one of the simplest form.

SECTION IV

ACOUSTICS, VIBRATION AND NOISE (Pp. 69-81)

(By C. W. Glover, M.I.C.E., M.I.Struct.E., Member Acoustical Society.)

SECTION IV

ACOUSTICS, VIBRATION AND NOISE.

(By C. W. Glover, M.I.C.E., M.I.Strupt.E., Member Acoustical Society.)

SCOPE.

The scope of the subject is set out diagrammatically below :-

The Science of Sound.

- Atmospheric acoustics.
  - Listening equipment.
    - National defence.
      - Design of high-speed machinery.
  - Navigation by sound.
    - Sound ranging.
      - Subaqueous sound ranging and signalling.
  - Mirage and weather.
- Physiological acoustics.
  - Speech.
  - Hearing.
    - Aids to the deaf.
      - Sound measurement.
        - Sound recording.
        - Sound reproducing.
        - Filtration of sound.
          - Design of equipment.
- Architectural acoustics.
  - Absorption and dissipation of noise.
    - Exterior noise.
    - Interior noise.
      - Provision of proper acoustica conditions in auditoria.
        - Problems demanding acoustical study in design.
        - Acoustical correction of auditoria after construction.
  - Sound Insulation.
    - Contact noises
      - Structural vibration.
    - Air borne sound.

SOUND.

Sound is a form of energy which finds its origin in the vibration of matter.

For the propagation of sound energy matter is necessary.

Sound is propagated radially outwards from the source on spherical wave fronts with a velocity which depends upon the elastic properties and density of the vibrating medium.

Velocities of Sound at 0° C.

Air	1,090 ft. per second.	*Note.*—The velocity is independent of both the frequency and amplitude of vibration.
Coal gas	1,600 " " "	
Water	4,700 " " "	
Pinewood	10,900 " " "	
Brick	11,980 " " "	
Iron	16,000 " " "	
Steel	16 360 " " "	

Sound Waves.

Sound waves are of the compression or longitudinal type. The distance from 'peak to 'peak' of pressures is called the *wave-length* and this depends upon the *frequency* or the number of complete waves passing in one second.

Limits of Audibility.

The slowest vibration producing audible sound is 20 cycles per second, the corresponding wave-length being 56 ft.

The highest audible frequency is 20,000 cycles per second (10 octaves higher), corresponding to a wave-length of 0·056 ft.

The maximum particle displacement from the normal is called the amplitude and this is only 0·004 ins. for the loudest sound and 0·00000005 ins. for a sound barely audible.

The velocity can be calculated from the formula $V = \sqrt{\frac{E}{p}}$

where E is the modulus of elasticity of the medium
in solids E is Young's Modulus;
in liquids E is the bulk modulus;
in gases E is the pressure modulus.
the adiabatic elasticity must be used and

$$V = \sqrt{\frac{1 \cdot 4\, p_1}{p}}$$ where p_1 is the pressure.

is the density of the medium.

If the temperature of the medium is raised, in air the velocity becomes $V = 332 + 0 \cdot 6t$ metres/sec., where t is the temperature in degree C.

There are three principal characteristics of audible sound : pitch, loudness and tone quality.

Pitch.

The pitch of a sound is the frequency of vibration measured as the number of complete vibrations passing in one second. It is independent of the amplitude.

Loudness.

The loudness of sound is described as a comparative statement of the strength of the sensation received through the ear.

Its measurement in physical values is not simple, loudness being largely a physiological effect which depends upon the ear. It corresponds to brightness in optics and depends upon the amount of energy in the sound waves entering the ear.

The amount of energy in ergs which passes in one second through an area of 1 sq. cm. is called the intensity of the sound.

The loudness of a sound varies as the square of the amplitude and inversely as the square of the distance from the source. This latter does not apply to sound in a closed space like an auditorium or to sound propagated from a source of dimensions comparable with its distance.

The ear perceives generally on a logarithmic basis, but loudness sensation also varies with the pitch. Fechner's law states that $S = c \log_{10} p + a$;

where p is the excess pressure (dynes per sq. cm. and c and a are parameters depending upon the frequency.

This law is considerably in error for loud sounds. The fundamental relationship between the loudness of a sound and the acoustic energy producing it is still much in dispute, but as a fundamental basis for the measurement of loudness, *aural comparison with a standard scale* is adopted internationally.

The equivalent loudness of a sound is then measured in 'phons' by the intensity of a standard reference and pure tone in d.b. considered by the observer to be as loud as the sound being assessed.

Units.

A 'bel' is the decimal logarithm of the ratio of two powers.

Two powers *differ* by $\log_{10} \frac{W_1}{W_2}$ bels, or a 'bel' is a 10-fold increase in power or energy.

A 'decibel' is a tenth of a 'bel' and equals a transmission unit.

The number of T.U.'s or d.b. = 10 $\log_{10}$ ratio of powers in watts,
or 20 $\log_{10}$ ratio of voltages,
or 20 $\log_{10}$ ratio of current intensity.

Twenty times the change in photographic density in sound films equals the change in electric power in d.b.

The ultimate standard of reference is an acoustical pressure of 0·0002 dynes per sq. cm. for a 1,000 cycles pure note. This is the adopted arbitrary zero for the d.b. scale at 1,000 cycles

THE DECIBEL SCALE.

Ratio of Sound Power to that at Threshold.	Decibels above Threshold.	Sound pressure at 1,000 ∼ in dynes per sq. cm.	Ratio of Sound Pressure to that at Threshold.
Threshold of feeling, 10^{13}	130	$2 \times 10^{2.5}$	$10^{6.5}$
" " " 10^{12}	120	2×10^{2}	10^{6}
" " " 10^{11}	110	2×10^{1}	$10^{5.5}$
" " " 10^{10}	100	2×10^{1}	10^{5}
" " " 10^{9}	90	2×10	$10^{4.5}$
" " " 10^{8}	80	$2 \times 10 = 2$	10^{4}
" " " 10^{7}	70	$2 \times 10^{-0.5}$	$10^{3.5}$
" " " 10^{6}	60	2×10^{-1}	10^{3}
" " " 10^{5}	50	$2 \times 10^{-1.5}$	$10^{2.5}$
" " " 10^{4}	40	2×10^{-2}	10^{2}
" " " 10^{3}	30	$2 \times 10^{-2.5}$	$10^{1.5}$
" " " 10^{2}	20	2×10^{-3}	10^{1}
" " " 10^{1}	10	$2 \times 10^{-3.5}$	$10^{0.5}$
Threshold of audibility, $10^{0} = 1$	0	$2 \times 10^{-4} = 0{\cdot}0002$	$10^{0} = 1$

The loudness of a tone is given in 'phons.'

An *equivalent loudness* of '*n* phons' is equal to that of the standard tone when operating at '*n* decibels' above its zero.

APPROXIMATE LOUDNESS SCALE OF 'PHONS.

130 phons	Threshold of feeling.
120 "	Heavy gun-fire.
110 "	Near aeroplane engine.
100 "	Near express train.
90 "	Pneumatic drill at 4 ft.
80 "	In tube train (windows open).
70 "	In express overland train (windows open).
60 "	Loud conversation.
50 "	Average conversation.
40 "	Suburban house.
30 "	Very quiet room.
20 "	Whispering.
10 "	Rustling of leaves in quiet garden.
0 "	Threshold of audibility.

Measurement of Loudness is effected by the use of—

(*a*) *Objective noise meters*, in which the sound received by a calibrated microphone is converted into electric energy, amplified and rectified and measured on a micro-ammeter.

(*b*) *Subjective noise meters*, in which the sound to be measured is matched with a pure reference tone produced by the instrument or alternatively masked by the latter. The degree of adjustment required affords the means of measurement.

Churcher suggests a *scale of loudness* such that the magnitude of a sound in phons is given by the d.b. above threshold pressure of 0·0002 dynes per sq. cm. of the tone of 1,000 cycles per sec., which, heard in free space with both ears, sounds as loud as the noise when the two are heard alternately.

$$\text{Loudness} = (\text{phons})^3 \times 10^{-3}.$$

Tone Quality.

The tone quality of a sound is that characteristic which distinguishes it from another sound of the same loudness and pitch. Nearly all sounds are composite and can be analysed into their separate component tones called partials. Those partials which have the lowest frequency are called fundamentals, the others being the overtones. Those overtones, having a frequency which is an exact multiple of that of the fundamental, are called harmonics.

The higher frequencies are essential to intelligibility in speech and to brilliance in music, whilst the lower frequencies contribute timbre or volume.

Reflection of Sound.

The laws of sound generally conform with those of light, but owing to the greatly increased wave-length of sound diffraction results. The higher frequencies are more directional in effect and the lower are not so subject to sound shadows.

Reflection and inter-reflection of sound within an auditorium introduces reverberation, which is desirable except when excessive.

Optimum reverberation times for auditoriums of the volumes tabulated are given below.

When the sound is reproduced or amplified, deduct 15 per cent. from the times given.

Volume of Auditorium. Cub. Ft.	Optimum Reverberation in Seconds with Auditorium Two-thirds Full. (512 Cycles.)			
	Music and Speech.	Choral Music.	Music.	Speech.
50,000	1·20	1·68	1·38	1·00
100,000	1·45	2·00	1·67	1·23
200,000	1·80	2·50	2·10	1·55
300,000	2·00	2·80	2·30	1·70
400,000	2·10	2·94	2·40	1·80
500,000	2·20	3·10	2·50	1·87
600,000	2·30	3·20	2·65	1·95
700,000	2·35	3·30	2·70	2·00
800,000	2·40	3·36	2·76	2·05
900,000	2·45	3·44	2·84	2·10
1,000,000	2·50	3·50	2·88	2·13

Special auditoria require reverberation times as below :—

Radio Studios.—0 to 10 secs. to suit the broadcast. Studios of various sizes, others with adjustable absorbents and others with echo rooms attached, provide the scope required.

Sound Film Studios.—About half the period given in the ' speech ' column above.

Gramophone Studios.—Speech *0·5 secs.* graduated to bands at *4 secs.*

The optimum reverberation time is not the same at all frequencies but having ascertained optimum at 512 ∼ as above, that at the other frequencies can be calculated by using the appropriate multiplier tabulated below.

Frequency in Cycles per Sec.	Multiplier Speech.	Multiplier Music.	Multiplier Speech and Music.
64	1·75	2·30	2·03
128	1·38	1·55	1·47
256	1·15	1·20	1·18
512	1·00	1·00	1·00
1,024	1·00	1·00	1·00
2,048	1·00	1·00	1·00
4,096	1·14	1·04	1·09
8,192	1·35	1·15	1·25
16,384	1·60	1·35	1·47

Reverberation varies directly as the volume and inversely as the acoustic absorption present.

When the average coefficient of acoustic absorption is less than 0·2, the W. C. Sabine formula may be used.

$$t = \frac{V}{20\,(s_1a_1 + s_2a_2 + s_3a_3 \ldots)}$$

where t is the time in seconds.

V is the volume in cubic feet.

a_1, a_2, a_3, etc., are the respective sound absorbing coefficients of the various materials having surfaces s_1, s_2, s_3, etc.

For ' dead ' conditions the Eyring formula is more correct.

$$= \frac{V}{20\,s_T\,(-\log_e(1-a))}$$

where s_T is the total interior surface and a is the total absorption $\div\ s_T$.

When a standard strength of source is used rather than an initial loudness, the P. E. Sabine formula is the more applicable.

$$= \frac{0{\cdot}0083\,V}{A}\,(9{\cdot}1 - \log_{10} A$$

where A is the total absorption present.

Absorption.

The absorption coefficients at 512 ∼ are now available for most materials and range from marble at 0·01 to fibre boards at 0·35 and the open window at 1·00.

The coefficient is not the same at all frequencies.

Draped and porous materials absorbing principally in the upper register.
Resilient materials like carpet—in the middle register.
Stretched membranes and resonating panels in the lower register.

Optimum absorption at all frequencies can only be secured by a judicious selection of acoustic materials.

It is desirable to dispose reflecting surfaces about the origin and to incline them in such a way that the acoustic images of the source of sound are as near as possible to the origin. Absorbent surfaces should be remote from the source.

The general requirements for good acoustics in buildings may be briefly stated as below :—

(1 Uniform distribution of sound throughout the room without distortion

(2) Proper reverberation time.

(3) Adequate loudness of sound throughout the room.

(4) Absence of disturbing extraneous noise.

A summary of acoustical defects, their causes, recommendations for their avoidance in design and the remedy to be applied in existing buildings, is given below.

SUMMARY OF ACOUSTICAL DEFECTS.

Defect.	Causes.	Avoidance in Design.	Remedy for Existing Buildings.
Excessive Reverberation.	A. Excessive volume.	Keep volume down to 100–150 cub. ft. per seat.	Increase seating, reduce volume or increase interior surface by bold decoration.
	B. Insufficient absorption.	Add absorbents.	Add absorbents.
Echoes	A. Unsuitable shape.	Avoid curves in design.	Alter shape or use absorbents on offending surfaces.
	B. Distant Reflecting surfaces.	Make distant surfaces highly absorbent.	Ditto.
Sound Foci	Concave reflecting interior surfaces.	Avoid curvilinear interiors.	Alter shape or use absorbents on focussing areas.
Dead Spots	A. Positions denuded by sound foci.	Avoid curvilinear interiors.	Alter shape, make focussing area absorbent, use parabolic sounding boards, or directional speakers.
	B. Screens.	Reflect sound into screened areas.	
Interference	Undiffused reflections of sustained note.	Design bold breaks in walls and ceilings. Use resonating wall covering.	Add bold decorative and/ or resonating interior eatment.
Insufficient Sound Volume.	A. Too voluminous a building.	Use orchestra of appriate size.	
	B. Lack of reflection close to origin.	Use electrical amplification. Dispose hard reflecting surfaces about origin.	
	C. Excessive absorption.	Adjust absorption to give optimum reverberation.	
Distortion	A. Selective absorption.	Use of collection of absorbents to obtain desired reverberation characteristic.	
	B. Screening of higher frequencies.	Avoid obstructions.	Reflect higher frequencies into sound shadows.
	C. Uncontrolled resonance.	Select board absorbents which resonate over wide range and fix on battens at irregular intervals.	
Resonance	Flimsy partitions and linings	Adopt rigid construction with studs, etc., at irregular spacings.	
Masking Noise	Insufficient sound insulation, bad fitting doors and windows or badly designed vents.	Select construction having least overall insulation value of 50 d.b.	

Sound Insulation.

Air-borne sounds may be transmitted to an adjoining room in three ways :—

(a) By direct passage of air waves through openings in the wall, through ventilators, etc.
(b) By vibration of the dividing medium itself as a diaphragm which by its motion re-creates sound waves on the farther side ; and
(c) By direct transmission of elastic-wave motion through the dividing medium which imparts to the air at the farther side a wave motion exactly timed to that of the origin.

Air-borne sounds are most effectively stopped by rigid walls that do not have openings for pipes or ventilators, though the construction of the wall itself, apart from the question of rigidity, has an important bearing upon the degree of insulation.

For the transmitted sound to be a minimum—*i.e.* for the greatest sound insulation—the sound energy dissipated in other ways must be a maximum.

(a) The reflected sound will be large in proportion to the incident sound if the reflecting surface is ' hard.'
(b) The sound absorbed by conversion to heat energy will be increased by using ' laminated ' construction with the inner layer very absorbent of sound.
(c) The sound energy absorbed as mechanical energy will be increased by using heavy ' massive ' construction.
(d) The diaphragm-like vibration can be minimised by the adoption of ' rigid ' construction.

Rigidity.

A partition may be bulged by pressure of sound, and if the periodicity of the sound waves corresponds with the natural period of vibration of the partition, the transmitted sound will be ' increased ' in loudness by resonance.

Measurement of Sound Insulation.

There is no simple formula for the calculation of sound insulation of composite construction, but the insulation of homogeneous partitions is proportional to the log. of the weight per square foot.

The insulaton in d.b. $= 14 \cdot 3 \log_{10}$ wt. per sq. ft. $+ 22 \cdot 7$ d.b. and the following values are of interest :—

Mass per sq. ft. of wall area.	Insulation in d.b. at 512 ~
1 lb. wt.	22·7
5 "	32
10 "	37
20 "	41
40 "	45
60 "	48
100 "	51·3
400 "	60

In porous-flexible materials the sound insulation in d.b. is proportional to the thickness of the material.

Thus a 10-fold increase in the thickness of a flexible porous material like hair felt will bring about a 10-fold increase in the sound insulation, whereas a 10-fold increase in the thickness of a rigid non-porous partition, say from 10 to 100 lb. wt. per sq. ft., will only bring about an increase of insulation from 37 to 51·3 d.b., *i.e.* less than a 2-fold increase.

Note that to provide an insulation of 60 d.b. by means of a solid non-porous wall, a mass of 400 lb. per sq. ft. would be necessary—say a 40-in. brick wall.

Owing to the slow rate of increase of insulation with weight of partition, it is generally uneconomic in practice to provide for insulation greater than 45 d.b. solely by mass and rigidity.

In panels of homogeneous nature, mass and rigidity are the determining factors, but when the construction departs from homogeneity, structure becomes of predominating importance.

Tests on construction of various kinds appear to show quite definitely that in practice refraction effects may be ignored, and that it is necessary only to consider the diaphragm-like vibrations.

An increase in stiffness and/or mass will therefore improve the sound insulation of a wall.

Cross connections are very detrimental to the sound-insulating value of lath and plaster or similar partitions. The one side should be constructed entirely independent of the other by the use of staggered studding.

An air space is generally better than a filling material, unless the filling material is quilted (to prevent its settlement and tight packing) and fixed to one side only.

CONCLUSIONS.

Undoubtedly the important points for the prevention of sound transmission in walls and partitions are :—

(a) A hard reflecting surface on the outside of the wall.
(b) A non-homogeneous structure containing inert air cells.
(c) An air gap to prevent continuity.
(d) A layer of insulating material.
(e) A sound-absorbent surface facing the other room.
(f) A massive and rigid construction.

It is quite certain that the important points for the prevention of sound transmission through floors are :—

(1) A floating floor isolated from the walls.
(2) An isolating material of non-homogeneous nature between the floor covering and the floor proper.
(3) The floor proper (of massive and rigid construction) ; and
(4) A suspended ceiling.

For the adequate insulation of structure borne sound or 'contact noise' in floors, the 'floating' surface must have sufficient inertia.

To isolate the noise of footfalls the 'floating' surface should weigh not less than 15 lb. per sq. ft.

INSULATION AGAINST VIBRATION

Structural Vibrations.

The full effect of sound transmission through the members of a structure are but imperfectly realised. Vibration in one part of the structure is conveyed easily through the continuity of solid material and by setting up resonant vibrations recreates sound waves in a position often very remote from the original source of sound.

If the design of the structure precludes any possibility of breaking continuity of structural contact, very special endeavours must be made to prevent linings of all rooms coming in structural contact with the framework of the buildings.

An effective method of control lies in the interposition of an elastic discontinuity in the path of the vibrations.

Sound Insulation in a Steel-framed Building.

Isolation of structural vibration in the various members of a steel-framed building may be effected by the use of lead and asbestos cloth sheets.

Note that in isolated connections, bolts sheathed in asbestos and bitumen must be used in place of rivets which would provide structural contact.

The use of isolator pads on every principal joint in a steel-framed building would reduce the general rigidity of the whole and so permit greater freedom in vibration.

It will usually be sufficient to isolate only the member shaken in the case of localised vibration, but to use pads under all foundations when earth vibrations have to be dealt with.

Sound Insulation of Reinforced Concrete Buildings.

The use of resilient pads under the bearing of pre-cast reinforced concrete units of a structure offers a solution in this type of construction, and the reinforced concrete floors in brick buildings can be similarly dealt with.

On the other hand, *in situ* constructed monolithic reinforced concrete buildings, from the mere nature of their design, are admirably suited to the effective transmission of vibration, and loaded reinforced concrete floors may act as very effective sounding boards in a framed reinforced concrete structure.

The effects of mass and rigidity have, however, a damping influence and it is not easy to determine in any particular design just the degree of discontinuity of structure, effecting as it does the overall rigidity, which would be productive of the greatest resistance to the transmission of noise and vibration.

The concensus of opinion inclines to the view that structures of reinforced concrete give less cause for complaint than those of steel-framed construction.

There is no doubt that, for the minimisation of vibration and noise in reinforced concrete buildings, the following points need careful attention :—

(1) Foundations should be built upon a resilient layer having the requisite elastic strength and 'compliance,' and particular care should be taken to prevent the arrangement becoming waterlogged, as water transmits vibration with the greatest facility.
(2) The various members of the structure should be isolated from each other by resilient pads of lead and asbestos, where such an arrangement would not result in serious loss in structural rigidity.
(3) The interior surface of all rooms should be carefully isolated from the structural elements.

Machinery Vibration.

In rotating parts eccentricities give rise to vibrations at frequencies equal to the speed of rotation, and at the 'critical' speed the frequency of the vibration corresponds to the natural period of the machine on its support—a coincidence which is marked by an alarming increase in the amplitude of vibration.

There are as many 'critical speeds' as there are modes of vibration.

Quiet machinery is usually secured by close attention to the following points

(1) The vibromotive forces should be kept down to the minimum by
- (a) carefully balancing reciprocating parts,
- (b) keeping reciprocating parts as light as possible,
- (c) dynamically balancing all rotating elements to eliminate unbalanced centrifugal forces and whipping effects, and/or
- (d) by the provision of a dynamic absorber tuned to neutralise the unbalanced vibration.

(2) Heavy, rigid frames should be provided.

(3) Loose and worn parts should be replaced.

(4) Smooth-meshed gearing and quiet, well-lubricated bearings should be used.

Human Susceptibility.

Susceptibility to vibration is dependent upon the velocity of motion, the amplitude of vibration, the acceleration of movement, or the total impulse of the vibrating mass.

The intensity of vibration is the product of the amplitude in m.m. and the frequency in cycles per second.

An intensity of one or more constitutes a nuisance.

In practice it is best to ascertain the dominant vibration of a machine at normal operating speeds and to design the mounting accordingly, since haphazard installation may aggravate the vibration.

Isolators.

The most important properties of good isolators are elasticity and deflection, and the degree of success in isolation of vibration will depend upon the manner in which the relative magnitudes of these properties are adjusted.

It is usually found that a reduction in area of supporting pads will result in an improvement in isolation of vibration.

The Design of Damping Devices.

If a machine of mass m be supported on a resilient pad, the natural frequency of the mass m on its elastic supports must be low compared with the frequency of the vibration to be isolated, otherwise the pad may be worthless.

The amount of vibration transmitted to the floor or foundation is determined by the elastic or viscous properties of the pad, and the 'transmissibility' of the support is given by the ratio of the vibratory force communicated with and without the isolating pad.

$$\frac{\text{Transmitted vibration with pad}}{\text{Transmitted vibration without pad}} = \tau = \sqrt{\frac{r^2 + \dfrac{1}{4\pi^2 n^2 c^2}}{r^2 + \left(2\pi nm - \dfrac{1}{2\pi nc}\right)^2}}$$

where c is the 'compliance' of the pad (*i.e.* the reciprocal of the force constant or of the modulus of elasticity).

(Rubber has low elasticity but considerable compliance. Steel has high elasticity but little compliance.)

r is the internal mechanical resistance owing to the viscous forces within the pad, and

n is the frequency of the vibration to be isolated.

The compliance c is calculated by observing the elastic deflection of a specimen loaded in compression for each additional unit of compressive force.

The compliance c for any elastic material will be directly proportional to the thickness and inversely proportional to the area of cross-section.

The internal resistance r can be ascertained from the 'decay' curve by observing the successive amplitudes of the free vibrations of a known mass on a measured specimen of the resilient material, or by measurements of the rate of return of the displaced mass.

The internal resistance is directly proportional to the area of cross-section of the flexible support and inversely proportional to its thickness.

The resistance factor r is not so important as the compliance factor, and in approximate calculations may be regarded as zero.

The relation between the transmissibility τ and the ratio of the frequency of the forced vibration to that of the natural vibration of the machine on its elastic support is given in the table below:

Ratio of Forced to Natural Frequencies.	Transmissibility.	Remarks.
0	1·0	..
0·5	2·2	Amplification
1·0	3·1	„
1·5	1·7	„
2·0	0·8	Damping
2·5	0·3	„
3·0	0·1	„
3·5	0·05	„
4·0	0·005	„

Examination of these figures will show that when the ratio of forced to natural vibrations is 1, *i.e.* when the natural frequency of the machine on its elastic support corresponds with that of the vibration to be isolated, a condition of resonance obtains and the vibration submitted will be three times that of the machine direct upon the floor without an isolator.

It will also be seen that when the ratio is 3 or more, there will be very little of the forced vibration transmitted to the floor or foundation.

In general, the elastic support should be sufficiently compliant and the mass supported should be sufficiently heavy to make the natural frequency of the arrangement low in comparison with the vibration to be isolated.

The isolator should have a relatively large compliance, should be loaded to its safe maximum within the elastic limit, and should have as large an internal resistance as is consistent with the above conditions.

The elastic support transmits all frequencies below about twice the natural frequency of the machine on its support, but isolates all frequencies above about

$$\frac{1}{\pi\sqrt{mc}}$$

In the design of spring damping devices the following data will be useful.

Coiled Springs.

D = mean diameter in inches.
d = diameter of round material or side of square material.
H = free height as made and unloaded.
h = height when pressed down solid.
R = range of deflection (H—h).
W = Load in tons (test load for deflection R).
N = Number of free coils.
δ = deflection per ton in inches.

Round Sections.

$$R = \frac{N.D^2}{36d}$$

$$\delta = \frac{N.D^3}{700d^4}$$

$$W = \frac{20d^3}{D}$$

Square Sections

$$R = \frac{N.D^2}{48d.}$$

$$\delta = \frac{N.D^3}{1000d^4}$$

$$W = \frac{21d^3}{D}$$

The natural frequency of a loaded spring is a function of the deflection per ton ( ).

Natural frequency $f_0 = \frac{1}{2\pi}\sqrt{\frac{S}{m}}$ where S = Spring factor (Force in absolute units causing unit deflection of spring).
and m = mass in absolute units.

$$\text{or } f_0 = 3 \cdot 13 \sqrt{\frac{\text{Load per spring}}{\text{Load per spring} \times \text{deflection.}}}$$

$$\text{Deflection under load} = \frac{10}{\text{Square of Frequency required}} \text{ (approx.).}$$

Building Vibrations.

These take place in the three planes due to machinery, traffic and other causes.

The frequencies are between 1 and 10 per sec., the full amplitudes being from 0·0014" to 0·00004". Vibrations at audio frequency may be induced in wall panels.

A vibrating telephone membrane produces audible sound when vibrating with full amplitudes between 0·000006" and 0·04".

NOISE.

Noise is a ubiquitous accessory of the age and may be defined as any unwanted sound.

Noise is usually of very complex nature, the acoustic spectrum being devoid of harmonic relationship.

Sounds are unpleasant on account of their pitch, their persistence and/or their loudness.

High-pitched tones are more annoying than low-pitched tones of the same equivalent loudness.

The effect of masking noise on the intelligibility of speech is shown in the table below.

Background of Noise in Phons.	Percentage Syllable Articulation.	Remarks.
0 to 15	95 to 85	Good hearing.
15 ,, 35	85 ,, 65	Satisfactory with attentive listening.
35 ,, 70	65 ,, 25	Unsatisfactory hearing conditions.

TRAFFIC NOISE.

A comprehensive noise survey of London shows daylight noise levels to be from 60 to 85 d.b.; day averages being as high as 78 d.b. in some places and the night average only 11 d.b. lower.

Isolated peaks, averaging 7 d.b. high, occur at the rate of 2 to 8 per minute.

Very roughly, up to 60 vehicles per second, twice the number of vehicles passing per minute equals the noise level in d.b.

Between 60 and 130 vehicles per minute the increase is only some 25 d.b.

If the number of commercial vehicles increase the traffic noise increases in proportion.

For a given total number of vehicles, if the percentage of commercial vehicles to total vehicles increase from zero to 100 per cent., the traffic noise increases proportionately by an amount up to 20 d.b. maximum.

The construction of lofty buildings, by the introduction of reverberation within street spaces, tends to maintain the sound level until the height of the surrounding buildings is passed.

Confined sites are more noisy than open ones, and tramway crossings and traffic starting places are the most noisy.

The greatest energy in average traffic noise occurs between 600 and 2,000 ~, though the higher frequencies (up to 15,000 ~) are the most annoying.

NOISE ABATEMENT.

Suppression of noise at its source is always more efficacious than abatement by acoustical methods.

In an enclosed space, increasing the absorption reduces the noise.

$$\text{Noise reduction in d.b.} = 10 \log_{10} \frac{a_2}{a_1}$$

where $\frac{a_2}{a_1}$ is the absorption ratio; doubling the absorption only reduces the noise 3 d.b. Ten times the absorption would be required for a 10 d.b. drop in equivalent loudness, though this would usually represent an annoyance reduction of over 60 per cent.

Absorption of the higher frequencies renders the noise more acceptable.

The effect of the absorption is greater the more remote from the source.

The absorption has a localising effect and reduces the 'fear reaction.'

The use of sound absorbing ceilings in noise abatement in buildings can be shown in extreme cases to effect a 5 per cent. increase in efficiency and to pay for itself in 6 months.

TRAIN NOISES.

Train noises are due to wheel rumble and joint impacts and increase largely with speed. Between 60 and 80 m.p.h. the noise level in a carriage with windows open is 74 d.b., and in the corridor 84 d.b. Closing the windows reduces the noise some 5 d.b. In tunnels the noise level increases by about 10 d.b.

The use of rubber treads on the wheels, offers the most effective remedy.

In the London tubes the noise levels average 87 d.b., with peaks up to 100 d.b. when running at 30 m.p.h. At points and crossings these values are up 7 d.b.

Acoustic treatment in the lower sections of the tube reduces the noise level some 8 d.b. The use of rubber pads under chairs on the sleepers and well-packed ballast, has quietened the track to some extent, though reverberation of air-borne sound seems to be the principal difficulty.

For the principles of insulation for the vehicles, see notes on aeroplane noise.

Aeroplane Noise.

Aircraft noise originates in engine clatter, exhausts, air-screw noise and 'flutter' of parts of the aircraft caused by air-speed eddies.

The frequency range is from 100 to 2,000 cycles per sec., the minimum energy occurring on a band between 200 and 800 cycles per sec.

About a third of the total noise is attributable to engine clatter, one-third to the airscrew and the balance to exhausts and 'flutter.'

Aircraft.

The modern all-steel monoplane is more rigid and produces less noise than the older type of biplane, with its multifarious struts and ties, which, despite careful streamlining, set up 'flutter.'

The engine nacelle fitted directly into the wing structure of a monoplane having tractor propellers, with the thrust axis in line with the centre line of the wing, gives the least interference, the greatest propulsive efficiency and the least noise from flutter.

Engine exhausts should preferably be located over the wings and over the cabin. If engines are kept well forward the noise level in the cabins from this source may be reduced by from 5 to 10 d.b.

Silencers.

The general principles involve the provision of sound absorbing baffles and a capacity effect for removing the explosive pulsation of the exhaust gases without introducing excessive back pressures.

Silencers of the long perforated tube type may be relied upon for no more than a 10 d.b. reduction in loudness level, but a special type of silencer, weighing 75 lbs., reduces the exhaust noise of a 500-h.p. engine by 25 d.b., the back pressure at full throttle not exceeding 2½ lbs. per sq. inch.

Propellers.

For some time air-screws were thought to be the principal source of aircraft noise, especially as high-speed thick propellers produce such a loud and irritating note.

Tests carried out by the National Physical Laboratory in conjunction with the Royal Aircraft Establishment show that air-screw noise is generally consistent with the following rough rules :—

Noise reductions in propeller design.
10 d.b. per 100 ft./sec. reduction in tip speed.
1 d.b. per degree decrease in pitch setting.
10 d.b. for change from thick to thin section.
5 d.b. per foot increase in diameter.
10 d.b. for change from 2 blades to 4 blades of the same diameter.

Capon reports that in a cruising flight at a distance of 15 ft. from a two-bladed propeller, the noise level was approximately $20 + 0 \cdot 1$ V. d.b., where V was the tip speed in feet per sec.

It is desirable to keep the tip speed down to 600 ft. per sec., reducing the noise level to 80 d.b. from this source.

Cabin Insulation in Aircraft.

(a) *Isolation of Structure-borne Noise in Aeroplanes.*

The noise vibrations arise from engine torque recoil or from flutter.

The former can only be dealt with by the mounting of the power unit in vibration damping devices and the latter by the provision of the most rigid design of structure.

The interior of the cabin can, however, be isolated from the structure if the lining is fixed to it through cork or sorbo-rubber pads or gaskets.

(b) *Insulation against Air-borne Sound.*

The cabin is a box in an atmosphere of noise and unless the interior is absorbent of sound the noise level inside will eventually build up to that outside.

In fact, equilibrium is reached only when the rate of sound admission into the cabin is equalled by the rate of sound absorption inside.

For the best results, therefore, the cabin walls must be *insulating* and the interior *absorbent*.

The exterior intensity × area × t = the interior intensity × area × a;

where t is the coefficient of transmission and a is the coefficient of absorption, whence

$$10\log_{10}\frac{\text{absorption}}{\text{transmittance}} = \text{d.b. reduction.}$$

By the adoption of laminated construction :—

(Corrugated duralumin exterior skin on 3-in. duralumin channel framing, loose filling of kapok. Inner duralumin lining in sorbo rubber packing on frame. Lined with fireproofed upholstering fabric on $\frac{3}{32}$-in. cork sheets.)

On a 12-seater plane, an all-in reduction of 25 d.b. can be obtained at 37 lb. weight of treatment per passenger.

This allows for transmittance through windows and doors.

Lighter treatment, not exceeding 10 lb. per passenger, results in an insulation of some 12 d.b.

Exhaust Silencers for Internal Combustion Engines.

The impulses in the exhausts from internal combustion engines may have a frequency of from 10 to 200 per second.

The noise created may cover the entire audible range, the lower frequency-band being attributable to the exhaust impulses and the upper to the vortex effects and the vibration of metallic parts.

The pressure in an exhaust system may rise momentarily to 50 lb. per sq. in., thus causing expansions in the casings of silencers timed to the impulses and causing low frequency sounds and their concomitant overtones.

Silencing is best affected by the use of a sound absorbent for the higher frequencies coupled with a capacity type silencer or expansion chamber with perforated baffle for damping out the low frequency components of the noise.

Motor cycle engine silencers using perforated tubes, surrounded with glass silk, through which the exhaust gases pass to expansion chambers before discharge to the atmosphere, have been found to reduce the noise some 20 d.b. without impairment of engine performance.

In large stationary installations the ' capacity ' exhaust silencer can conveniently be provided by a large pit filled with rubble, through which the exhaust gases are discharged after passing through the ordinary engine silencers.

A low velocity of discharge to atmosphere is desirable if air pulsations are to be removed.

Silencing Exhausts from Ventilating Fans.

In exhausting a noisy atmosphere into a quiet zone it is desirable to keep the velocity of discharge from the fans down to 1,000 ft. per min. and to use an expansion chamber arranged to discharge to atmosphere through weathered louvres at a velocity not exceeding 300 ft. per minute.

The expansion chamber should be packed with sound absorbing tubes (of, say, Seapak on perforated zinc) carefully aligned to the flow of air so as to prevent air turbulence and to divide the flow into passages such that the mean dimension of the passage is not greater than one-tenth the mean travel of the air through it.

Noise Due to Electrical Machinery.

In transformers the laminations are attracted to each other twice each electrical cycle and therefore give a note of double the supply frequency.

A 100 k.v.a. outdoor distribution transformer, designed for quiet operation, gave the following results :—

Frequency. Cycles per Second.	Loudness Value. Phons.
100	18
200	7
300	6
600	14

Note the comparatively high equivalent loudness of the upper harmonics.

The cases must be stiffened to prevent amplification of the noise by vibration.

Analysis of the frequency components of noise from electrical machinery enable the causes to be traced with certainty.

B. G. Churcher quotes test results on an induction motor as below :—

Frequency of Component.	Frequency. Revs. per Sec.	Corresponding to
225	9	9 blades on fan.
675	27	Third harmonic of fan note.
1,120	45	45 rotor slots.
100	---	Vibration of stator punchings at twice supply frequency.

See also Descriptive Section IV.
Fibreglass, Ltd.

SECTION V

PROPERTIES OF ENGINEERING MATERIALS

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

PART I

DIMENSION STANDARDS (pp. 85–118)

PART II

WEIGHT OF MATERIALS (pp. 119–157)

PART III

STRENGTH OF MATERIALS (pp. 159–183)

SECTION V

PART I

PROPERTIES OF ENGINEERING MATERIALS.

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

DIMENSION STANDARDS.

(a) WIRE AND SHEET METAL GAUGES.

IMPERIAL STANDARD WIRE GAUGE * (S.W.G.)
(Legalised by the Board of Trade, March 1, 1884.)

No.	Diameters. Mils.†	Differences.	Millimetres.	No.	Diameters. Mils.	Differences.	Millimetres.	No.	Diameters. Mils.	Differences.	Millimetres.
0,000,000	500	..	12·70	13	92	12	2·34	32	10·8	·8	·274
000,000	464	36	11·78	14	80	12	2·03	33	10·0	·8	·254
00,000	432	32	10·97	15	72	8	1·83	34	9·2	·8	·234
0,000	400	32	10·16	16	64	8	1·63	35	8·4	·8	·213
000	372	28	9·45	17	56	8	1·42	36	7·6	·8	·193
00	348	24	8·84	18	48	8	1·22	37	6·8	·8	·173
0	324	24	8·23	19	40	8	1·016	38	6·0	·8	·152
1	300	24	7·62	20	36	4	·914	39	5·2	·8	·132
2	276	24	7·01	21	32	4	·813	40	4·8	·4	·122
3	252	24	6·40	22	28	4	·711	41	4·4	·4	·112
4	232	20	5·89	23	24	4	·610	42	4·0	·4	·102
5	212	20	5·38	24	22	2	·559	43	3·6	·4	·0914
6	192	20	4·88	25	20	2	·508	44	3·2	·4	·0813
7	176	16	4·47	26	18	2	·457	45	2·8	·4	·0711
8	160	16	4·06	27	16·4	1·6	·417	46	2·4	·4	·0610
9	144	16	3·66	28	14·8	1·6	·376	47	2·0	·4	·0508
10	128	16	3·25	29	13·6	1·2	·345	48	1·6	·4	·0406
11	116	12	2·95	30	12·4	1·2	·315	49	1·2	·4	·0305
12	104	12	2·64	31	11·6	·8	·295	50	1·0	·2	·0254

* This gauge is the only legal standard wire-gauge for the United Kingdom.
† 1 Mil. = $\frac{1}{1000}$th of an inch.

BIRMINGHAM WIRE GAUGE ‡ (B.W.G.)

Number.	Dimensions. Mils.	Number.	Dimensions. Mils.	Number.	Dimensions. Mils.	Number.	Dimensions. Mils.
0000	454	7	180	17	58	27	16
000	425	8	165	18	49	28	14
00	380	9	148	19	42	29	13
0	340	10	134	20	35	30	12
1	300	11	120	21	32	31	10
2	284	12	109	22	28	32	9
3	259	13	95	23	25	33	8
4	238	14	83	24	22	34	7
5	220	15	72	25	20	35	5
6	203	16	65	26	18	36	4

‡ The Birmingham Wire Gauge differs from the Birmingham Gauge (see next page).

(See also pages 88 and 132.)

Birmingham Wire Gauge for Silver and Gold.

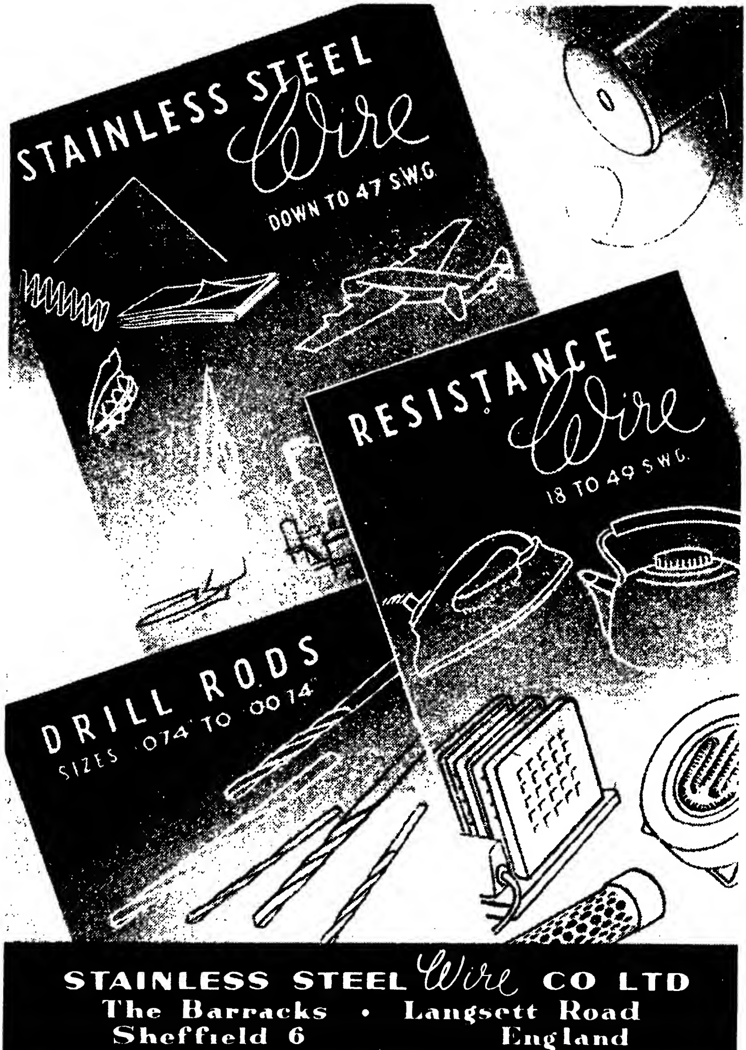
No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.
1	·004	6	·013	11	·029	16	·051	21	·072	25	·095	29	·120	33	·145
2	·005	7	·015	12	·034	17	·057	22	·074	26	·103	30	·126	34	·148
3	·008	8	·016	13	·036	18	·061	23	·077	27	·113	31	·133	35	·158
4	·010	9	·019	14	·041	19	·064	24	·082	28	·120	32	·143	36	·167
5	·013	10	·024	15	·047	20	·067								

Sheet Zinc Trade Gauge.

No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.
1	·00395	5	·0097	9	·0177	12	·0260	15	·0375	18	·0526	21	·0699	24	·0915
2	·00554	6	·0114	10	·0196	13	·0292	16	·0426	19	·0577	22	·0768	25	·0980
3	·0067	7	·0132	11	·0228	14	·0323	17	·0478	20	·0632	23	·0843	26	·1052
4	·0082	8	·0149												

Birmingham Gauge ('B.G.') for Sheets and Hoops.
(Legalised by the Board of Trade, Nov. 1, 1914.)

Descriptive Number.	Equivalents in parts of an Inch.	Descriptive Number.	Equivalents in parts of an Inch.
No.	Inch.	No.	Inch.
15/0 B.G.	1·000	20 B.G.	·0392
14/0 B.G.	·9583	21 B.G.	·0349
13/0 B.G.	·9167	22 B.G.	·03125
12/0 B.G.	·8750	23 B.G.	·02782
11/0 B.G.	·8333	24 B.G.	·02476
10/0 B.G.	·7917	25 B.G.	·02204
9/0 B.G.	·750	26 B.G.	·01961
8/0 B.G.	·7083	27 B.G.	·01745
7/0 B.G.	·6666	28 B.G.	·015625
6/0 B.G.	·625	29 B.G.	·0139
5/0 B.G.	·5833	30 B.G.	·0123
4/0 B.G.	·5416	31 B.G.	·0110
3/0 B.G.	·500	32 B.G.	·0098
2/0 B.G.	·4452	33 B.G.	·0087
1/0 B.G.	·3964	34 B.G.	·0077
1 B.G.	·3532	35 B.G.	·0069
2 B.G.	·3147	36 B.G.	·0061
3 B.G.	·2804	37 B.G.	·0054
4 B.G.	·250	38 B.G.	·0048
5 B.G.	·2225	39 B.G.	·0043
6 B.G.	·1981	40 B.G.	·00386
7 B.G.	·1764	41 B.G.	·00343
8 B.G.	·1570	42 B.G.	·00306
9 B.G.	·1398	43 B.G.	·00272
10 B.G	·1250	44 B.G.	·00242
11 B.G	·1113	45 B.G.	·00215
12 B.G	·0991	46 B.G.	·00192
13 B.G.	·0882	47 B.G.	·00170
14 B.G.	·0785	48 B.G.	·00152
15 B.G	·0699	49 B.G.	·00135
16 B.G.	·0625	50 B.G.	·00120
17 B.G.	·0556	51 B.G.	·00107
18 B.G.	·0495	52 B.G.	·00095
19 B.G.	·0440		


STAINLESS STEEL Wire
DOWN TO 47 S.W.G.
RESISTANCE Wire
18 TO 49 S.W.G.
DRILL RODS
SIZES ·074" TO ·0014"
STAINLESS STEEL Wire CO LTD
The Barracks • Langsett Road
Sheffield 6 England
Telegrams 'FINEWIRE, Sheffield

STUBS' STEEL WIRE GAUGE (DIMENSIONS IN MILS.).

No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.	No.	Mils.
Z	413	Q	332	H	266	1	227	9	194	17	172	25	148	33	112	41	95	49	72	57	42	65	33	73	23
Y	404	P	323	G	261	2	219	10	191	18	168	26	146	34	110	42	92	50	69	58	41	66	32	74	22
X	397	O	316	F	257	3	212	11	188	19	164	27	143	35	108	43	88	51	66	59	40	67	31	75	20
W	386	N	302	E	250	4	207	12	185	20	161	28	139	36	106	44	85	52	63	60	39	68	30	76	18
V	377	M	295	D	246	5	204	13	182	21	157	29	134	37	103	45	81	53	58	61	38	69	29	77	16
U	368	L	290	C	242	6	201	14	180	22	155	30	127	38	101	46	79	54	55	62	37	70	27	78	15
T	358	K	281	B	238	7	199	15	178	23	153	31	120	39	99	47	77	55	50	63	36	71	26	79	14
S	348	J	277	A	234	8	197	16	175	24	151	32	115	40	97	48	75	56	45	64	35	72	24	80	13
R	339	I	272																						

TABLE SHOWING RELATIVE VALUES OF IMPERIAL STANDARD, BIRMINGHAM, AND AMERICAN (BROWN AND SHARPE) WIRE GAUGES.

S.W.G.	B.W.G.	A.W.G.	Equivalent in Mils.	Equivalent in Mms.
7/0			500	12·699
6/0			464	11·785
		0000	460	11·683
	0000		454	11·531
5/0			432	10·972
	000		425	10·794
		000	410	10·413
0000			400	10·159
	00		380	9·651
000			372	9·448
		00	365	9·271
00			348	8·839
	0		340	8·635
		0	325	8·254
0			324	8·229
1	1		300	7·620
		1	289	7·340
	2		284	7·213
2			276	7·010
	3		259	6·578
		2	258	6·642
3			252	6·400
	4		238	6·045
4			232	5·892
		3	229	5·816
	5		220	5·588

S.W.G.	B.W.G.	A.W.G.	Equivalent in Mils.	Equivalent in Mms.
5			212	5·384
		4	204	5·181
6	6		203	5·156
			192	4·876
		5	182	4·622
	7		180	4·571
7			176	4·470
	8		165	4·191
		6	162	4·114
8			160	4·064
	9		148	3·759
9		7	144	3·657
	10		134	3·403
10		8	128	3·251
	11		120	3·047
11			116	2·946
		9	114	2·895
	12		109	2·768
12			104	2·641
		10	102	2·590
	13		095	2·412
13			092	2·336
		11	091	2·311
	14		083	2·108
14		12	081	2·057

S.W.G.	B.W.G.	A.W.G.	Equivalent in Mils.	Equivalent in Mms.
15	15	13	072	1·828
	16		065	1·650
16		14	064	1·625
	17		058	1·472
		15	057	1·447
17			056	1·421
		16	051	1·295
	18		049	1·244
18			048	1·218
		17	045	1·142
	19		042	1·066
19		18	040	1·016
20		19	036	·9140
	20		035	·8886
21	21	20	032	·8124
		21	0284	·7213
22	22		028	·7109
		22	0253	·6126
	23		025	·6347
23			024	·6093
24	24	23	022	·5585
25	25	24	020	·5078
26	26	25	018	·4570
27	27	26	016	·4062

S.W.G.	B.W.G.	A.W.G.	Equivalent in Mils.	Equivalent in Mms.
28	28	27	014	·3555
29	29		013	·3300
		28	0126	·3202
30	30		012	·3046
31		29	011	·2800
32			0108	·2743
33	31	30	010	·2539
34	32	31	009	·2300
35	33	32	008	·2031
36	34	33	007	·1777
37			0068	·1727
38		34	006	·1523
		35	0056	·1422
39	35	36	005	·1269
40			0048	·1219
41		37	0044	·1118
42	36	38	004	·1015
43		39	0035	·0889
44		40	0031	·0788
45			0028	·0713
46			0024	·0610
47			002	·0507
48			0016	·0406
49			0012	·0305
50			001	·0253

Wire Gauges used in the United States.

Dimensions of Sizes in Decimal Parts of an Inch.

Number of Wire Gauge.	American or Brown and Sharpe.	Birmingham or Stubs' Iron Wire.	Washburn and Moen Mfg. Co., Worcester, Mass.	Imperial Wire Gauge.	Stubs' Steel Wire.	U.S. Standard for Plate.	Number of Wire Gauge.
000000	—	—	—	·464	—	·46875	000000
00000	—	—	—	·432	—	·4375	00000
0000	·46	·454	·3938	·400	—	·40625	0000
000	·40964	·425	·3625	·372	—	·375	000
00	·3648	·38	·3310	·348	—	·34375	00
0	·32486	·34	·3065	·324	—	·3125	0
1	·2893	·3	·2830	·300	·227	·28125	1
2	·25763	·284	·2625	·276	·219	·265625	2
3	·22942	·259	·2437	·252	·212	·25	3
4	·20431	·238	·2253	·232	·207	·234375	4
5	·18194	·22	·2070	·212	·204	·21875	5
6	·16202	·203	·1920	·192	·201	·203125	6
7	·14428	·18	·1770	·176	·199	·1875	7
8	·12849	·165	·1620	·160	·197	·171875	8
9	·11443	·148	·1483	·144	·194	·15625	9
10	·10189	·134	·1350	·128	·191	·140625	10
11	·090742	·12	·1205	·116	·188	·125	11
12	·080808	·109	·1055	·104	·185	·109375	12
13	·071961	·095	·0915	·092	·182	·09375	13
14	·064084	·083	·0800	·080	·180	·078125	14
15	·057068	·072	·0720	·072	·178	·0703125	15
16	·05082	·065	·0625	·064	·175	·0625	16
17	·045257	·058	·0540	·056	·172	·05625	17
18	·040303	·049	·0475	·048	·168	·05	18
19	·03589	·042	·0410	·040	·164	·04375	19
20	·031961	·035	·0348	·036	·161	·0375	20
21	·028462	·032	·0317	·032	·157	·034375	21
22	·025347	·028	·0286	·028	·155	·03125	22
23	·022571	·025	·0258	·024	·153	·028125	23
24	·0201	·022	·0230	·022	·151	·025	24
25	·0179	·02	·0204	·020	·148	·021875	25
26	·01594	·018	·0181	·018	·146	·01875	26
27	·014195	·016	·0173	0164	·143	·0171875	27
28	·012641	·014	·0162	0148	·139	·015625	28
29	·011257	·013	·0150	·0136	·134	·0140625	29
30	·010025	·012	·0140	·0124	·127	·0125	30
31	·008928	·01	·0132	·0116	·120	·0109375	31
32	·00795	·009	·0128	·0108	·115	·01015625	32
33	·00708	·008	·0118	·0100	·112	·009375	33
34	·006304	·007	·0104	·0092	·110	·00859375	34
35	·005614	·005	·0095	·0084	·108	·0078125	35
36	·005	·004	·0090	·0076	·106	·00703125	36
37	·004453	—	—	·0068	·103	·006640625	37
38	·003965	—	—	·0060	·101	·00625	38
39	·003531	—	—	·0052	·099	—	39
40	·003144	—	—	·0048	·097	—	40

Care should be taken to distinguish the difference between Stubs' Iron Wire Gauge and Stubs' Steel Wire Gauge. The former is commonly known as the Birmingham Wire Gauge, and designates the Stubs' *Soft* wire sizes.

The Stubs' Steel Wire Gauge is the one employed in measuring drawn steel wire or drill rod, of Stubs' Make, and is also used by some makers of American drill rods.

(*Brown and Sharpe.*)

WARRINGTON WIRE GAUGE.

Mark.	Size.	Mark.	Size.	Mark.	Size.	Mark.	Size.
No.	Inch.	No.	Inch.	No.	Inch.	No.	Inch.
7/0	$\frac{1}{2}$	2	·274	10	·133	16	·0625, or $\frac{1}{16}$
6/0	$\frac{15}{32}$	3	·25, or $\frac{1}{4}$	10½	·125, or $\frac{1}{8}$	17	·053
5/0	$\frac{7}{16}$	4	·229	11	·117	18	·047
4/0	$\frac{13}{32}$	5	·209	12	·10, or $\frac{1}{10}$	19	·041
3/0	$\frac{3}{8}$	6	·191	13	·090	20	·036
2/0	$\frac{11}{32}$	7	·174	14	·079	21	·0315, or $\frac{1}{32}$
0	·326	8	·159	15	·069	22	·028
1	·300	9	·146				

WHITWORTH'S WIRE GAUGE, IN DECIMALS OF AN INCH.

Mark.	Size.	Mark.	Size.	Mark.	Size.	Mark.	Size.	Mark.	Size.
1	·001	14	·014	34	·034	85	·085	240	·240
2	·002	15	·015	36	·036	90	·090	260	·260
3	·003	16	·016	38	·038	95	·095	280	·280
4	·004	17	·017	40	·040	100	·100	300	·300
5	·005	18	·018	45	·045	110	·110	325	·325
6	·006	19	·019	50	·050	120	·120	350	·350
7	·007	20	·020	55	·055	135	·135	375	·375
8	·008	22	·022	60	·060	150	·150	400	·400
9	·009	24	·024	65	·065	165	·165	425	·425
10	·010	26	·026	70	·070	180	·180	450	·450
11	·011	28	·028	75	·075	200	·200	475	·475
12	·012	30	·030	80	·080	220	·220	500	·500
13	·013	32	·032						

(b) SCREW THREADS AND SCREWS.

Forms of Screw Threads.

Figs. 1, 2, 3, and 4 show four usual forms of thread, each of 1 inch pitch.

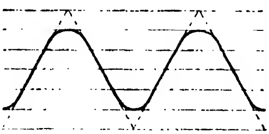


FIG. 1.—Whitworth Thread-form.

Fig. 2.—American Thread-form.

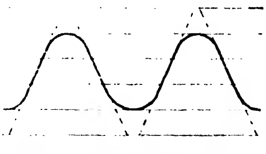


FIG. 3.—Swiss Thread-form.

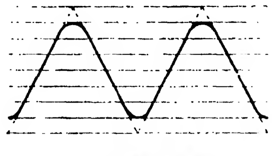


FIG. 4.—Steinlen Thread-form.

In the *Whitworth* or *English Standard* thread (fig. 1) the angle made by the two sides of the thread is 55° for all sizes of thread. One-sixth of the depth of the triangle is rounded off the top and bottom, so that the angle of 55° gives for the actual depth rather more than three-fifths and less than two-thirds of the pitch.

In the *American* (*Sellers'*) thread (fig. 2) the angle made by the two sides of the triangle is 60°, and one-eighth of the depth is cut off the top and bottom of the triangle, leaving a flat one-eighth of the pitch in width.

In the *Swiss* or *Thury* thread (fig. 3) the angle made by the two sides of the thread is $47\frac{1}{2}°$ and one-fifth of the depth is rounded off the top and bottom of the triangle.

For screw-plate Swiss threads the depth of the thread is three-fifths the length of the pitch.

The top of the ridge is formed by a circular arc, the radius of which equals one-sixth of the pitch. The space between two consecutive ridges is formed at the side next the core by a circular arc, the radius of which equals one-fifth of the pitch.

The Swiss thread for screws made in the lathe is the *Steinlen* thread (fig. 4). The angle made by the two sides of the thread is 58° 8′, and the radius of the connecting arcs is ·1011 of the pitch. The generating figure is an isosceles triangle, of which the base and the height are equal to the pitch. One-eighth of the depth is rounded off the top and bottom of the triangle.

The *British Association* thread is similar to that of Thury, except that the top and bottom of the threads are both rounded to the same radius, namely, two-elevenths of the pitch.

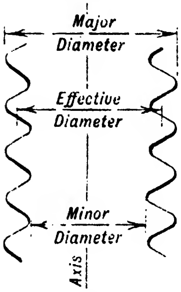


FIG. 5.

BRITISH STANDARD SCREW THREADS.

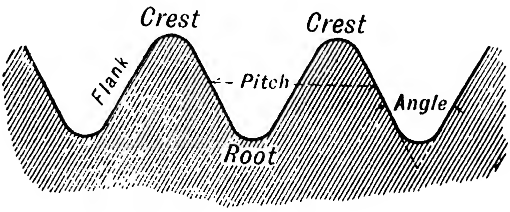


FIG. 6.

DEFINITIONS.*

The Effective Diameter of a parallel thread is the diameter of the imaginary co-axial cylinder which intersects the surface of the thread in such a manner that the intercept on a generator of the cylinder, between the points where it meets the opposite flanks of a thread groove, is equal to half the nominal pitch of the thread (fig. 5).

The *Major Diameter* of a thread is the diameter of the imaginary co-axial cylinder which just touches the crest of an external thread or the root of an internal thread (fig. 5).

The *Minor Diameter* is the diameter of the cylinder which just touches the root of an external thread or the crest of an internal thread (fig. 5).

The *Crest* of a thread is the prominent part of a thread, whether external or internal (fig. 6).

The *Root* is the bottom of the groove between the two flanking surfaces of the thread, whether external or internal (fig. 6).

The *Flanks* of a thread are the straight sides which connect the crest and the root (fig. 6).

The *Angle* of a thread is the angle between the flanks, measured in an axial plane section (fig. 6).

The *Pitch* of a thread is the distance, measured parallel to its axis, between corresponding points on adjacent thread forms, in the same axial plane (fig. 6).

Definition of Whitworth Form of Thread.

The British Standard Whitworth form of thread is a symmetrical V-thread in which the angle between the flanks, measured in the axial plane, is 55°; one-sixth of the sharp vee is truncated at top and bottom, the thread being rounded equally at crests and roots by circular arcs blending tangentially with the flanks, the theoretical depth of thread being thus 0·640327 times the nominal pitch.

The basic thread depths calculated from the above definition are rounded off to the nearest 0·0001 inch.

Tolerances.

British Standard Specification No. 84—1940 gives dimensions for standard Whitworth threads of 'close,' 'medium,' and 'free' fit respectively. The tolerance on effective diameter for 'medium' fit is calculated from the formula:

$$\text{Tolerance in inches} = 0\cdot002\sqrt[3]{D} + 0\cdot003\sqrt{L} + 0\cdot005\sqrt{P}.$$

where D = major diameter of thread,
L = length of engagement,
P = pitch. (All in inches.)

* By permission of the British Standards Institution.

Tolerances for 'close' fit are two-thirds of those for 'medium' fit, and for 'free fit,' 1½ times those for 'medium' fit.

When the above formula is applied to B.S. Whitworth threads from ¼ in. to 3 in. and to B.S. Fine threads from ¼ in. to 1½ in. diameter (taking the length of engagement as equal to the diameter of the thread), it gives tolerances in agreement with the 'standard' effective diameter tolerances in B.S. Specifications 92—1919 and 84—1918 (now superseded by B.S.S. 84—1940).

Note.—In introducing B.S.S. No. 84—1940, it was made clear by the British Standards Institution that caution should be exercised in changing over to the new standard if, during the present emergency, such a change might hamper production in any way. If its adoption is likely to introduce any difficulty in production or in gauging, arrangements can be made, by discussion between the manufacturer and the Government inspector, for production to continue to B.S.S. 84—1918 and 92—1919 or to other authorised specifications operative at the time of the introduction of B.S.S. 84—1940. (See also Amendment Slip, 1942.)

British Standard Whitworth (B.S. Whit.) Screw Threads.*
(B.S.S. No. 84—1940.) (*Abstract.*)

Nominal diameter.	Number of threads per inch.	Pitch.	Depth of thread.	Major diameter.	Effective diameter.	Minor diameter.	Cross sectional area at bottom of thread.
in.		in.	in.	in.	in.	in.	in.
⅛†	40	0·025 00	0·0160	0·1250	0·1090	0·0930	0·0068
3/16	24	0·041 67	0·0267	0·1875	0·1608	0·1341	0·0141
¼	20	0·050 00	0·0320	0·2500	0·2180	0·1860	0·0272
5/16	18	0·055 56	0·0356	0·3125	0·2769	0·2413	0·0457
⅜	16	0·062 50	0·0400	0·3750	0·3350	0·2950	0·0683
7/16	14	0·071 43	0·0457	0·4375	0·3918	0·3461	0·0941
½	12	0·083 33	0·0534	0·5000	0·4466	0·3932	0·1214
9/16	12	0·083 33	0·0534	0·5625	0·5091	0·4557	0·1631
⅝	11	0·090 91	0·0582	0·6250	0·5668	0·5086	0·2032
11/16‡	11	0·090 91	0·0582	0·6875	0·6293	0·5711	0·2562
¾	10	0·100 00	0·0640	0·7500	0·6860	0·6220	0·3039
⅞	9	0·111 11	0·0711	0·8750	0·8039	0·7328	0·4218
1	8	0·125 00	0·0800	1·0000	0·9200	0·8400	0·5542
1⅛	7	0·142 86	0·0915	1·1250	1·0335	0·9420	0·6969
1¼	7	0·142 86	0·0915	1·2500	1·1585	1·0670	0·8942
1½	6	0·166 67	0·1067	1·5000	1·3933	1·2866	1·300
1¾	5	0·200 00	0·1281	1·7500	1·6219	1·4938	1·753
2	4·5	0·222 22	0·1423	2·0000	1·8577	1·7154	2·311
2¼	4	0·250 00	0·1601	2·2500	2·0899	1·9298	2·925
2½	4	0·250 00	0·1601	2·5000	2·3399	2·1798	3·732
2¾	3·5	0·285 71	0·1830	2·7500	2·5670	2·3840	4·464
3	3·5	0·285 71	0·1830	3·0000	2·8170	2·6340	5·449
3¼	3·25	0·307 69	0·1970	3·2500	3·0530	2·8560	6·406
3½	3·25	0·307 69	0·1970	3·5000	3·3030	3·1060	7·577
3¾	3	0·333 33	0·2134	3·7500	3·5366	3·3232	8·674
4	3	0·333 33	0·2134	4·0000	3·7866	3·5732	10·03
4½	2·875	0·347 83	0·2227	4·5000	4·2773	4·0546	12·91
5	2·75	0·363 64	0·2328	5·0000	4·7672	4·5344	16·15
5½	2·625	0·380 95	0·2439	5·5000	5·2561	5·0122	19·73
6	2·5	0·400 00	0·2561	6·0000	5·7439	5·4878	23·65

* By permission of the British Standards Institution.

† Dimensionally the ⅛-in. × 40 t.p.i. thread belongs more appropriately to the B.S. Fine series, but it has for so long been associated with the Whitworth series that it is now included herein.

‡ To be dispensed with wherever possible.

WHITWORTH'S STANDARD SCREWS FOR WATCH AND INSTRUMENT MAKERS.

No. denoting thousandths of an Inch in Diameter.	Threads per Inch.	No. denoting Thousandths of an Inch in Diameter.	Threads per Inch.	No. denoting Thousandths of an Inch in Diameter.	Threads per Inch.	No. denoting Thousandths of an Inch in Diameter.	Threads per Inch.
10	400	18 & 19	250	34	150	65	80
11	400	20	210	36	150	70	80
12	350	22	210	38	120	75	80
13	350	24	210	40	120	80	60
14	300	26	180	45	120	85	60
15	300	28	180	50	100	90	60
16	300	30	180	55	100	95	50
17	250	32	150	60	100	100	50

BRITISH STANDARD FINE (B.S. FINE) SCREW THREADS.
(No. 84—1940.) (*Abstract.*)

1	2	3	4	5	6	7	8
Nominal diameter.	Number of threads per inch.	Pitch.	Depth of thread.	Major diameter.	Effective diameter.	Minor diameter.	Cross sectional area at bottom of thread.
in.		in.	in.	in.	in.	in.	sq. in.
3/16	32	0·031 25	0·0200	0·1875	0·1675	0·1475	0·0171
7/32	28	0·035 71	0·0229	0·2188	0·1959	0·1730	0·0235
1/4	26	0·038 46	0·0246	0·2500	0·2254	0·2008	0·0317
9/32	26	0·038 46	0·0246	0·2812	0·2566	0·2320	0·0423
5/16	22	0·045 45	0·0291	0·3125	0·2834	0·2543	0·0508
3/8	20	0·050 00	0·0320	0·3750	0·3430	0·3110	0·0760
7/16	18	0·055 56	0·0356	0·4375	0·4019	0·3663	0·1054
1/2	16	0·062 50	0·0400	0·5000	0·4600	0·4200	0·1385
9/16	16	0·062 50	0·0400	0·5625	0·5225	0·4825	0·1828
5/8	14	0·071 43	0·0457	0·6250	0·5793	0·5336	0·2236
11/16	14	0·071 43	0·0457	0·6875	0·6418	0·5961	0·2791
3/4	12	0·083 33	0·0534	0·7500	0·6966	0·6432	0·3249
13/16	12	0·083 33	0·0534	0·8125	0·7591	0·7057	0·3911
7/8	11	0·090 91	0·0582	0·8750	0·8168	0·7586	0·4520
1	10	0·100 00	0·0640	1·0000	0·9360	0·8720	0·5972
1 1/8	9	0·111 11	0·0711	1·1250	1·0539	0·9828	0·7586
1 1/4	9	0·111 11	0·0711	1·2500	1·1789	1·1078	0·9639
1 3/8	8	0·125 00	0·0800	1·3750	1·2950	1·2150	1·159
1 1/2	8	0·125 00	0·0800	1·5000	1·4200	1·3400	1·410
1 5/8	8	0·125 00	0·0800	1·6250	1·5450	1·4650	1·686
1 3/4	7	0·142 86	0·0915	1·7500	1·6585	1·5670	1·928
2	7	0·142 86	0·0915	2·0000	1·9085	1·8170	2·593
2 1/4	6	0·166 67	0·1067	2·2500	2·1433	2·0366	3·258
2 1/2	6	0·166 67	0·1067	2·5000	2·3933	2·2866	4·106
2 3/4	6	0·166 67	0·1067	2·7500	2·6433	2·5366	5·054
3	5	0·200 00	0·1281	3·0000	2·8719	2·7438	5·913
3 1/4	5	0·200 00	0·1281	3·2500	3·1219	2·9938	7·039
3 1/2	4·5	0·222 22	0·1423	3·5000	3·3577	3·2154	8·120
3 3/4	4·5	0·222 22	0·1423	3·7500	3·6077	3·4654	9·432
4	4·5	0·222 22	0·1423	4·0000	3·8577	3·7154	10·84
4 1/4	4	0·250 00	0·1601	4·2500	4·0899	3·9298	12·13

Note.—It is recommended that for larger diameters in this series four threads per inch be used

British Standard Brass Thread.

This is a pipe thread system adopted for thin brass tubing, gas burner fittings, and general brass work; it has a Whitworth thread of a constant pitch of $\frac{1}{26}$-in., or 26 threads per inch.

Diameter.	In.	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$
Threads per inch	.	26 throughout		—	—	—	—
Depth of Thread	In.	0·02461 throughout		—	—	—	—
Core Diameter .	In.	0·0758	0·2008	0·3258	0·4508	0·5758	0·700
Tapping Size .	In.	No. 47	No. 6	Q	$\frac{29}{64}$	$\frac{37}{64}$	$\frac{45}{64}$
Diameter.	In.	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	—
Threads per Inch	.	26 throughout		—	—	—	—
Depth of Thread	In.	0·02461 throughout		—	—	—	—
Core Diameter .	In.	0·8258	0·9508	1·0758	1·2008	1·4508	—
Tapping Size .	In.	$\frac{53}{64}$	$\frac{61}{64}$	$1\frac{5}{64}$	$1\frac{13}{64}$	$1\frac{29}{64}$	—

British Association (B.A.) Screw Threads.

(No. 93—1919.) (Amended 1940.) (*Abstract.*)

No.	Approximate Dimensions in Decimals of an Inch. Diam.	Pitch.	Threads per Inch.	Exact Dimensions in Millimetres. Diam.	Pitch.	No.	Approximate Dimensions in Decimals of an Inch. Diam.	Pitch.	Threads per Inch.	Exact Dimensions in Millimetres. Diam.	Pitch.
25	·010	·0028	353	0·25	0·072	12	·051	·0110	90·7	1·3	0·28
24	·011	·0031	317	0·29	0·080	11	·059	·0122	81·9	1·5	0·31
23	·013	·0035	285	0·33	0·089	10	·067	·0138	72·6	1·7	0·35
22	·015	·0039	259	0·37	0·098	9	·075	·0154	65·1	1·9	0·39
21	·017	·0043	231	0·42	0·11	8	·087	·0169	59·1	2·2	0·43
20	·019	·0047	212	0·48	0·12	7	·098	·0189	52·9	2·5	0·48
19	·021	·0055	181	0·54	0·14	6	·110	·0209	47·9	2·8	0·53
18	·024	·0059	169	0·62	0·15	5	·126	·0232	43·0	3·2	0·59
17	·028	·0067	149	0·70	0·17	4	·142	·0260	38·5	3·6	0·66
16	·031	·0075	134	0·79	0·19	3	·161	·0287	34·8	4·1	0·73
15	·035	·0083	121	0·90	0·21	2	·185	·0319	31·4	4·7	0·81
14	·039	·0091	110	1·0	0·23	1	·209	·0354	28·2	5·3	0·90
13	·047	·0098	101	1·2	0·25	0	·236	·0394	25·4	6·0	1·00

Acme Screw Threads.

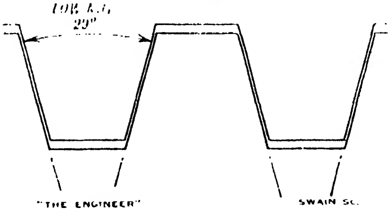


Fig. 7.

The contour of the Acme thread is given in fig. 7. It is the subject of the B.S.I. Standard 1104—1943. (Now withdrawn, but to be re-issued.) The angle is 29° and the depth of the thread one-half the pitch. The standard series goes from $\frac{1}{2}$-in. up to 5 ins. and from 10 threads per inch down to 2 per inch.

Screw Heads.

British Standard Heads for Small Screws.

(No. 450—1932.) (*Abstract.*)

For all classes of heads except cheese, the diameter to be 1·75 times the full diameter of the thread ; for cheeseheads, 1·5.

The width of saw-cuts to be according to the formula,

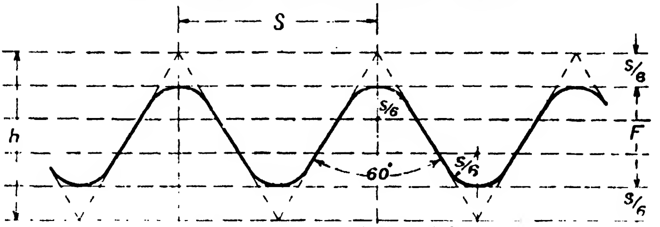
$$S = \cdot 2\,D + \cdot 1 \text{ mm.},$$

where

S = width of saw-cut, and D = full diameter of the thread in millimetres.

The standard depth of saw-cut to equal one-half the total depth of the head in all cases, such depth being measured at the centre of the head.

Cycle Engineers' Institute Standard Threads.



$h = 0{\cdot}866$ S. F = 0·5327 S.

Fig. 8.

The dimensions given below are for right-hand threads except those marked ** which are for left-hand threads only, and that marked * (i.e. $\frac{9}{16}$ in.) is for right or left hand threads.

Diameter in Inches. Decimals and Fractions.	No. of Threads per In.	Diameter in Inches. Decimals and Fractions.	No. of Threads per In.	Diameter in Inches. Decimals and Fractions.	No. of Threads per In.
·056 —	62	·154 —	40	·375 $\frac{3}{8}$	26
·064 —	62	·175 —	32	*·5625 $\frac{9}{16}$	20
·072 —	62	·1875 $\frac{3}{16}$	32	1·000 1	26
·080 —	62	·250 $\frac{1}{4}$	26	**1·290 —	24
·092 —	56	·266 —	26	1·370 —	24
·104 —	44	·281 —	26	**1·4375 $1\frac{7}{16}$	24
·125 $\frac{1}{8}$	40	·3125 $\frac{5}{16}$	26	1·5000 $1\frac{1}{2}$	24

Continental and American Threads.

International System of Metrical Screw Threads.

(See B.S.S. 1095—1943. Amended 1944–45.)

(*Système International.*)

Dia. of Bolt in mm.	Pitch in mm.	Dia. of Bolt in mm.	Pitch in mm.	Dia. of Bolt in mm.	Pitch in mm.	Dia. of Bolt in mm.	Pitch in mm.	Dia. of Bolt in mm.	Pitch in mm.
3	0·55	8	1·25	18	2·5	36	4	60	5·5
3·5	0·55	9	1·25	20	2·5	39	4	64	6
4	0·7	10	1·5	22	2·5	42	4·5	68	6
4·5	0·7	11	1·5	24	3	45	4·5	72	6·5
5	0·85	12	1·75	27	3	48	5	76	6·5
6	1	14	2	30	3·5	52	5	80	7
7	1	16	2	33	3·5	56	5·5		

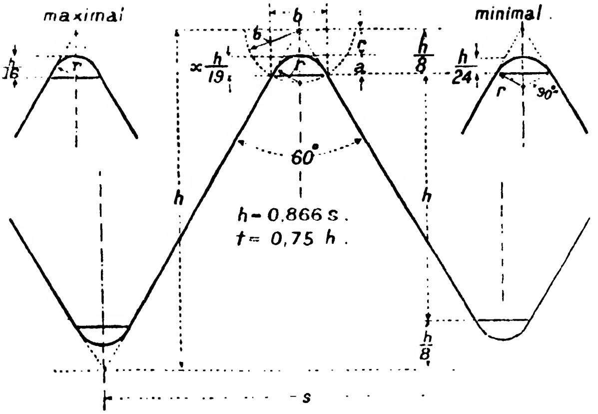


FIG. 9.

Fig. 9 shows the shape and proportions of the threads of the International System of metrical screw threads, also the maximum and minimum deviations allowed from the normal.

FRENCH STANDARD THREADS.

Diam. in Millimètres.	Pitch in Millimètres.			Diam. in Millimetres.	Pitch in Millimètres.		
	Cail or Railway.	Ducommun.	Armengaud's.		Cail or Railway.	Ducommun.	Armengaud's.
3		·5		25	3	3	3
4		·75	–	28	3	3	-
5	1	·75	1·4	30	3	3·5	3·4
6	1	1	-	32	3	3·5	–
7	1·25	1·2	–	35	3·5	4	3·8
7·5	—	—	1·6	38	3·5	4	—
8	1·25	1·25		40	4	4	4·2
9	1·5	1·5		42	4	4·5	-
10	1·5	1·5	1·8	45	4	4·5	4·6
12	1·5	1·75		47	4·5	5	—
12·5	—	—	2	50	4·5	5	5
14	2	—	-	55	-	5	5·4
15	2	2	2·2	60	–	6	5·8
17·5	—	—	2·4	65	-	6	6·2
18	2	2·5		70	—	7	6·6
20	2	2·5	2·6	75	—	7	7
22·5		—	2·8	80	—	7	7·4
23	2·5	3	-				

U.S.A. National Coarse Threads.

Size.	Threads per Inch	Major Diameter	Pitch Diameter	Minor Diameter	Pitch	Basic Depth of Thread	Basic Width of Flat
1/4	20	0·2500	0·2175	0·1850	0·0500000	0·03248	0·00625
5/16	18	0·3125	0·2764	0·2403	0·0555556	0·03608	0·00694
3/8	16	0·3750	0·3344	0·2938	0·0625000	0·04059	0·00781
7/16	14	0·4375	0·3911	0·3447	0·0714286	0·04639	0·00893
1/2	13	0·5000	0·4500	0·4001	0·0769231	0·04996	0·00962
9/16	12	0·5625	0·5084	0·4542	0·0833333	0·05413	0·01042
5/8	11	0·6250	0·5660	0·5069	0·0909091	0·05905	0·01136
3/4	10	0·7500	0·6850	0·6201	0·1000000	0·06495	0·01250
7/8	9	0·8750	0·8028	0·7307	0·1111111	0·07217	0·01389
1	8	1·0000	0·9188	0·8376	0·1250000	0·08119	0·01562
1 1/8	7	1·1250	1·0322	0·0394	0·1428571	0·09279	0·01786
1 1/4	7	1·2500	1·1572	1·0644	0·1428571	0·09279	0·01786
1 3/8	6	1·3750	1·2667	1·1585	0·1666667	0·10825	0·02083
1 1/2	6	1·5000	1·3917	1·2835	0·1666667	0·10825	0·02083
1 3/4	5	1·7500	1·6201	1·4902	0·2000000	0·12990	0·02500
2	4½	2·0000	1·8557	1·7113	0·2222222	0·14434	0·02778
2 1/4	4½	2·2500	2·1057	1·9613	0·2222222	0·14434	0·02778
2 1/2	4	2·5000	2·3376	2·1752	0·2500000	0·16238	0·03125
2 3/4	4	2·7500	2·5876	2·4252	0·2500000	0·16238	0·03125
3	4	3·0000	2·8376	2·6752	0·2500000	0·16238	0·03125
3 1/4	4	3·2500	3·0876	2·9252	0·2500000	0·16238	0·03125
3 1/2	4	3·5000	3·3376	3·1752	0·2500000	0·16238	0·03125
3 3/4	4	3·7500	3·5876	3·4252	0·2500000	0·16238	0·03125
4	4	4·0000	3·8376	3·6752	0·2500000	0·16238	0·03125

All dimensions given in inches.

Swiss Screw Threads. (*Thury.*)

No.	Exact Dimensions in Millimetres.		Ratio of Successive Diameters	Approx. Diam. in Decimals of an Inch.	Threads per Inch	No.	Exact Dimensions in Millimetres.		Ratio of Successive Diameters	Approx. Diam. in Decimals of an Inch.	Threads per Inch.
	Pitch.	External Diameter.					Pitch.	External Diameter.			
25	0·0718	0·254	0·879	0·0100	353·70	2	0·810	4·66	0·881	0·1835	31·35
24	0·0798	0·289	0·881	0·0114	318·30	1	0·900	5·29	0·882	0·2083	28·22
23	0·0886	0·328	0·882	0·0129	286·65	0	1	6	0·881	0·2362	25·40
22	0·0985	0·372	0·872	0·0146	255·32	— 1	1·11	6·81	0·881	0·2681	22·88
21	0·109	0·426	0·889	0·0168	233·02	— 2	1·23	7·73	0·881	0·3043	20·65
20	0·122	0·479	0·882	0·0189	208·19	— 3	1·37	8·77	0·881	0·3453	18·54
19	0·135	0·543	0·882	0·0214	188·40	— 4	1·52	9·95	0·880	0·3917	16·73
18	0·150	0·616	0·881	0·0243	169·33	— 5	1·69	11·3	0·883	0·4449	15·02
17	0·167	0·699	0·880	0·0275	152·32	— 6	1·88	12·8	0·883	0·5039	13·48
16	0·185	0·794	0·881	0·0313	137·29	— 7	2·09	14·5	0·879	0·5709	12·15
15	0·206	0·901	0·883	0·0355	123·30	— 8	2·32	16·5	0·882	0·6496	10·94
14	0·229	1·02	0·879	0·0402	110·91	— 9	2·58	18·7	0·882	0·7362	9·84
13	0·254	1·16	0·879	0·0457	100·00	—10	2·87	21·2	0·880	0·8346	8·85
12	0·282	1·32	0·886	0·0520	90·07	—11	3·19	24·1	0·880	0·9488	7·97
11	0·314	1·49	0·909	0·0587	80·89	—12	3·54	27·4	0·884	1·0787	7·17
10	0·349	1·64	0·854	0·0646	72·77	—13	3·93	31·0	0·881	1·2048	6·46
9	0·387	1·92	0·881	0·0756	65·63	—14	4·37	35·2	0·880	1·3858	5·81
8	0·430	2·18	0·879	0·0858	59·06	—15	4·86	40·0	0·881	1·5748	5·22
7	0·478	2·48	0·883	0·0976	53·13	—16	5·40	45·4	0·882	1·7874	4·84
6	0·531	2·81	0·881	0·1106	47·83	—17	6·00	51·5	0·882	2·0276	4·23
5	0·590	3·19	0·881	0·1256	43·05	—18	6·66	58·4	0·881	2·2992	3·82
4	0·656	3·62	0·881	0·1425	38·71	—19	7·40	66·3	0·882	2·6103	3·43
3	0·729	4·11	0·882	0·1618	34·84	—20	8·28	75·2	0·882	2·9607	3·08

Comparison of U.S. Standard and Whitworth Screw Threads.

The difference in form that exists between the Whitworth and the U.S. standard screw thread is shown in fig. 10. The comparison is based upon a screw with 1 thread per inch. Clearances and interferences for the two forms of threads having any other pitch may therefore be obtained by dividing the respective values given in the diagram by the number of threads per inch under consideration.

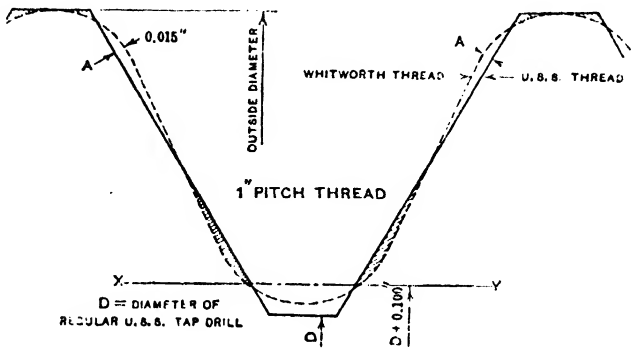


FIG. 10.

A new series of Unified Screw Threads has now been agreed on for common use as an alternativ to the existing Whitworth and U.S. Standards.

PIPE THREADS.

Gauge Diameter.

The full diameter of the Standard **Male Parallel Screw Gauge** which the Parallel Coupler, to be used **with a tube of that size, is required to fit.**

U.S.A. National Pipe Threads.

Nominal Diameter in Inches.	Outside Diameter in Inches.	No. of Threads per Inch.	Nominal Diameter in Inches.	Outside Diameter in Inches.	No. of Threads per Inch.
1/8	0·405	27	2½	2·875	8
¼	0·54	18	2¾	—	—
3/8	0·675	18	3	3·5	8
½	0·84	14	3½	4·0	8
5/8	—	—	4	4·5	8
¾	1·05	14	4½	5·0	8
7/8	—	—	5	5·563	8
1	1·315	11½	6	6·625	8
1¼	1·66	11½	7	7·625	8
1½	1·9	11½	8	8·625	8
1¾	—	—	9	9·688	8
2	2·375	11½	10	10·75	8
2¼	—	—	12	12·75	8

BASIC SIZES FOR B.S. PIPE THREADS. (B.S.S. No. 21 – 1938.)

1	2	3	4	5	6	7	8	9	10	11	12
B.S.P. Size (Nom. Bore of Tube).	Outside Diameter of Black Tube.				No. of Threads per inch.	Pitch	Depth of Thread.	Diameters at Gauge Plane (Basic).			Gauge Length.
	Max.	Min.	Mean.	Tol.				Major (Gauge Diam.)	Effective.	Minor.	
in.	in.	in.	in.	in.		in.	in.	in.	in.	in.	in.
⅛	0·412	0·387	0·400	0·025	28	0·03571	0·0229	0·383	0·3601	0·3372	0·1563
¼	0·550	0·525	0·538	0·025	19	0·05263	0·0337	0·518	0·4843	0·4506	0·2367
⅜	0·688	0·663	0·676	0·025	19	0·05263	0·0337	0·656	0·6223	0·5886	0·2500
½	0·859	0·834	0·847	0·025	14	0·07143	0·0457	0·825	0·7793	0·7336	0·3214
¾	1·075	1·050	1·063	0·025	14	0·07143	0·0457	1·041	0·9953	0·9496	0·3750
1	1·351	1·320	1·336	0·031	11	0·09091	0·0582	1·309	1·2508	1·1926	0·4091
1¼	1·692	1·661	1·677	0·031	11	0·09091	0·0582	1·650	1·5918	1·5336	0·5000
1½	1·924	1·893	1·909	0·031	11	0·09091	0·0582	1·882	1·8238	1·7656	0·5000
2	2·403	2·358	2·381	0·045	11	0·09091	0·0582	2·347	2·2888	2·2306	0·6250
2½	3·021	2·971	2·996	0·050	11	0·09091	0·0582	2·960	2·9018	2·8436	0·6875
3	3·526	3·471	3·499	0·055	11	0·09091	0·0582	3·460	3·4018	3·3436	0·8125
3½	4·021	3·961	3·991	0·060	11	0·09091	0·0582	3·950	3·8918	3·8336	0·8750
4	4·526	4·461	4·494	0·065	11	0·09091	0·0582	4·450	4·3918	4·3336	1·0000
5	5·536	5·461	5·498	0·075	11	0·09091	0·0582	5·450	5·3918	5·3336	1·1250
6	6·541	6·461	6·501	0·080	11	0·09091	0·0582	6·450	6·3918	6·3336	1·1250
7	7·575	7·463	7·519	0·112	10	0·10000	0·0640	7·450	7·3860	7·3220	1·3750
8	8·585	8·463	8·524	0·122	10	0·10000	0·0640	8·450	8·3860	8·3220	1·5000
9	9·595	9·463	9·529	0·132	10	0·10000	0·0640	9·450	9·3860	9·3220	1·5000
10	10·605	10·463	10·534	0·142	10	0·10000	0·0640	10·450	10·3860	10·3220	1·6250
11	11·615	11·465	11·540	0·150	8	0·12500	0·0800	11·450	11·3700	11·2900	1·6250
12	12·625	12·465	12·545	0·160	8	0·12500	0·0800	12·450	12·3700	12·2900	1·6250

NOTE.—*Tubes 7 in. and upwards*: The ends shall be specially sized prior to screwing, in order to ensure ample thickness below the root of the thread. This condition shall be complied with for the screwing of cut tube at site.

BOLTS AND NUTS.

According to the Whitworth system of standard sizes of bolts and nuts, the thickness of the bolt-head is seven-eighths of the diameter, and that of the nut is equal to the diameter.

In the Sellers' (American) system the heads and nuts are hexagonal, the thickness of the heads and of the nuts being equal to 1 diameter minus one-sixteenth inch. The breadth across the flats is equal to 1½ diameters plus one-sixteenth inch.

Dimensions of B.S. Black Bolts, Nuts and Lock Nuts (Hexagon and Square).

Nominal Size and Maximum Diameter of Bolt.	Number of Threads per Inch.		Dimension across Flats of Bolt Heads and Nuts (Hexagon and Square).		Approx. Maximum Dimension across Corners.		Thickness of Bolt Heads (Hexagon and Square).		Thickness of Nuts (Hexagon and Square).		Thickness of Lock Nuts (Hexagon) F.	
—	B.S. Whit.	B.S. Fine.	Max.	Min.	Hexagon.	Sq.	Max.	Min.	Max.	Min.	Max.	Min.
in.			in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
1/4	20	26	0·445	0·435	0·51	0·63	0·20	0·18	0·22	0·20	0·14	0·12
5/16	18	22	0·525	0·515	0·61	0·74	0·23	0·21	0·27	0·25	0·18	0·16
3/8	16	20	0·600	0·585	0·69	0·85	0·28	0·26	0·33	0·31	0·22	0·20
7/16	14	18	0·710	0·695	0·82	1·00	0·34	0·32	0·39	0·37	0·26	0·24
1/2	12	16	0·820	0·800	0·95	1·16	0·40	0·37	0·46	0·43	0·31	0·28
9/16	12	16	0·920	0·900	1·06	1·30	0·46	0·43	0·53	0·50	0·35	0·32
5/8	11	14	0·940	0·985	1·17	1·43	0·51	0·48	0·60	0·56	0·41	0·37
3/4	10	12	1·200	1·175	1·39	1·70	0·62	0·59	0·72	0·68	0·49	0·45
7/8	9	11	1·300	1·270	1·50	1·84	0·69	0·65	0·81	0·75	0·55	0·49
1	8	10	1·480	1·450	1·71	2·09	0·80	0·76	0·93	0·87	0·63	0·57
1 1/8	7	9	1·670	1·640	1·93	2·36	0·91	0·87	1·06	1·00	0·72	0·65
1 1/4	7	9	1·860	1·815	2·15	2·63	1·02	0·96	1·20	1·12	0·81	0·73
1 3/8	6*	8	2·050	2·005	2·37	2·90	1·13	1·07	1·33	1·25	0·89	0·81
1 1/2	6	8	2·220	2·175	2·56	3·14	1·24	1·18	1·45	1·37	0·98	0·90
1 5/8	5*	8	2·410	2·365	2·78	3·41	1·39	1·29	1·58	1·50	1·06	0·98
1 3/4	5	7	2·580	2·520	2·98	3·65	1·50	1·40	1·72	1·62	1·16	1·06
2	4·5	7	2·760	2·700	3·19	3·90	1·61	1·51	1·85	1·75	1·25	1·15
2 1/4	4	6	3·150	3·090	3·64	4·45	1·77	1·67	1·97	1·87	1·43	1·33
2 1/2	4	6	3·550	3·490	4·10	5·02	1·99	1·89	2·22	2·12	1·60	1·50
2 3/4	3·5	6	3·890	3·830	4·49	5·50	2·16	2·06	2·47	2·37	1·77	1·67
3	3·5	5	4·180	4·080	4·83	5·91	2·43	2·28	2·77	2·62	1·98	1·83
3 1/4	3·25	5	4·530	4·430	5·23	6·41	2·75	2·50	3·12	2·87	2·15	2·00
3 1/2	3·25	4·5	4·850	4·750	5·60	6·86	2·84	2·71	3·27	3·12	2·32	2·17
3 3/4	3	4·5	5·180	5·080	5·98	·33	3·08	2·93	3·52	3·37	2·48	2·33
4	3	4·5	5·550	5·450	6·41	7·85	3·30	3·15	3·77	3·62	2·65	2·50
4 1/2	2·875	4	6·380	6·255	7·37	9·02	3·73	3·53	4·20	4·00	3·03	2·83
5	2·75	4	7·300	7·175	8·43	10·32	4·17	3·97	4·70	4·50	3·37	3·17
5 1/2	2·625	4	8·350	8·225	9·64	11·81	4·60	4·40	5·20	5·00	3·70	3·50
6	2·5	4	9·450	9·325	10·91	13·36	5·01	4·84	5·70	5·50	4·03	3·83

* The Institution recommends that for general use these sizes be dispensed with.

WHITWORTH STANDARD BOLTS AND NUTS. SAFE LOAD IN LBS.

Diam. of Bolt.	Diam. at Bottom of Thread.	Area at Bottom of Thread.	APPROXIMATE SAFE LOAD IN LBS.					
			At 4,000 lbs. per Sq. In.	At 5,000 lbs. per Sq. In.	At 6,000 lbs. per Sq. In.	At 7,000 lbs. per Sq. In.	At 8,000 lbs. per Sq. In.	At 9,000 lbs. per Sq. In.
Ins.	Ins.	Sq. Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{3}{16}$	0·134	0·014	56	70	84	98	112	126
$\frac{1}{4}$	0·186	0·027	108	135	162	189	216	243
$\frac{5}{16}$	0·241	0·046	184	230	276	322	378	414
$\frac{3}{8}$	0·295	0·068	272	340	408	478	546	614
$\frac{7}{16}$	0·346	0·094	376	470	504	628	752	846
$\frac{1}{2}$	0·393	0·121	484	605	726	850	970	1,090
$\frac{9}{16}$	0·455	0·155	620	775	930	1,085	1,240	1,395
$\frac{5}{8}$	0·508	0·204	816	1,020	1,224	1,418	1,620	1,824
$\frac{11}{16}$	0·571	0·256	1,024	1,280	1,536	1,792	2,048	2,304
$\frac{3}{4}$	0·622	0·304	1,216	1,520	1,824	2,126	2,430	2,734
$\frac{13}{16}$	0·684	0·367	1,468	1,835	2,202	2,569	2,936	3,303
$\frac{7}{8}$	0·733	0·422	1,688	2,110	2,532	2,954	3,376	3,800
$\frac{15}{16}$	0·795	0·496	1,984	2,480	2,976	3,472	3,968	4,464
1	0·840	0·554	2,216	2,770	3,324	3,880	4,430	4,990
$1\frac{1}{8}$	0·942	0·697	2,788	3,485	4,182	4,880	5,575	6,270
$1\frac{1}{4}$	1·067	0·894	3.576	4,470	5,364	6,250	7,145	8,035
$1\frac{3}{8}$	1·162	1·058	4,232	5,290	6,348	7,400	8,455	9,510
$1\frac{1}{2}$	1·287	1·299	5,196	6,495	7,794	9,030	10,320	11,610
$1\frac{5}{8}$	1·369	1·472	5,888	7,360	8,832	10,290	11,760	13,230
$1\frac{3}{4}$	1·494	1·753	7,012	8,765	10,518	12,210	13,950	15,700
$1\frac{7}{8}$	1·590	1·986	7,944	9,930	11,916	13,900	15,890	17,875
2	1·720	2·311	9,244	11,555	13,866	16,170	18,480	20,790
$2\frac{1}{4}$	1·930	2·926	11,704	14.630	17.556	20,475	23,400	26,325
$2\frac{1}{2}$	2·180	3·733	14,932	18,665	22,398	26,125	29,855	33,590
$2\frac{3}{4}$	2·384	4·464	17,856	22,320	26,784	31,220	35,680	40,140
3	2·634	5·450	21,800	27,250	32,700	38,080	43,520	48,960
$3\frac{1}{4}$	2·856	6·402	25,608	32,010	38,412	44,814	51,216	57,618
$3\frac{1}{2}$	3·105	7·563	30,252	37,815	45,378	57,940	60,504	68,065
$3\frac{3}{4}$	3·320	8·673	34,692	43,365	52,038	60,710	69,384	78,057
4	3·573	10·027	40,108	50,135	60,162	70,190	80,216	90,240
$4\frac{1}{4}$	3·804	11·365	45,480	56,825	68,190	79,575	90,960	102,305
$4\frac{1}{2}$	4·054	12·908	51,632	64,540	77,448	90,356	103,264	116,172
$4\frac{3}{4}$	4·284	14·404	57,616	72,020	86,424	100,830	115,232	129,636
5	4·534	16·146	64,584	80,730	96,876	113,020	129,168	115,314
$5\frac{1}{2}$	5·012	19·720	78,880	98,600	118,320	138,040	157,760	177,480
6	5·487	23·640	94,560	118,200	141,840	165,200	189,120	212,760

FRENCH STANDARD BOLTS AND NUTS.

Diameter of Screw.		Diameter of Bottom of Thread.		Pitch of Thread.	Number of Threads per Inch.	Width across Flats.		Width across Corners.		Thickness of Head.	
Millimètres.	Inches.	Millimètres.	Inches.	Millimètres.		Millimètres.	Inches.	Millimètres.	Inches.	Millimètres.	Inches.
	·196	3·30	·13	1·4	18·14	13·97	·55	16·0	·63	5	·196
7·5	·295	5·58	·22	1·6	15·87	17·27	·68	19·81	·78	7	·275
10	·393	7·87	·31	1·8	14·11	22·35	·88	25·90	1·02	9	·355
12·5	·492	9·90	·39	2	12·70	26·12	1·04	30·48	1·20	11	·433
15	·589	12·19	·48	2·2	11·55	30·48	1·20	35·05	1·38	13	·51
17·5	·688	14·73	·58	2·4	10·58	34·54	1·36	39·87	1·57	15	·59
20	·787	16·76	·66	2·6	9·77	38·60	1·52	44·45	1·75	17	·67
22·5	·885	19·20	·76	2·8	9·07	42·67	1·68	49·27	1·94	19	·75
25	·984	21·34	·84	3	8·46	46·74	1·84	53·84	2·12	21	·83
30	1·178	25·90	1·02	3·4	7·47	54·86	2·16	63·50	2·50	24	·94
35	1·376	30·48	1·20	3·8	6·68	62·99	2·48	72·64	2·86	27	1·06
40	1·574	35·56	1·40	4·2	6·05	71·12	2·80	82·04	3·23	31	1·22
45	1·770	39·62	1·56	4·6	5·52	79·25	3·12	91·44	3·60	35	1·38
50	1·968	44·19	1·74	5	5·08	87·37	3·44	100·80	3·97	38	1·50
55	2·162	48·76	1·92	5·4	4·70	95·50	3·76	110·2	4·34	41	1·65
60	2·356	52·83	2·08	5·8	4·38	103·63	4·08	119·8	4·72	44	1·73
65	2·554	57·42	2·26	6·2	4·09	111·76	4·40	129·0	5·08	48	1·89
70	2·752	61·97	2·44	6·6	3·85	118·88	4·72	138·4	5·45	52	2·05
75	2·950	66·04	2·60	7·0	3·63	128·02	5·04	147·8	5·82	56	2·20
80	3·149	70·61	2·78	7·4	3·43	136·14	5·36	157·5	6·20	60	2·36

AMERICAN STANDARD HEXAGON NUTS AND BOLTS (*Sellers'*).

Dimensions in Inches.

Size of Bolt and Thickness of Nut.	Nut across Flats.	Nut across Corners.	Tapping Hole.	Thickness of Bolt-head.	Size of Bolt and Thickness of Nut.	Nut across Flats.	Nut across Corners.	Tapping Hole.	Thickness of Bolt-head.
1/4	·5	·577	·185	·197	2	3·125	3·609	1·711	1·576
5/16	·597	·685	·240	·2472	2 1/4	3·5	4·042	1·961	1·7628
3/8	·687	·794	·294	·2954	2 1/2	3·875	4·475	2·175	1·97
7/16	·781	·902	·345	·3447	2 3/4	4·25	4·908	2·425	2·1528
1/2	·875	1·0106	·4	·394	3	4·625	5·341	2·628	2·364
9/16	·968	1·1187	·454	·4432	3 1/4	5	5·775	2·878	2·561
5/8	1·062	1·227	·506	·4925	3 1/2	5·375	6·208	3·1	2·758
3/4	1·25	1·443	·62	·591	3 3/4	5·75	6·641	3·317	2·954
7/8	1·437	1·66	·731	·6895	4	6·125	7·074	3·567	3·152
1	1·625	1·876	·837	·788	4 1/4	6·5	7·507	3·798	3·349
1 1/8	1·812	2·093	·939	·8864	4 1/2	6·875	7·94	4·027	3·5256
1 1/4	2	2·31	1·064	·985	4 3/4	7·25	8·373	4·255	3·743
1 3/8	2·187	2·526	1·158	1·0764	5	7·625	8·806	4·48	3·94
1 1/2	2·375	2·743	1·283	1·182	5 1/4	8	9·24	4·73	4·137
1 5/8	2·562	2·959	1·389	1·2805	5 1/2	8·375	9·67	4·953	4·334
1 3/4	2·75	3·176	1·49	1·379	5 3/4	8·75	10·106	5·203	4·531
1 7/8	2·937	3·393	1·615	1·477	6	9·125	10·539	5·423	4·728

(See also B.S. Spec. 325—1947, 'Cup and Countersunk Bolts, etc.')

BRITISH STANDARD WHITWORTH CASTLE AND SLOTTED NUTS.*

B.S.S. 1083/1942. (*Abstract.*)

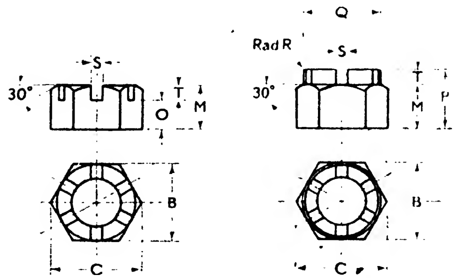


FIG. 11.

* **By permission of the British Standards Institution.**

Dimensions of Slotted Nuts and Castle Nuts.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
	Slotted Nuts				Castle Nuts							Slotted and Castle Nuts	
Nominal size	Thickness M		Face of nut to bottom of slot O		Thickness of hexagon, and from face of nut to bottom of slot M		Total thickness P		Castellated portion Diameter Q		Approx. radius R	Approximate dimensions of slot Width S	Depth T
A	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	R	S	T
*1/4	0.260	0.250	0.170	0.160	0.200	0.190	0.290	0.280	0.430	0.425	0.04	0.060	0.070
5/16	0.280	0.270	0.190	0.180	0.250	0.240	0.340	0.330	0.510	0.500	0.05	0.090	0.090
3/8	0.312	0.302	0.221	0.211	0.312	0.302	0.402	0.392	[illegible]	0.575	0.05	0.090	0.090
7/16	0.375	0.365	0.235	0.225	0.375	0.365	0.515	0.505	0.695	0.685	0.06	0.125	0.140
1/2	0.437	0.427	0.297	0.287	0.437	0.427	0.577	0.567	0.805	0.795	0.06	0.125	0.160
9/16	0.500	0.490	0.313	0.303	0.500	0.490	0.687	0.677	0.905	0.895	0.06	0.165	0.187
5/8	0.562	0.552	0.375	0.365	0.562	0.552	0.749	0.739	0.995	0.985	0.07	0.165	0.187
3/4	0.687	0.677	0.453	0.443	0.687	0.677	0.921	0.911	1.185	1.165	0.08	0.208	0.234
7/8	0.750	0.740	0.516	0.506	0.750	0.740	0.984	0.974	1.285	1.265	0.08	0.208	0.234
1	0.875	0.865	0.595	0.585	0.875	0.865	1.155	1.145	1.465	1.445	0.09	0.250	0.280
1 1/8	1.000	0.990	0.720	0.710	1.000	0.990	1.280	1.270	1.625	1.605	0.09	0.250	0.280
1 1/4	1.125	1.105	0.797	0.777	1.125	1.105	1.455	1.435	1.645	[illegible]	0.10	0.290	0.328
*1 3/8	1.250	1.230	0.922	0.902	1.250	1.230	1.588	1.568	[illegible]	[illegible]	0.11	0.290	[illegible]
1 1/2	1.375	1.355	1.047	1.027	1.375	1.355	1.705	1.685	2.200	2.180	0.12	0.290	[illegible]
1 3/4	1.625	1.605	1.250	1.230	1.625	1.605	2.000	1.980	[illegible]	[illegible]	0.13	0.333	[illegible]
2	1.750	1.730	1.282	[illegible]	1.750	1.730	2.218	2.198	[illegible]	[illegible]	0.14	[illegible]	[illegible]

* [illegible] B.S.W. [illegible]

Lock Nuts for Feeding Screws of Machine Tools.*

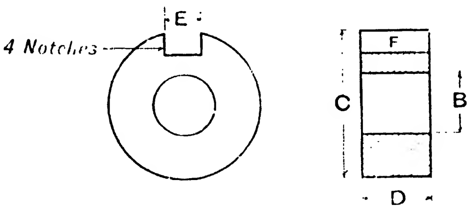


Fig. 12.

Dimensions in Inches.					Number of Threads per Inch.	Dimensions in Inches.					Number of Threads per Inch.
B.	C.	D.	E.	F.		B.	C.	D.	E.	F.	
1/2	1	1/4	3/16	3/32	14	1 5/8	2 7/8	13/16	3/8	3/16	11
5/8	1 1/8	5/16	"	"	14	1 3/4	3 1/16	7/8	"	"	11
3/4	1 1/4	3/8	"	1/8	14	1 7/8	3 1/4	15/16	"	"	11
7/8	1 3/8	7/16	1/4	"	11	2	3 1/2	1	"	"	11
1	1 1/2	1/2	"	"	11	2 1/4	4	1	7/16	7/32	8
1 1/8	2	9/16	5/16	5/32	11	2 1/2	4 3/8	1 1/8	"	"	8
1 1/4	2 1/4	5/8	"	"	11	2 3/4	4 3/4	1 1/4	1/2	1/4	8
1 3/8	2 3/8	11/16	"	3/16	11	3	5 1/4	1 3/8	"	"	8
1 1/2	2 1/2	3/4	"	"	11						

* A pair of these nuts is used for locking purposes.

ROUND BOLT HEADS.

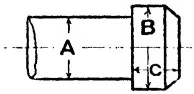


FIG 13.

PROPORTIONS OF ROUND-HEADED BOLTS.

A	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$
B	[illegible]	$\frac{9}{16}$	$\frac{3}{4}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{9}{16}$	$1\frac{3}{4}$	$1\frac{7}{8}$	$2\frac{1}{16}$	$2\frac{1}{4}$	$2\frac{3}{8}$
C	$\frac{3}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{5}{16}$

CAPPED NUTS

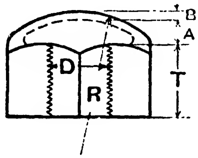


FIG. 14.

D = dia. of bolt ; R = D;

$T = D$; $A = \frac{D}{2}$; $B = \frac{1}{8}$ in.

WING NUTS.

(See B.S.S. 856—1939.)

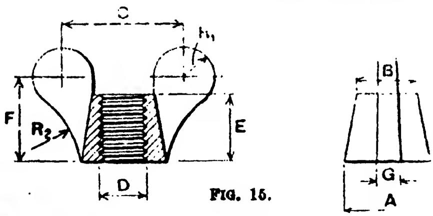


FIG. 15.

WING NUT PROPORTIONS.

The following table gives various dimensions for wing nuts threaded for bolts from $\frac{1}{4}$-inch to 1-inch Whitworth. The dimensions chosen are representative of usual practice.

D.	A.	B.	C.	E.	F.	G.	H.	R_1.	R_2.
	in.	in.	in.	in.	in.	in.	in.	in.	in.
$\frac{1}{4}$" Whit.	$\frac{3}{8}$	$\frac{1}{4}$	[illegible]	$\frac{5}{16}$	[illegible]	$\frac{3}{32}$	$\frac{1}{16}$	[illegible]	[illegible]
$\frac{5}{16}$ "	$\frac{7}{16}$	$\frac{5}{16}$	[illegible]	[illegible]	$\frac{15}{32}$	$\frac{3}{32}$	$\frac{1}{16}$	$\frac{3}{32}$	[illegible]
[illegible] "	[illegible]	[illegible]	[illegible]	$\frac{7}{16}$	$\frac{9}{16}$	[illegible]	$\frac{3}{32}$	$\frac{3}{16}$	[illegible]
[illegible] "	[illegible]	$\frac{11}{32}$	$\frac{29}{32}$	[illegible]	$\frac{21}{32}$	$\frac{5}{32}$	[illegible]	$\frac{7}{32}$	[illegible]
[illegible] "	$\frac{11}{16}$	$\frac{17}{32}$	$1\frac{1}{32}$	$\frac{9}{16}$	[illegible]	$\frac{3}{16}$	$\frac{5}{32}$	[illegible]	[illegible]
$\frac{7}{16}$ "	[illegible]	$\frac{9}{16}$	$1\frac{1}{8}$	[illegible]	$\frac{13}{16}$	$\frac{3}{16}$	$\frac{5}{32}$	$\frac{9}{32}$	[illegible]
[illegible] "	[illegible]	[illegible]	$1\frac{1}{4}$	$\frac{11}{16}$	[illegible]	$\frac{7}{32}$	$\frac{3}{16}$	$\frac{5}{16}$	[illegible]
$\frac{9}{16}$ "	$\frac{15}{16}$	$\frac{11}{16}$	$1\frac{7}{16}$	[illegible]	$\frac{29}{32}$	$\frac{7}{32}$	$\frac{3}{16}$	[illegible]	[illegible]
[illegible] "	1	$\frac{23}{32}$	[illegible]	$\frac{13}{16}$	$\frac{15}{16}$	[illegible]	$\frac{7}{32}$	$\frac{7}{16}$	1
$\frac{11}{16}$ "	$1\frac{1}{8}$	$\frac{27}{32}$	[illegible]	[illegible]	$\frac{31}{32}$	[illegible]	$\frac{7}{32}$	[illegible]	1
[illegible] "	$1\frac{1}{4}$	$\frac{15}{16}$	$1\frac{15}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$	[illegible]	$\frac{7}{32}$	[illegible]	[illegible]
$\frac{13}{16}$ "	$1\frac{5}{16}$	1	$2\frac{1}{8}$	1	$1\frac{1}{8}$	$\frac{9}{32}$	[illegible]	$\frac{9}{16}$	[illegible]
[illegible] "	$1\frac{3}{8}$	$1\frac{1}{16}$	$2\frac{3}{16}$	$1\frac{1}{16}$	$1\frac{3}{16}$	$\frac{9}{32}$	[illegible]	$\frac{9}{16}$	[illegible]
$\frac{15}{16}$ "	$1\frac{7}{16}$	$1\frac{1}{8}$	$2\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$\frac{5}{16}$	[illegible]	[illegible]	[illegible]
1 "	$1\frac{1}{2}$	$1\frac{3}{16}$	$2\frac{7}{16}$	$1\frac{3}{16}$	$1\frac{5}{16}$	[illegible]	$\frac{9}{32}$	[illegible]	2

Spanners.

BRITISH STANDARD SPANNERS.* (No. 192—1943.) (*Abstract.*)

The angle between the head and the shank of spanners should be 15°, 30°, or 45°.

DIMENSIONS OF BRITISH STANDARD SPANNERS.

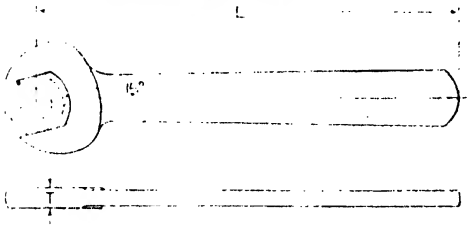


FIG. 16.
Single-ended B.S. Spanner.

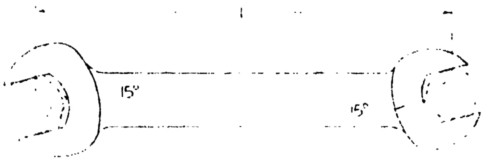


FIG. 17.
Double-ended B.S. Spanner.

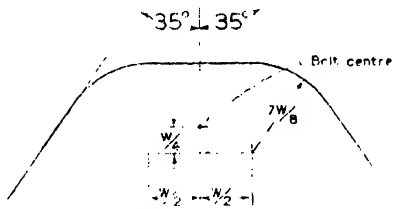


FIG. 18.
Limiting Size of Spanner Head.

BOX SPANNERS OR KEYS.

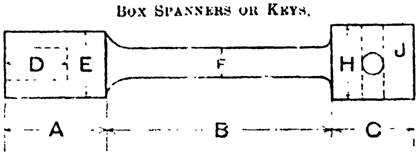


FIG. 19.

Dimensions in Inches.								
Nut.	A.	B.	C.	D.	E.	F.	H.	J.
5/8	1 1/4	3 1/2	1	7/8	1 5/8	3/4	1 1/8	7/16
3/4	1 1/2	3 3/4	1 1/4	1	1 7/8	7/8	1 1/4	9/16
7/8	1 3/4	4	1 3/4	1 1/4	2 1/4	1	1 1/2	11/16
1	2	4 1/2	2	1 1/2	2 1/2	1 1/4	2	
1 1/8	2 3/8	5	2 1/4	1 5/8	2 3/4	1 3/8	2 1/4	13/16
1 1/4	2 5/8	5 1/4	2 1/2	1 3/4	3	1 1/2	2 1/2	1 1/16
1 1/2	3	5 1/2	3	2	3 3/8	1 5/8	2 3/4	1 1/16

* By permission of the British Standards Institution.

DIMENSIONS AND PROOF TEST MOMENTS

1	2	3	4	5	6	7	8	9	10
Nominal size of spanner (dia. of corresponding bolt) †		Width between jaws W		Thickness of spanner head (see Clause 5) (T. Fig. 1)		Length of spanner (see Clause 6) (L. Figs. 1 and 2)			Turning moment for proof test
B.S.W.	B.S.F. and war emergency B.S.W.	Min.	Max.	Min.	Max.	Normal	Long	Extra long	
in.	in.	in.	in.	in.	in.	in.	in.	in.	inch-pounds
—	7/32	0·416	0·419	0·10	0·12	2¼			110
[illegible]	1/4	0·448	0·451	0·12	0·14	3			250
1/4	5/16	0·529	0·533	0·15	0·18	3¾			380
5/16	3/8	0·604	0·608	0·19	0·22	4¼			560
3/8	7/16	0·715	0·720	0·23	0·27	5			780
7/16	1/2	0·825	0·830	0·27	0·31	6			1 000
1/2	9/16	0·926	0·932	0·31	0·36	7	8	9	1 300
9/16	5/8	1·016	1·022	0·35	0·40	8	9	10	1 700
5/8	11/16*	1·107	1·114	0·39	0·44	9	10	11	2 100
11/16*	3/4	1·207	1·214	0·39	0·44	9	10	11	2 500
3/4	7/8	1·308	1·316	0·47	0·53	10½	11½	12½	3 100
13/16*	15/16*	1·398	1·406	0·47	0·53	10½	11½	12½	3 800
7/8	1	1·489	1·498	0·55	0·62	12	13	14	4 700
15/16*	—	1·589	1·598	0·55	0·62	12	13	14	5 700
1	1⅛	1·680	1·690	0·63	0·71	13½	14½	16½	7 000
1⅛	1¼	1·871	1·882	0·71	0·80	15	16½	18	10 000
1¼	1⅜*	2·062	2·074	0·79	0·88	16½	18	20	15 000
1⅜*	1½	2·233	2·246	0·87	0·97	18	20	22	21 000
1½	1⅝*	2·424	2·438	0·95	1·06	20	22	24	30 000
1⅝*	1¾	2·595	2·610	1·03	1·14	22	24	26	41 000
1¾	2	2·776	2·792	1·11	1·23	24	26	28	55 000
1⅞*	—	3·037	3·054	1·22	1·35	26	28	31	75 000
2	2¼	3·168	3·186	1·27	1·40	28	31	34	96 000

* In the relevant specifications for bolts and nuts, it is recommended that, for general use, these sizes be dispensed with.

† For the period of the war emergency, the standard widths across flats of B.S.W. bolts and nuts have been reduced to the standard widths across flats for B.S.F. bolts and nuts (see war emergency B.S. 916 'Black Bolts and Nuts' and 1083 'B.S.W. and B.S.F. Machine Bolts.'

Wood Screws.

Twist Drill Sizes for Wood Screws.

Size of Screw.	Shank Diam.	Drill for Shank.	Mean Core Diam.	Drill for Core.
4/0	0·054	No. 54	0·036	No. 65
3/0	0·057	1·45 mm.	0·0375	No. 62.
2/0	0·060	1·55 mm.	0·0395	No. 60
0	0·063	No. 52	0·0415	No. 58
1	0·066	No. 51	0·0435	1·15 mm.
2	0·080	No. 46	0·052	No. 55
3	0·094	No. 41	0·061	1·55 mm.
4	0·108	2·75 mm.	0·0705	1·80 mm.
5	0·122	3·10 mm.	0·0785	No. 47
6	0·136	No. 29	0·088	2·25 mm.
7	0·150	No. 24	0·0975	No. 40
8	0·164	4·20 mm.	0·106	2·70 mm.
9	0·178	No. 15	0·115	No. 32
10	0·192	4·90 mm.	0·1245	⅛
11	0·206	5·25 mm.	0·1335	3·40 mm.
12	0·220	5·60 mm.	0·143	No. 27
13	0·234	A	0·152	No. 24
14	0·248	6·30 mm.	0·160	No. 20
15	0·262	7·00 mm.	0·170	11/64
16	0·276	J	0·179	No. 15
17	0·290	L	0·189	No. 12
18	0·304	7·75 mm.	0·197	No. 8
20	0·332	Q	0·216	5·50 mm.
22	0·360	9·20 mm.	0·2335	A
24	0·388	9·90 mm.	0·252	6·50 mm.
26	0·416	27/64	0·2695	6·90 mm.
28	0·444	11·50 mm.	0·288	L
30	0·472	12·00 mm.	0·306	7·80 mm.
32	0·500	½	0·326	8·30 mm.
34	0·528	17/32	0·337	R
36	0·556	[illegible]	0·360	9·20 mm.
38	0·584	15·00 mm	0·378	9·60 mm.
40	0·612	⅝	0·397	X

Shank and Core Diameters are given in decimals of an inch.
Fractions in Drill sizes are fractions of an inch.
Mean Core Diameter, is the mean of the variations which the Makers give as possible, but these variations are only in the third place of decimals.

Bolt Spacing.

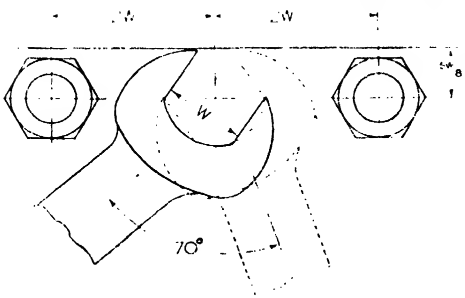


Fig. 20.

Fig. 20 shows minimum spacings and pitches for Whitworth studs, bolts, and nuts. which will give sufficient space to enable a standard spanner to be used. Being for use with hexagonal nuts a movement of 70° full is allowed for the spanner.

Pipe Flanges.

Flanges and Bolts of Cast-iron Pipes.

Diameter of Pipe.	Diameter of Flange.	Flange Thickness.	Size of Bolts.	Number of Bolts.	Bolt Circle.	Weight per Foot.
Ins.	Ins.	Ins.	Ins.		Ins.	Lbs.
1½	6	¾	½	3	4½	—
2	6½	⅞	½	4	4	8·74
2½	7½	1	⅝	4	5⅜	10·58
3	8	1	⅝	4	6⅛	14·8
3½	8½	1	¾	4	6⅝	22·1
4	9¼	$1\frac{1}{16}$	¾	6	7⅛	30·8
4½	10	$1\frac{1}{16}$	¾	6	7⅝	36·3
5	11	1⅛	¾	6	8⅝	52
6	12½	1⅛	¾	6	—	58·7
7	14	$1\frac{3}{16}$	¾	8	11½	71·8
8	15	$1\frac{3}{16}$	¾	8	12½	79·1
9	16½	1¼	¾	8	14	—
10	17½	1¼	¾	10	14¾	—
11	19	$1\frac{5}{16}$	¾	10	16	—
12	20	$1\frac{5}{16}$	⅞	10	17	—

For steam pipes allow 2½ inches per 100 feet for expansion.

British Standard Pipe Flanges.

(No. 10—Part 1—1947.)

Relates to Pipe Flanges (for Land Use) for Gas Pipes up to 30 lbs. pressure per square inch, **and Water Pipes up to 173 lbs. pressure per square inch.**

(No. 10—Part 2—1926.) (*Abstract.*)*

Table M.—Welded-on Flanges for Pipe Lines for Working Steam Pressures up to 250 lbs.

Nominal Pipe Size.	Actual Outside Diameter of Wrought Pipe.	Diameter of Flange.	Diameter of Bolt Circle.	Number of Bolts.	Diameter of Bolts.	Thickness of Flange. Steel (stamped or forged).
ins.	ins.	ins.	ins.		ins.	ins.
2	2⅜	6	4½	4	⅝	$\frac{11}{16}$
2½	3	6½	5	4	⅝	¾
3	3½	7½	5¾	8	⅝	¾
3½	4	8	6½	8	⅝	⅞
4	4½	8½	7	8	⅝	⅞
5	5½	10	8¼	8	⅝	1
6	6½	11	9¼	8	¾	1⅛
7	7⅝	12	10¼	12	¾	1⅛
8	8⅝	13¼	11½	12	¾	1¼
9	9⅝	14¼	12¾	12	¾	1¼
10	10¾	16	14	12	⅞	1⅜
12	12¾	18	16	16	⅞	1½
14	15	20¾	18½	16	1	1⅝
15	16	21¾	19½	16	1	1⅝
16	17	22¾	20½	16	1	1¾
18	19	25¼	23	20	1	1⅞
20	21	27¾	25¼	20	1⅛	2
21	22	29	26½	24	1⅛	2
24	25	32½	29¼	24	1¼	2¼

Bolt holes.—For ⅝-in. and ¾-in. bolts the diameters of the holes to be $\frac{1}{16}$-in. larger than the diameter of the bolts, and for larger sizes of bolts ⅛-in. Bolt holes to be drilled off centre lines.

Tables L and P relate to pressures up to 150 lbs. and 350 lbs. per sq. in., respectively.

* By permission of the British Standards Institution.

Table D.—For Working Steam Pressures up to 50 lbs. per sq. in.

Nominal Pipe Size.	Diameter of Flange.	Diameter of Bolt Circle.	Number of Bolts.	Diameter of Bolts.	Thickness of Flanges. Cast Iron.	Thickness of Flanges. Cast Steel and Bronze.	Thickness of Flanges. Stamped or Forged Iron or Steel.
Ins.	Ins.	Ins.		Ins.	Ins.	Ins.	Ins.
½	3¾	2⅝	4	⅜	½	⅜	3/16
¾	4	2⅞	4	½	½	⅜	3/16
1	4¼	3¼	4	½	⅝	⅜	5/16
1¼	4⅝	3 7/16	4	½	⅝	½	¼
1½	5¼	3⅞	4	½	⅝	½	¼
2	6	4½	4	⅝	¾	9/16	5/16
2½	6⅞	5	4	⅝	¾	9/16	5/16
3	7⅜	5¾	4	⅝	¾	9/16	⅜
3½	8	6½	4	⅝	¾	9/16	⅜
4	8½	7	4	⅝	⅞	11/16	⅜
5	10	8¼	8	⅝	⅞	11/16	½
6	11	9¼	8	⅝	⅞	11/16	½
7	12	10¼	8	⅝	1	¾	½
8	13½	11⅝	8	⅝	1	¾	½
9	14⅝	12¾	8	⅝	1	¾	½
10	16	14	8	¾	1	¾	⅝
12	18	16	12	¾	1⅛	⅞	⅝
14	20¾	18⅝	12	⅞	1¼	1	⅝
15	21¾	19⅝	12	⅞	1¼	1	⅝
16	22¾	20⅝	12	⅞	1¼	1	⅝
18	25¼	23	12	⅞	1⅜	1⅛	¾
20	27¾	25¼	16	⅞	1½	1¼	1
21	29	26½	16	⅞	1⅝	1¼	1
24	32	29¾	16	1	1⅞	1⅜	1⅛

Tables E, F, H, and J relate to pressures up to 100 lbs., 150 lbs., 250 lbs., and 350 lbs., respectively.

Table K.—For Working Steam Pressures up to 450 lbs. per sq. in.

Nominal Pipe Size.	Actual Outside Diameter of Wrought Pipe.	Diameter of Flange.	Diameter of Bolt Circle.	Number of Bolts.	Diameter of Bolts.	Thickness of Flange. Cast Steel and Bronze; Stamped or Forged Steel.
ins.	ins.	ins.	ins.		ins.	ins.
½	27/32	4¼	3¼	4	½	¾
¾	1 1/16	4⅝	3¼	4	½	¾
1	1 11/32	5	3½	4	½	⅞
1¼	1 21/32	5½	3⅞	4	½	⅞
1½	1 29/32	6	4½	4	⅝	1
2	2⅜	6½	5	8	⅝	1
2½	3	7½	5¾	8	⅝	1⅛
3	3½	8	6½	8	⅝	1¼
3½	4	9	7¼	8	¾	1¼
4	4½	9½	7¾	8	¾	1⅜
5	5 9/16	11	9¼	12	¾	1½
6	6⅝	12	10½	12	¾	1⅝
7	7⅝	13½	11¾	12	1	1¾
8	8⅝	14½	12¾	12	1	1⅞
9	9⅝	16	14	16	1	2
10	10¾	17	15	16	1	2
12	12¾	19½	17	16	1⅛	2¼
14	15	22½	20	16	1¼	2⅜
15	16	23¾	21¼	20	1¼	2½
16	17	24¾	22½	20	1¼	2⅝

Bolt holes.—For ½-in. and ⅝-in. bolts the diameters of the holes to be 1/16-in. larger than the diameter of the bolts, and for larger sizes of bolts ⅛-in. Bolt holes to be drilled off centre lines

AMERICAN STANDARD PIPE FLANGES

Pipe Size, Inches.	Pipe Thickness, Inches.	Thickness nearest Fraction, Inches.	Stress on Pipe per Square Inch at 200 lbs.	Radius of Fillet, Inches.	Flange Diameter, Inches.	Flange Thickness at Hub for Iron Pipe, Inches.	Flange Thickness at Edge, Inches.	Width Flange Face, Inches.	Bolt Circle Diameter, Inches.	Number of Bolts.	Bolt Size Diameters, Inches.	Bolt Length, Inches.	Stress on each Bolt per Square Inch at Bottom of Thread at 200 lbs.
2	·409	7/16	460	1/8	6	1	5/8	2	4 3/4	4	1/2 5/8	2	825
2 1/2	·429	7/16	550	1/8	7	1 1/8	11/16	2 1/4	5 1/2	4	1/2 5/8	2 1/4	1,050
3	·448	7/16	690	1/8	7 1/2	1 1/4	3/4	2 1/4	6	4	1/2 5/8	2 1/2	1,330
3 1/2	·466	1/2	700	1/8	8 1/2	1 1/4	13/16	2 1/4	7	4	1/2 5/8	2 1/2	2,530
4	·486	1/2	800	1/8	9	1 5/16	15/16	2 1/4	7 1/2	4	5/8 3/4	2 3/4	2,100
4 1/2	·498	1/2	900	1/8	9 1/4	1 3/8	15/16	2 3/8	7 3/4	8	5/8 3/4	3	1,430
5	·525	1/2	1,000	1/8	10	1 1/2	15/16	2 1/2	8 1/2	8	5/8 3/4	3	1,630
6	·563	9/16	1,060	1/8	11	1 1/2	1	2 1/2	9 1/2	8	5/8 3/4	3	2,360
7	·60	5/8	1,120	1/8	12 1/2	1 1/2	1 1/16	2 3/4	10 3/4	8	5/8 3/4	3 1/4	3,200
8	·639	5/8	1,280	1/8	13 1/2	1 3/4	1 1/8	2 3/4	11 3/4	8	5/8 3/4	3 1/2	4,190
9	·678	11/16	1,310	3/16	15	1 3/4	1 1/8	3	13 1/4	12	5/8 3/4	3 1/2	3,610
10	·713	3/4	1,330	3/16	16	2	1 3/16	3	14 1/4	12	3/4 7/8	3 5/8	2,970
12	·79	13/16	1,470	3/16	19	2	1 1/4	3 1/2	17	12	3/4 7/8	3 3/4	4,280
14	·864	7/8	1.600	3/16	21	2	1 3/8	3 1/2	18 3/4	12	7/8 1	4 1/4	4,280
15	·904	15/16	1,600	3/16	22 1/4	2	1 3/8	3 5/8	20	16	7/8 1	4 1/4	3,660
16	·946	1	1,600	3/16	23 1/2	2 1/4	1 7/16	3 3/4	21 1/4	16	7/8 1	4 1/4	4,210
18	1·02	1 1/16	1,690	3/16	25	...	1 9/16	3 1/2	22 3/4	16	1 1 1/8	4 3/4	4,540
20	1·09	1 1/8	1.780	3/16	27 1/2	...	1 11/16	3 3/4	25	20	1 1 1/8	5	4,490
22	1·18	1 3/16	1.850	1/4	29 1/2	...	1 13/16	3 3/4	27 1/4	20	1 1 1/4	5 1/2	4,320
24	1·25	1 1/4	1,920	1/4	31 1/2 32	...	1 1/4 1 7/8	3 3/4 4	29 1/4 29	20	1 1 1/4	5 1/2	5,130
26	1·30	1 5/16	1,980	1/4	33 3/4 34 1/4	...	1 3/8 2	3 7/8 4 1/8	31 1/4 31 3/4	24	1 1 1/4	5 3/4	5,030
28	1·38	1 3/8	2,040	1/4	36 36 1/2	...	1 7/16 2 1/16	4 4 1/4	33 1/2 34	28	1 1 1/4	6	5,000
30	1·48	1 1/2	2,000	1/4	38 38 3/4	...	1 1/2 2 1/8	4 4 3/8	35 1/2 36	28	1 1/8 1 3/8	6 1/4	4,590
36	1·71	1 3/4	1,920	1/4	44 1/2 45 3/4	...	1 3/4 2 3/8	4 1/4 4 5/8	42 42 3/4	32	1 1/8 1 3/8	6 1/2	5,790
42	1·87	2	2.100	1/4	51 52 3/4	...	1 7/8 2 5/8	4 1/8 5 3/8	48 1/2 49 1/2	36	1 1/4 1 1/2	7 1/4	5,700
48	2·17	2 1/4	2,130	1/4	57 1/2 59 1/2	..	2 2	4 3/4 5 3/4	54 3/4 56	44	1 3/8 1 1/2	7 3/4	6,090

The double sets of figures for flanges of pipe sizes above 24 in. are intended one for pressures of 200 lbs. and the other for less; this also applies to the two sizes of bolts.

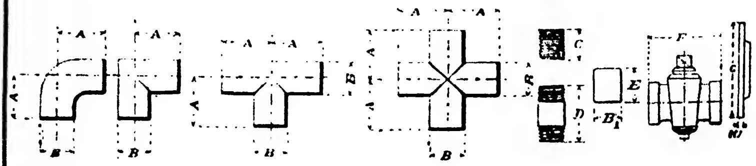
DIMENSIONS OF BRITISH STANDARD STEEL TUBES AND SOCKETS FOR SCREWING.

(See B.S. 1387—1947.)

1	2	3	4	5	6	7	8	9	10	11	12	13	14
		Outside Diameter of Black Tube.				Thicknesses.						Ordinary Sockets.*	
Nom. Bore.	Approx. outside Diam.	Class A.		Classes B and C.		Class A.		Class B.		Class C.		Approx. outside Diam.	Min. Length.
		Max.	Min.	Max.	Min.								
in.	in.	in.	in.	in.	in.	S.W.G.	in.	S.W.G.	in.	S.W.G.	in.	in.	in.
$\frac{1}{8}$	[illegible]	0·396	0·383	0·412	0·387	15	0·072	14	0·080	12	0·104	[illegible]	$\frac{3}{4}$
$\frac{1}{4}$	[illegible]	0·532	0·518	0·550	0·525	15	0·072	14	0·080	12	0·104	$\frac{3}{4}$	1
$\frac{3}{8}$	[illegible]	0·671	0·656	0·688	0·663	15	0·072	13	0·092	11	0·116	[illegible]	$1\frac{1}{8}$
$\frac{1}{2}$	[illegible]	0·841	0·825	0·859	0·834	14	0·080	12	0·104	10	0·128	$1\frac{3}{32}$	$1\frac{1}{2}$
$\frac{3}{4}$	$1\frac{1}{16}$	1·059	1·041	1·075	1·050	13	0·092	11	0·116	9	0·144	$1\frac{11}{32}$	$1\frac{5}{8}$
1	$1\frac{11}{32}$	1·328	1·309	1·351	1·320	12	0·104	10	0·128	8	0·160	[illegible]	$1\frac{7}{8}$
$1\frac{1}{4}$	$1\frac{11}{16}$	1·670	1·650	1·692	1·661	12	0·104	9	0·144	7	0·176	$2\frac{1}{32}$	$2\frac{1}{8}$
$1\frac{1}{2}$	[illegible]	1·903	1·882	1·921	1·893	11	0·116	8	0·160	6	0·192	[illegible]	$2\frac{1}{4}$
2	$2\frac{3}{8}$	2·370	2·347	2·405	2·358	11	0·116	8	0·160	6	0·192	[illegible]	$2\frac{1}{2}$
$2\frac{1}{2}$	3	2·991	2·960	3·021	2·971	10	0·128	7	0·176	5	0·212	$3\frac{7}{16}$	$2\frac{3}{4}$
3	$3\frac{1}{2}$	3·491	3·460	3·526	3·471	10	0·128	7	0·176	5	0·212	4	3
$3\frac{1}{2}$	4	3·981	3·950	4·021	3·961	9	0·144	7	0·176	5	0·212	$4\frac{1}{2}$	$3\frac{1}{4}$
4	$4\frac{1}{2}$	4·481	4·450	4·526	4·461	9	0·144	7	0·176	5	0·212	$5\frac{1}{16}$	$3\frac{1}{2}$
5	$5\frac{1}{2}$	—	—	5·536	5·461	—	—	7	0·176	5	0·212	$6\frac{1}{8}$	$3\frac{3}{4}$
6	$6\frac{1}{2}$	—	—	6·541	6·461	—	—	7	0·176	5	0·212	$7\frac{1}{4}$	$3\frac{3}{4}$

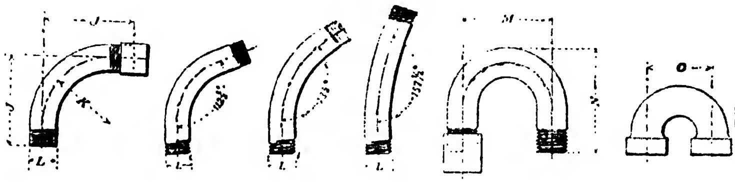
* Longscrew sockets may be $\frac{1}{8}$ in. shorter than ordinary sockets.

APPROXIMATE DIMENSIONS OF FITTINGS



		Size of Tube.	ins.	¼	⅜	½	¾
A	Centre to Face of Elbows, Tees and Crosses .	Light Weight	ins.	⅝	¾	$\frac{15}{16}$	1⅛
		Heavy Weight	,,	⅞	$\frac{17}{16}$	1	1¼
B	Outside Diam. of Elbows, Tees, and Crosses .	Light Weight	,,	⅝	¾	⅞	1$\frac{1}{16}$
		Heavy Weight	,,	$\frac{11}{16}$	$\frac{13}{16}$	$\frac{15}{16}$	1⅛
B_1	Outside Diam. of Sockets	Light Weight / Heavy Weight	,,	$\frac{19}{32}$	¾	$\frac{29}{32}$	1$\frac{3}{32}$
C	Length of Nipples . .	Light and Heavy Weight	,,	½	⅝	¾	⅞
D	Length of Barrel Nipples .	,, ,,	,,	1⅜	1¾	1⅞	2¼
E	Length of Sockets . .	,, ,,	,,	¾	1	1⅛	1½
J	Centre to Face of Bends .	,, ,,	,,	2⅛	2⅜	2⅞	3¼
K	Radii of Bends . .	,, ,,	,,	1¼	1⅜	1⅞	2¼
L	Outside Diam. of Bends and Springs . . .	,, ,,	,,	$\frac{15}{32}$	$\frac{17}{32}$	$\frac{13}{16}$	$\frac{27}{32}$
Nos. 6, 7, 8, 9	Length of Tube in Bends and Springs . . .	,, ,,	,,	3¾	4¼	4⅞	5½
M	Centre to Centre of Wrought Double Bends . .	Heavy Weight	,,	1¾	2	2⅛	3½
N	Back to Face of Wrought Double Bends . .	Heavy Weight	,,	2	2⅛	2⅜	4
O	Centre to Centre of Malleable Double Bends .	Heavy Weight	,,	—	—	—	1½

FOR SCREWED AND SOCKETED TUBES.



No. 6 No. 7 No. 8 No. 9

¾	1	1¼	1½	2	2½	3	3½	4	5	6	
1⅝ 1½	1⅝ 1¾	1 [illegible] 1 [illegible]	2 [illegible] 2 [illegible]	2½ 2⅝	3⅛ 3¼	3⅝ 3¾	4 4⅛	4½ 4¾	5¾ 5¾	6⅛ 6½	A
1 [illegible] 1⅝	1 [illegible] 1⅝	1 [illegible] 2 [illegible]	2¼ 2 [illegible]	2⅞ 2 [illegible]	3⅜ 3 [illegible]	3 [illegible] 4	4 [illegible] 4½	5 5 [illegible]	6⅛ 6⅛	7⅛ 7⅛	B
1 [illegible]	1 [illegible]	2 [illegible]	2 [illegible]	2 [illegible]	3 [illegible]	4	4½	5 [illegible]	6⅛	7¼	B_1
1	1⅛	1⅜	1⅜	1⅞	2¼	2⅜	2½	2¾	3	3¼	C
2¼	2½	3	3	3	3½	3½	4	4	4½	5	D
1⅜	1⅞	2⅛	2¼	2⅜	2¾	3	3¼	3⅜	3¾	3¾	E
4	4⅝	6	6⅝	8	9¾	11½	13¼	15	21	24½	J
2⅞	3½	4¼	5	6¼	7¾	9⅜	10⅝	12⅛	18	21	K
1 [illegible]	1 [illegible]	1 [illegible]	1 [illegible]	2⅜	3	3½	4	4½	5½	6½	L
6⅝	8	10⅛	11⅜	13¼	16¼	19	22	24¾	34¼	40	Nos. 6, 7, 8, 9
4	4½	5	6½	7½	12	14	16	18	30	36	M
4¾	5½	6	7	8¼	11½	13	14¼	15¾	22¾	27¼	N
2	2⅛	3	3⅛	4	4⅛	5	—	6	--	—	O

(*Stewarts and Lloyds, Ltd.*)

Hydraulic Test Pressure for B.S. Steel Tubes.

The test pressure for tubes of all classes is 700 lb. per sq. in.

British Standard Specification for Cast-Iron Pipes and Special Castings for Water, Gas, and Sewage*

(No. 78—1938.) (*Abstract.*)

3. The metal used for casting the pipes shall be a suitable mixture of pig iron and scrap, and shall be melted in the cupola or air-furnace, or may, if approved by the purchaser, be refined in a molten condition in a mixer of an active type. The pig iron shall be best tough grey foundry pig iron, and the scrap shall be clean, unburnt, and of good quality. There shall be no admixture of cinder iron or any material calculated to render the metal inferior in quality, and the resulting casting shall not be white or vitreous on fracture.

10. The tensile breaking strength of the metal shall be not less than 10 tons per sq. in. for a test piece of 0·875 in. diameter or less, or 9 tons per sq. in. for a test piece between 0·875 in. and 2·1 in. in diameter.

12. Straight pipes shall be cast vertically, in dry sand moulds formed from turned iron patterns and in accurately faced and truly jointed boxes, and without the use of core nails, chaplets, or thickness pieces or any substitute therefor. They shall be cast with a sufficient head of metal to ensure soundness, which head shall be afterwards cut off in a lathe and the finished pipe left the length and shape required. Straight spigot and socket pipes shall be cast socket downwards.

25. The straight pipes shall, before being coated, respectively withstand the following hydraulic test pressures without showing any leakage, sweating, or other defect:—

—	Gas.	Water & Sewage.		
	Class A.	Class B.	Class C.	Class D.
Test pressure in feet head . .	200	400	600	800

While under the test pressure each pipe shall be smartly struck with a suitable hand hammer weighing not less than one and a half pounds.

Standard Diameters, Thicknesses, and Test Pressures for Steel Pipes to be Interchangeable with Cast-Iron Pipes.

Notes.—(i) This table applies only to spigot and socket pipes which are required to be interchangeable with cast-iron pipes in accordance with B.S.S. No. 78—1938, Cast-Iron Pipes for Water, Gas and Sewage.

(ii) Where weldless pipes are ordered to be made of steel of from 35–41 tons/sq. in. tensile strength the test pressures set out in the table may, if desired by the purchaser, be increased by 40 per cent. with a maximum of 2,300 ft. head.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
		Class A.			Class B.			Class C.			Class D.		
Nom. Size.	External Diameter.	Thickness.		Test Pressure	Thickness.		Test Pressure	Thickness.		Test Pressure	Thickness.		Test Pressure
in.	in.	SWG.	in.	ft. head.	SWG.	in.	ft. head.	SWG.	in.	ft. head.	SWG.	in.	ft. head.
3	3·76	11	0·116	2100	9	0·144	2300	7	0·176	2300	6	0·192	2300
4	4·80	10	0·128	1800	9	0·144	2000	7	0·176	2300	6	0·192	2300
5	5·90	9	0·144	1600	8	0·160	1800	7	0·176	2000	6	0·192	2200
6	6·98	8	0·160	1500	7	0·176	1700	6	0·192	1900	5	0·212	2100
7	8·06	7	0·176	1500	6	0·192	1600	5	0·212	1800	—	0·250	2100
8	9·14	7	0·176	1300	6	0·192	1400	5	0·212	1600	—	0·250	1900
9	10·20	6	0·192	1200	5	0·212	1400	—	0·250	1600	—	0·281	1900
10	11·26	6	0·192	1100	5	0·212	1200	—	0·250	1500	—	0·281	1700
12	13·14	6	0·192	950	5	0·212	1000	—	—	—	—	—	—
12	13·60	—	—	—	—	—	—	—	0·250	1200	—	0·281	1400

* By permission of the British Standards Institution.

British Standard Cast-Iron Pipes for Water, Gas, and Sewage.

Standard Thicknesses and External Diameters for Spigot and Socket and Flange Straight Pipes.

Nominal Internal Diameter of Pipe (ins.)	Gas.		Water and Sewage.						Nominal Internal Diameter of Pipe (ins.)
	Class A.		Class B.		Class C.		Class D.		
	Test Pressure 200 ft. hd.		Test Pressure 400 ft. hd.		Test Pressure 600 ft. hd.		Test Pressure 800 ft. hd.		
	Thickness. (ins.)	External Diameter. (ins.)	Thickness. (ins.)	External Diameter. (ins.)	Thickness. (ins.)	External Diameter. (ins.)	Thickness. (ins.)	External Diameter. (ins.)	
3	·38	3·76	·38	3·76	·38	3·76	·40	3·76	3
4	·39	4·80	·39	4·80	·40	4·80	·46	4·80	4
5	·41	5·90	·41	5·90	·45	5·90	·52	5·90	5
6	·43	6·98	·43	6·98	·49	6·98	·57	6·98	6
7	·45	8·06	·45	8·06	·53	8·06	·61	8·06	7
8	·47	9·14	·47	9·14	·57	9·14	·65	9·14	8
9	·49	10·20	·49	10·20	·60	10·20	·69	10·20	9
10	·52	11·26	·52	11·26	·63	11·26	·73	11·26	10
12	·55	13·14	·57	13·14	·69	13·60	·80	13·60	12
14	·57	15·22	·61	15·22	·75	15·72	·86	15·72	14
15	·59	16·26	·63	16·26	·77	16·78	·89	16·78	15
16	·60	17·30	·65	17·30	·80	17·84	·92	17·84	16
18	·63	19·38	·69	19·38	·85	19·96	·98	19·96	18
20	·65	21·46	·73	21·46	·89	22·06	1·03	22·06	20
21	·67	22·50	·75	22·50	·92	23·12	1·06	23·12	21
22	·68	23·54	·77	23·54	·94	24·16	1·08	24·16	22
24	·71	25·60	·80	25·60	·98	26·26	1·13	26·26	24
26	·74	27·66	·83	27·66	1·02	28·36	1·18	28·36	26
27	·75	28·70	·85	28·70	1·04	29·40	1·20	29·40	27
28	·76	29·72	·86	29·72	1·06	30·44	1·22	30·44	28
30	·79	31·78	·89	31·78	1·09	32·52	1·26	32·52	30
32	·82	33·84	·92	33·84	1·13	34·62	1·31	34·62	32
33	·83	34·88	·94	34·88	1·15	35·66	1·33	35·66	33
36	·87	37·96	·98	37·96	1·20	38·76	1·38	38·76	36
38	·90	40·02	1·01	40·02	1·23	40·84	1·42	40·84	38
40	·92	42·06	1·03	42·06	1·26	42·92	1·46	42·92	40
42	·95	44·12	1·06	44·12	1·30	45·00	1·50	45·00	42
44	·98	46·16	1·08	46·16	1·33	47·06	1·53	47·06	44
46	1·00	48·22	1·11	48·22	1·36	49·14	1·57	49·14	46
48	1·03	50·26	1·13	50·26	1·38	51·20	1·60	51·20	48

British Standard Steel Spigot and Socket Pipes for Water, Gas and Sewage.

(B.S.S. No. 534—1934.)

Standard Diameters, Thicknesses and Test Pressures.

Notes.—Weldless pipes of the type dealt with in this Specification are not at present manufactured larger than 15 ins. nominal size.

When weldless pipes are ordered to be made of steel of from 35–41 tons sq. in. tensile strength, the test pressures set out in the table may, if desired by the purchaser, be increased by 40 per cent., with a maximum of 2,300 ft. head.

Reference should be made to the Table on p. 114 for pipes which are required to be interchangeable with cast-iron pipes in accordance with B.S.S. No. 78—1938, Cast-Iron Pipes for Water, Gas and Sewage.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Nom. Size.	External Diameter			Class A.			Class B.			Class C.			Class D.		
				Thickness.		Test Pressure	Thickness.		Test Pressure	Thickness.		Test Pressure	Thickness.		Test Pressure
in.	in.			SWG.	in.	ft. head.	SWG.	in.	ft. head.	SWG	in.	ft. head.	SWG.	in.	ft. head.
2	2·375	2·25	—	12	0·104	2300	11	0·116	2300	10	0·128	2300	9	0·144	2300
2½	3	2·81	—	11	0·116	2300	10	0·128	2300	9	0·144	2300	8	0·160	2300
3	3·5	3·31	—	11	0·116	2300	9	0·144	2300	7	0·176	2300	6	0·192	2300
4	4·5	4·31	—	10	0·128	1900	9	0·144	2200	7	0·176	2300	6	0·192	2300
5	5·5	5·375	—	9	0·144	1800	8	0·160	2000	7	0·176	2200	6	0·192	2300
6	6·5	6·375	—	9	0·144	1500	7	0·176	1800	6	0·192	2000	5	0·212	2300
7	7·5	—	—	7	0·176	1600	6	0·192	1700	5	0·212	1900	—	0·250	2300
8	8·5	—	—	7	0·176	1400	6	0·192	1500	5	0·212	1700	—	0·250	2000
9	9·5	—	—	6	0·192	1300	5	0·212	1500	—	0·250	1800	—	0·281	2000
10	10·5	—	—	6	0·192	1200	5	0·212	1300	—	0·250	1600	—	0·281	1800
12	12·5	12·75	—	6	0·192	1000	5	0·212	1100	—	0·250	1300	—	0·281	1500
14	14·5	14·75	15·0	6	0·192	750	—	0·219	900	—	0·250	1100	—	0·281	1200
15	15·5	15·75	16·0	6	0·192	700	—	0·219	800	—	0·250	1000	—	0·281	1100
16	16·5	—	—	6	0·192	650	—	0·219	750	—	0·250	900	—	0·281	1000
18	18·5	—	—	6	0·192	550	—	0·219	700	—	0·250	800	—	0·313	1000
20	20·5	—	—	—	0·219	600	—	0·250	700	—	0·281	850	—	0·313	950
21	21·5	—	—	—	0·219	600	—	0·250	700	—	0·281	800	—	0·313	900
22	22·5	—	—	—	0·250	650	—	0·281	750	—	0·313	850	—	0·375	1000
24	24·5	—	—	—	0·250	600	—	0·313	800	—	0·344	900	—	0·375	950
26	26·75	—	—	—	0·281	650	—	0·313	700	—	0·344	800	—	0·375	900
27	27·75	—	—	—	0·281	600	—	0·313	700	—	0·344	800	—	0·375	900
28	28·75	—	—	—	0·313	650	—	0·344	750	—	0·375	800	—	0·406	900
30	30·75	—	—	—	0·313	600	—	0·344	700	—	0·375	750	—	0·406	900
32	32·75	—	—	—	0·313	600	—	0·375	700	—	0·406	750	—	0·438	900
33	33·75	—	—	—	0·313	550	—	0·375	650	—	0·406	750	—	0·438	900
36	36·75	—	—	—	0·313	500	—	0·375	600	—	0·406	700	—	0·438	900
38	39	—	—	—	0·375	600	—	0·406	650	—	0·438	700	—	0·500	900
40	41	—	—	—	0·375	550	—	0·406	600	—	0·438	700	—	0·500	900
42	43	—	—	—	0·375	500	—	0·406	600	—	0·438	700	—	0·500	900
44	45	—	—	—	0·375	500	—	0·438	600	—	0·500	700	—	0·563	900
46	47	—	—	—	0·375	500	—	0·438	550	—	0·500	700	—	0·563	900
48	49	—	—	—	0·375	450	—	0·438	550	—	0·500	700	—	0·563	900
50	51	—	—	—	0·438	550	—	0·500	600	—	0·563	700	—	0·625	900
52	53	—	—	—	0·438	500	—	0·500	600	—	0·563	700	—	0·625	900
54	55	—	—	—	0·438	500	—	0·500	550	—	0·563	700	—	0·625	900
56	57	—	—	—	0·500	550	—	0·563	650	—	0·625	700	—	0·688	900
58	59	—	—	—	0·500	550	—	0·563	600	—	0·625	700	—	0·688	900
60	61	—	—	—	0·500	500	—	0·563	600	—	0·625	700	—	0·688	900
63	64·25	—	—	—	0·563	500	—	0·625	600	—	0·688	700	—	0·750	900
66	67·25	—	—	—	0·563	500	—	0·625	600	—	0·750	700	—	0·813	900
69	70·25	—	—	—	0·625	500	—	0·688	600	—	0·750	700	—	0·875	900
72	73·25	—	—	—	0·625	500	—	0·688	600	—	0·813	700	—	0·875	900

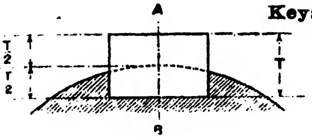


FIG. 21.

Keys and Keyways.

BRITISH STANDARD KEYS.

(No. 46—Part I—1929.) (*Abstract.*)

The depth of immersion of the key on the centre line (A–B) is calculated so as to give approximately half and half immersion at the sides.

In 1929 the British Standards Institution issued a revised specification, No. 46, Part I, for keys and keyways. The Specification includes exhaustive tables for Rectangular Parallel, Square Parallel, Rectangular and Square Taper Keys, Gib-headed Keys, Woodruff Keys, etc. The tables are far too long to reprint and users are advised to consult them. It is, however, to be noted particularly that the depth of immersion of a rectangular key is no longer half and half at the tangent of the shaft as was the old practice, shown in the sketch, but as near as possible half and half at the edge of the keyway. A few typical dimensions are tabulated below.

RECTANGULAR PARALLEL KEYS.

Designation.	Shaft Diam.		Key.		Keyway.		
						Depth.	
B.S.K.B.	Over	Up to	Width.	Thickness.	Width.	Shaft.	Hub.
$\frac{5}{16}$	1	$1\frac{1}{4}$	0·3125	0·2187	0·3125	0·1325	0·0883
$\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{3}{4}$	0·4375	0·28125	0·4375	0·1721	0·1122
$\frac{9}{16}$	2	$2\frac{1}{4}$	0·5625	0·375	0·5625	0·2269	0·1511
$\frac{11}{16}$	$2\frac{1}{2}$	$2\frac{3}{4}$	0·6875	0·46875	0·6875	0·2822	0·1906
$\frac{7}{8}$	3	$3\frac{1}{4}$	0·875	0·625	0·875	0·3746	0·2544
$1\frac{3}{8}$	5	$5\frac{1}{2}$	1·375	0·9375	1·375	0·5629	0·3796
$1\frac{7}{8}$	7	$7\frac{1}{2}$	1·875	1·25	1·875	0·7513	0·5047
3	11	12	3	2	3	1·2031	0·8049

FOUR AND SIX SPLINES.

Shafts with four and six splines are covered by tables in Specification No. 46, Part 2—1929. These tables begin at $\frac{1}{2}$-in. diameter over splines and go up to 6 in. diameter. The fit is on the minor diameter, but throughout the entire range of sizes the whole dimensions and the limits on the shaft diameters for the various fits are in accordance with British Standard Limits and Fits for Engineering (*B.S. Report*, No. 164—1924), whereas the splines standardised for the automobile industry are based on the standards of the American Society of Automotive Engineers.

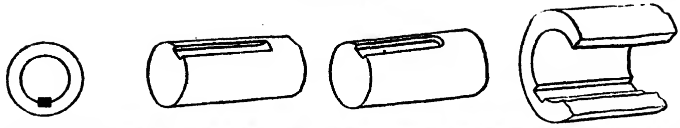
In another Table in No. 46, Part 2, tentative standards for multiple spline shafts and for serrated shafts are given. (See also B.S. Aircraft Standard A20—1946.)

CONED AND KEYED SHAFTS.

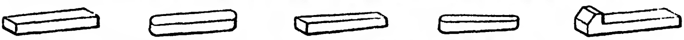
The standard taper of the cone is 1 in 10. The key is parallel to the side of the cone, and its size corresponds with the diameter at the larger end of the cone. The length of the key should not be less than one and a half times the larger diameter of the cone.

TAPER PINS. No. 46, Part 3.—1935.

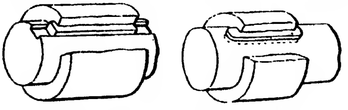
The taper is 1 in 48 or $\frac{1}{4}$ in. per ft. In split taper pins the split should be one-fifth of the length.



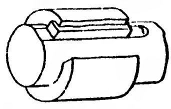
1 AND 2—*Key.*—A key is a piece inserted between the joint of two parts to prevent relative movement. For the purpose of this specification a key is defined as a piece inserted in an axial direction between a shaft and a hub to prevent relative rotation. *Keyway.*—A recess in a shaft or hub to accommodate a key.



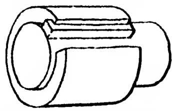
3 TO 7—*Plain Parallel Key.*—A key of rectangular or square section; uniform in thickness and width and with squared ends. *Round-ended Parallel Key.*—A plain parallel key with ends rounded as to width. *Plain Taper Key.*—A key of rectangular or square section uniform in width; tapered in thickness and with squared ends. *Round-ended Taper Key.*—A plain taper key as defined above, but with ends rounded as to width. *Gib-head Key.*—A key provided with a head to facilitate withdrawal.



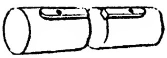
8—*Sunk Keys.*—A key partly in the shaft and partly in the hub.



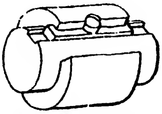
9—*Flat Saddle Key.*—A taper key fitting a keyway in the hub and a flat on the shaft.



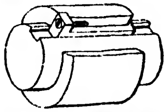
10—*Hollow Saddle Key.*—A taper key fitting a keyway in the hub, the bottom of the key being formed to fit the cylindrical surface of the shaft.



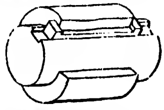
11—*Feather Key.*—A key attached to one member of a pair and permitting relative axial movement.



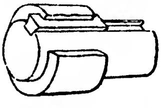
12—*Peg Feather Key.*—A feather key with a peg to prevent its relative movement in the part to which it is attached.



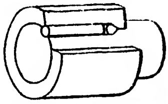
13—*Single Head Feather Key.*—A feather key with a projection at the end to prevent its axial movement in hub.



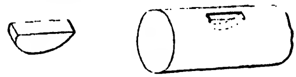
14—*Double Head Feather Key.*—A feather key with a projection at each end to prevent its axial movement in hub.



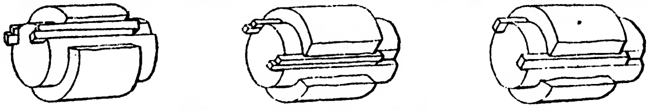
15—*Dovetail Key.*—A key having one portion of its cross section of dovetail form.



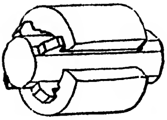
16—*Round Key.*—A key of circular section fitted in a hole drilled partly in the shaft and partly in the hub.



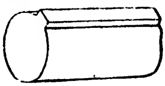
17 AND 18—*Woodruff Key and Keyway.*—A key in the form of a segment of a parallel-sided circular disc and a segmental recess in a shaft to accommodate a Woodruff key.



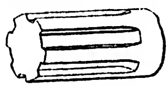
19—*Tangential Keying.*—The use of keys acting tangentially in opposite directions.



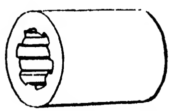
20—*Stake Keying.*—The use of taper keys for centralising the hub on the shaft and for preventing relative movement. They may be hollow or flat saddle keys.



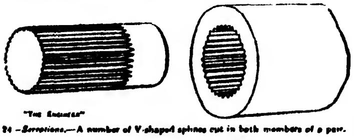
21—*Spline.*—A key-like projection integral with a member of a pair.



22—*Spline Shaft.*—A shaft having one or more splines. Spline shafts are designated by the number of splines, e.g., six-spline shaft.

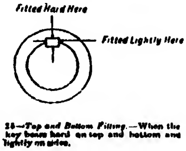


23—*Spline Hub.*—A hub having one or more splines.

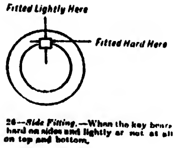


"The Engineer"

24—*Serrations.*—A number of V-shaped splines cut in both members of a pair.



25—*Top and Bottom Fitting.*—When the key bears hard on top and bottom and lightly on sides.



26—*Side Fitting.*—When the key bears hard on sides and lightly or not at all on top and bottom.

SECTION V

PART II

WEIGHT OF MATERIALS.

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

A.—DENSITY AND SPECIFIC GRAVITY.

Definitions.—Density is mass per unit volume. In the C.G.S. system this is expressed in grammes per millilitre, and the F.P.S. system, in lb. per cub. foot.

Specific gravity is the ratio of the mass of a given volume to the mass of the same volume of some reference substance (usually water or air) at a fixed reference temperature and pressure. The reference conditions for water are 62° F. and 30 inches of mercury.

RELATION BETWEEN VOLUME AND WEIGHT OF WATER.

Detail.	Fresh.	Sea.
Greatest density at	39·2° F.	Freezing-point
1 cubic foot at 62° F. weighs . . .	62·288 lbs.	64 lbs.
1 cubic inch „ „	0·036047 lb.	0·037037 lb.
1 cubic foot „ equals	6·2288 gals.	
1 gallon 62° at F. weighs	10 lbs.	10·3 lbs.
1 ton equals	35·962 cub. ft.	35 cub. ft.
1 ton contains	224 gals.	217 gals.
Freezes at	32° F.	27·4° F.

From 4° C. water expands regularly as the temperature rises; so that 1,000 parts at 0° C. become at 0° = 1,000·00, at 4° = 999·88, at 10° = 1,000·14, at 20° = 1,001·67, at 30° = 1,004·21, at 40° = 1,007·61, at 50° = 1,011·93, at 60° = 1,016·86, at 70° = 1,022·43, at 80° = 1,028·73, at 90° = 1,035·58, at 100° = 1,043·03.

DENSITY OF ATMOSPHERIC AIR.

1 cubic foot of dry atmospheric air at 62° F. and 30 inches of mercury weighs 532·5 grains (7,000 grains = 1 lb. avoir.).

To find Specific Gravity.

Solids.

W = weight of a body in the air,
S = specific gravity of the body,
W = weight of the body (heavier than water) immersed in water;

then,

$$S = \frac{W}{W - \mathit{W}};$$

When the body is lighter than water, annex to it a heavier body that is able to sink the lighter one.

S = specific gravity of the heavier annexed body;
s = specific gravity of the lighter body;
W = weight of the two bodies in air;
W = weight of the two bodies in water;
w = weight of the heavier body in air;
w = weight of the lighter body in air;

then,

$$s = \frac{w}{W - \mathit{W} - \frac{w}{S}}$$

A simple way to obtain the specific gravity of a wood is to form it into a parallel rod, and place it vertically in water; then, when in equilibrium, the immersed end is to the whole rod as the specific gravity is to 1.

Liquids.

Measure the weight of a stoppered-glass bottle; then fill the same with distilled water and weigh it again; the difference between the two measurements gives the weight of the distilled water. Then completely empty out the water and refill the bottle with the liquid whose specific gravity is required and again weigh; this weight less the weight of the bottle gives the weight of the liquid; this weight divided by the weight of the water gives the specific gravity of the liquid.

In practice it is convenient to employ a bottle (called a specific gravity bottle) that holds exactly 10 grammes of distilled water at 4° C., because, when such a bottle is filled with the liquid under trial the weight in centigrammes of the liquid taken at 0° C. represents the specific gravity at once, without calculation. For convenience a counterpoise of brass is adjusted to the weight of the bottle.

Gases.

Measure in the same way as liquids, taking Air as the standard. A large and very light, *i.e.* thin, glass vessel must be used for the purpose, otherwise the weight of the glass will swamp, as it were, the weight of the enclosed gas, and accurate results cannot be obtained. Certain corrections have to be made in practice, for great accuracy.

Atomic Weights and Specific Gravities of the Chemical Elements.

Name.	Symbol.	Atomic Weight.	Specific Gravity.	Name.	Symbol.	Atomic Weight.	Specific Gravity.
Aluminium	Al	26·97	2·70	Holmium	Ho	164·9	—
Antimony	Sb	121·76	6·68	Hydrogen	H	1·008	0·0000899
Argon	A	39·94	0·001782	Indium	In	114·8	7·3
Arsenic	As	74·91	5·7	Iodine	I	126·92	4·93
Barium	Ba	137·36	3·5	Iridium	Ir	193·1	22·4
Bismuth	Bi	209·0	9·80	Iron	Fe	55·85	7·86
Boron	B	10·82	2·5	Krypton	Kr	83·7	0·00371
Bromine	Br	79·92	3·12	Lanthanum	La	138·92	6·15
Cadmium	Cd	112·41	8·6	Lead	Pb	207·21	11·34
Cæsium	Cs	132·91	1·90	Lithium	Li	6·94	0·53
Calcium	Ca	40·08	1·55	Lutetium	Lu	175·0	—
Carbon	C	12·01	Diamond 3·51	Magnesium	Mg	24·32	1·74
				Manganese	Mn	54·93	7·4
			Graphite 2·25	Mercury	Hg	200·6	13·55
				Molybdenum	Mo	95·95	10·2
Cerium	Ce	140·13	6·90	Neodymium	Nd	144·3	6·9
Chlorine	Cl	35·46	0·00321	Neon	Ne	20·2	0·000900
Chromium	Cr	52·01	7·1	Nickel	Ni	58·69	8·90
Cobalt	Co	58·94	8·9	Niton (radium emanation)	Nt	222	—
Columbium (*Niobium*)	Cb	92·91	8·4	Nitrogen	N	14·01	0·00125
Copper	Cu	63·57	8·92	Osmium	Os	190·2	22·48
Dysprosium	Dy	162·5	—	Oxygen	O	16·00	0·001429
Erbium	Er	167·2	—	Palladium	Pd	106·7	12·0
Europium	Eu	152·0	—	Phosphorus	P	30·98	Yellow 1·82
Fluorine	F	19·0	0·001695				Red 2·20
Gadolinium	Gd	156·9	—	Platinum	Pt	195·2	21·45
Gallium	Ga	69·7	5·91	Potassium	K	39·10	0·86
Germanium	Ge	72·6	5·46	Praseodymium	Pr	140·9	6·5
Glucinium (*Beryllium*)	Gl	9·02	1·8	Radium	Ra	226·05	5
				Rhodium	Rh	102·9	12·5
Gold	Au	197·2	19·3	Rubidium	Rb	85·48	1·53
Hafnium	Hf	178·6	—	Ruthenium	Ru	101·7	12·2
Helium	He	4·00	0·000178	Samarium	Sa,Sm	150·4	7·7

ATOMIC WEIGHTS AND SPECIFIC GRAVITIES OF THE CHEMICAL ELEMENTS (*continued*).

Name.	Symbol.	Atomic Weight.	Specific Gravity.	Name.	Symbol.	Atomic Weight.	Specific Gravity.
Scandium	Sc	45·1	2·5	Thulium	Tu	169·4	—
Selenium	Se	78·96	4·8	Tin	Sn	118·70	White 7·31 Grey 5·75
Silicon	Si	28·1	2·4				
Silver	Ag	107·88	10·5	Titanium	Ti	47·9	4·5
Sodium	Na	23·0	0·97	Tungsten	W	183·92	19·3
Strontium	Sr	87·63	2·6	Uranium	U	238·07	18·7
Sulphur	S	32·06	2·0	Vanadium	V	50·95	5·96
Tantalum	Ta	180·88	16·6	Xenon	Xe	131·3	0·00585
Tellurium	Te	127·6	6·24	Ytterbium	Yb	173·04	—
Terbium	Tb	159·2	—	Yttrium	Y, Yt	88·9	3·8
Thallium	Tl	204·4	11·85	Zinc	Zn	65·38	7·14
Thorium	Th	232·12	11·2	Zirconium	Zr	91·2	6·4

N.B.—Specific gravities given above are those for the state of the element at normal temperatures (*i.e.* gas, liquid or solid).

SPECIFIC GRAVITIES OF METALS AND ALLOYS OF COMMERCE.

Metal.	Specific Gravity.	Lbs. per cu. ft.	Metal.	Specific Gravity	Lbs. per cu. ft.
Aluminium, wrought	2·70	168	Iron, cast	6·90 to 7·50	430 to 467
,, cast	2·56	160			
Brass, cast	8·10	505	Iron, wrought	7·40 to 7·80	461 to 486
,, sheet, 75%–25%	8·45	527			
,, yellow, 66%–34%	8·40	518	Nickel, malleable	8·85	551
Muntz metal, 60%–40%	8·20	511	Platinum, hammered	21·5	1,340
Bronze (gunmetal)			,, wire	21·4	1,334
84% Cu, 16% Sn	8·80	548	Silver, cast	10·50	655
70% Cu, 30% Sn	8·84	551	,, drawn	10·44	651
88% Cu, 8% Sn, 4% Zn.	8·5	529	Steel, carbon	7·7 to 7·9	482 to 493
Copper, Cast	8·85	551	,, chromium	7·7	482
,, Drawn	8·89	554	,, nickel-chromium	7·84	488
Gold, pure cast	19·30	1,200	Lead-antimony 92%–8%	10·7	667
,, cold rolled	19·30	1,200	Lead-tin-antimony		
,, 22-carat	17·50	1,090	80%–5%–15%	10·04	631
,, 15-carat	13·60	847	Tin-antimony-copper		
Iridium, hammered	22·4	1,395	91%–4½%–4½%	7·34	457

To Discover the Adulteration in Metals, or to find the Proportions of Two Ingredients in a Compound.

$$w = \frac{W - s(W - W')}{1 - \frac{s}{S}}.$$

EXAMPLE.—A metal compounded of silver and gold weighs, $W=6$ pounds in the air, and in water, $W'=5{\cdot}636$ pounds. Required the proportions of silver and gold.

$S=19{\cdot}36$ specific gravity of gold. $s=10{\cdot}51$ specific gravity of silver.

$$\text{Weight } w = \frac{6 - 10{\cdot}51(6 - 5{\cdot}636)}{1 - \frac{10{\cdot}51}{19{\cdot}36}} = 4{\cdot}755 \text{ pounds of gold.}$$

and 1·245 pounds of silver.

SPECIFIC GRAVITIES OF VARIOUS SOLIDS AND LIQUIDS.

Substance.	Sp. Gr.	Weight per Cubic Foot.	Substance.	Sp. Gr.	Weight per Cubic Foot.	Substance.	Sp. Gr.	Weight per Cubic Foot.
Minerals								
		Lbs.			Lbs.			Lbs.
Agate	2·59	161	Cornelian	2·61	163	Peat	0·60 / 1·33	37 / 83
Alabaster, white	2·73	170	Diamond, Oriental	3·52	219	Plaster of Paris	1·18	73
,, yellow	2·70	168	,, Brazilian	3·44	214	Plumbago	2·10	131
Alum	1·71	106	Emery	4·00	249	Porcelain, China	2·30	143
Amber	1·08	67	Flint, black	2·58	161	Porphyry, red	2·77	173
Ambergris	0·87	54	,, white	2·50	156	Pumice-stone	0·92	57
Asbestos, starry	3·07	191	Garnet	4·19	261	Quartz	2·66	166
Asphaltum	0·91 / 1·65	57 / 103	,, black	3·75	233	Red-lead	8·94	557
Barytes, sulphate	4·00 / 4·87	249 / 303	Glass, bottle	2·73	170	Resin	1·09	68
Basalts	2·74 / 2·86	171 / 178	,, Crown	2·49	155	Rockcrystal	2·74	171
Borax	1·71	106	,, flint	2·93 / 3·20	182 / 199	Rotten-stone	1·98	123
Brick	1·90 / 1·37	118 / 85	,, green	2·64	164	Ruby	4·28	267
,, fire	2·20	137	,, optical	3·45	215	Salt, common	2·13	133
,, (pressed) in cement	2·17	135	,, white	2·89	180	Saltpetre	2·09	130
,, work in mortar	1·80 / 2·00	112 / 125	,, window	2·64	164	Sapphire	3·99	248
Cement, Portland	3·0	187	Granite, Scotch	2·63	164	Schorl	3·17	197
,, Roman	1·56	97	Grindstone	2·14	133	Shale	2·60	162
Chalk	1·52 / 2·78	95 / 173	Gypsum, opaque	2·17	135	Slate	2·90 / 2·67	181 / 166
Chrysolite	2·78	173	Hone, white, razor	2·88	179	Smalt	2·44	152
Clay	1·93	120	Hornblende	3·54	220	Spar, calcareous	2·74	171
,, with gravel	2·48	154	Jet	1·30	81	,, Feld	2·69	168
Coal, anthracite	1·44 / 1·64	90 / 102	Lime, hydraulic	2·75	171	,, Fluor	3·40	212
,, Cannel	1·24 / 1·32	77 / 82	,, quick-	0·80	50	Stalactite	2·42	151
,, Derbyshire	1·29	80	Limestone, green	3·18	198	Stone, Bath	1·96	122
,, Newcastle	1·27	79	,, white	3·16	197	,, Basalt	2·63	164
,, Scotch	1·26 / 1·30	79 / 81	Magnesia, carbonate	2·40	149	,, Bristol	2·51	156
,, Wales, mean	1·32	82	Marble, Carrara	2·72	169	,, Caen, Normandy	2·07	129
Coke	1·00	62	,, common	2·69	168	,, Craigleith	2·32	144
Concrete, mean	2·24	140	,, French	2·65	165	,, Kentish rag	2·65	165
Copal	1·05	65	Mica	2·80	174	,, Portland	2·37	148
Coral, red	2·70	168	Millstone	2·48	154	,, sandstone	2·20	137
,, white	2·55	159	Mortar	1·38 / 1·75	86 / 109	Sulphur, native	2·03	126
			Mud	1·63	102	Talc, mean	2·50	156
			Nitre	1·90	118	,, black	2·90	181
			Opal	2·11	131	Tile	1·82	113
			Oyster-shell	2·09	130	Topaz, Oriental	4·01	250
			Paving-stone	2·42	151	Trap	2·72	169
			Pearl, Oriental	2·65	165	Turquoise	2·75	171
Miscellaneous Solids.								
		Lbs.			Lbs.			Lbs
Atmospheric air	·00121	·075	Gunpowder, solid	1·55 / 1·80	97 / 112	Opium	1·34	83
Beeswax	0·97	60	Gutta-percha	0·98	61	Rubber	0·92	57
Butter	0·94	59	Horn	1·69	105	Soap, Castile	1·07	67
Camphor	0·99	62	Ice, at 32°	0·92	57	Spermaceti	0·94	59
Ebonite	1·2 / 1·6	75 / 100	Indigo	1·01	63	Starch	1·53	95
Egg	1·09	68	Isinglass	1·11	69	Sugar	1·61	100
Fat, animal	0·92	57	Ivory	1·83	114	,, 0·66	1·33 / 0·97	83 / 60
Gamboge	1·22	76	Lard	0·95	59	Tallow	0·94	59
Gum Arabic	1·45	90	Mastic	1·07	67	Wax, paraffin	0·87 / 0·93	54 / 58
			Myrrh	1·36	85			

Specific Gravities of Various Solids and Liquids (*continued*).

Liquids.

Substance.	Sp. Gr.	Weight per Cubic Foot.	Substance.	Sp. Gr.	Weight per Cubic Foot.	Substance.	Sp. Gr.	Weight per Cubic Foot.
		Lbs.			Lbs.			Lbs.
Acid, acetic	1·05	65	Alcohol, proof spirit, 50 per cent. 80°	0·88	55	Oil, olive	0·92	57
„ benzoic, solid	1·27	79	Ammonia, 27·9 per cent.	0·89	55	„ palm	0·93	58
„ citric, solid	1·54	96	Aquafortis, double	1·30	81	„ petroleum	0·88	55
„ oxalic „	1·65	103	„ single	1·20	75	„ rape	0·91	57
„ hydrochloric	1·20	75	Beer	1·03	64	„ sunflower	0·93	58
„ hydrofluoric	0·99	62	Bitumen, liquid	0·85	53	„ turpentine	0·87	54
„ nitric	1·50	93	Blood (human)	1·05	65	„ whale	0·92	57
„ phosphoric, solid	1·84	115	Brandy, $\frac{5}{8}$ of spirit	0·92	57	Spirit, rectified	0·82	51
„ sulphuric	1·85	115	Cider	1·02	64	Tar	1·02	64
Alcohol, pure, 60°	0·79	49	Ether, acetic	0·87	54	Vinegar	1·08	67
„ 95 per cent.	0·81	50	„ muriatic	0·85	53	Water, Dead Sea	1·24	77
„ 80 „	0·85	53	„ sulphuric	0·72	45	„ 60°	1·00	62
„ 50 „	0·91	57	Glycerine	1·28	79·5	„ 212°	0·96	59
„ 40 „	0·94	59	Honey	1·45	90	„ distilled, 39°	1·00	62
„ 25 „	0·96	60	Milk	1·03	64	„ Mediterranean	1·03	64
„ 10 „	0·98	61	Oil, aniseed	0·99	62	„ rain	1·00	62
„ 5 „	0·99	62	„ codfish	0·92	57	„ sea	1·03	64
„ proof spirit, 50 per cent., 60°	0·93	58	„ linseed	0·94	59	Wine, Burgundy	0·99	62
			„ naphtha	0·85	53	„ Champagne	1·00	62
						„ Madeira	1·04	65
						„ port	1·00	62

Timbers in the Air-dry Condition (15 per Cent. Moisture Content.)

(For descriptions of these timbers, see Section IX, Part I.)

Substance.	Sp. Gr.	Weight per Cubic Foot.	Substance.	Sp. Gr.	Weight per Cubic Foot.	Substance.	Sp. Gr.	Weight per Cubic Foot.
		Lbs.			Lbs.			Lbs.
Alder, common or black	0·53	33	Cedar, Borneo (see Seraya)			Elm rock	0·82	51
Apitong (Bagac)	0·70	44	Cedar, Cen. American (cigar-box cedar)	0·48	30	„ wych	0·69	43
Ash, European	0·70	44	Cedar, Port Orford	0·50	31	Eng (In)	0·88	55
„ American white	0·67	42	„ Western red	0·38	24	Fir, Douglas (Oregon pine, Columbian pine)	0·53	33
Bagac (see Apitong)			Chestnut, Sweet or Spanish	0·56	35	Fir, silver	0·48	30
Balsa	0·16	10	Chuglam, White	0·69	43	Gaboon	0·43	27
Basswood	0·42	26	Crabwood (Andiroba mahogany)	0·69	43	Greenheart	1·04	65
Beech, European	0·74	46	Deal, red or yellow (see Redwood, Baltic, and Pine, Scots)			Gurjun	0·74	45
Birch, European	0·67	42	Deal, white (see Whitewood, Baltic)			Hemlock, Western	0·50	31
„ Canadian yellow	0·70	44	Douglas fir (see Fir, Douglas)			Hickory	0·82	51
Black Bean	0·75	47	Ebony	1·19	74	Hornbeam	0·74	46
Blackwood, Australian	0·70	44	Elm, Common English	0·56	35	Horse Chestnut	0·51	32
Boxwood	0·93	58				In (see Eng)		
Camphor, Borneo (Kapur)	0·78	49				Iroko (Mvule)	0·66	41
Camphorwood, East African	0·58	36				Jarrah	0·85	53
						Kapur (see Camphorwood, Borneo)		
						Karri	0·93	58

TIMBERS IN THE AIR-DRY CONDITION (15 PER CENT. MOISTURE CONTENT) (*continued*).

Substance.	Sp. Gr.	Weight per Cubic Foot. Lbs.
Kauri, New Zealand	0·61	38
Kauri, Queensland	0·48	30
Keruing	0·78	49
Larch	0·59	37
Lauan	0·56	35
Laurelwood, Indian	0·86	54
Lignum vitae	1·25	78
Lime, American (see Basswood)		
Lime, Common European	0·56	35
Magnolia	0·56	35
Mahogany, African	0·56	35
Mahogany, Andiroba (see Crabwood)		
,, Cuban	0·69	43
,, Gaboon (see Gaboon)		
,, Honduras	0·55	34
,, Sapele	0·64	40
Mansonia	0·61	38
Maple, Rock or Sugar	0·74	46
Matai	0·62	39
Meranti	0·56	35
Mora	1·03	64
Mvule (see Iroko)	0·74	46
Oak, American white	0·77	48
,, American red	0·72	45
,, European	0·74	46
,, Japanese	0·67	42
,, silky	0·58	36
Oak, Tasmanian	0·72	45
Obeche (African Whitewood)	0·38	24
Okoume (see Gaboon)		
Olive, East African	0·90	56
Oregon pine (see Fir, Douglas)		
Padauk, Andaman	0·78	49
,, Burma	0·85	53
Pine, Columbian (see Fir, Douglas)		
,, Corsican	0·51	32
,, Jack	0·50	31
,, Kauri (see Kauri)		
,, Oregon (see Fir, Douglas)		
,, Parana	0·55	34
,, Pitch	0·66	41
,, red (Canada)	0·53	33
,, Scots (see also Redwood, Baltic)		
,, Siberian	0·42	26
,, Sugar	0·43	27
,, Western white	0·45	28
,, yellow or white (Canada)	0·42	26
Poplar	0·45	28
,, Yellow (see Whitewood, American)		
Pyinkado	0·99	62
Redwood, Baltic	0·50	31
Redwood, Californian (see Sequoia)		
Rosewood, Indian	0·86	54
Sapele Mahogany (see Mahogany, Sapele)		
Sequoia or Californian Redwood	0·46	29
Seraya (Borneo cedar)	0·56	35
Silver fir, (see Fir, Silver)		
Spruce, European (see Whitewood, Baltic)		
Spruce, Canadian	0·45	28
,, Sitka	0·45	28
Sycamore	0·62	39
Teak	0·66	41
,, Rhodesian	0·91	57
Turpentine	0·95	59
Walnut, African	0·56	35
,, American black	0·66	41
,, European	0·66	41
,, Queensland	0·74	46
Whitewood, African (see Obeche)		
Whitewood, American (Canary Whitewood, Yellow Poplar)	0·50	31
Whitewood, Baltic	0·46	29
Willow	0·45	28
Yew	0·67	42

NOTE ON SPECIFIC GRAVITIES OF WOODS.

The specific gravity or weight per cubic foot of wood varies with the amount of water it contains. In the preceding list the figures refer to timber at a moisture content of 15 per cent. of the oven-dry weight, which represents a condition intermediate between kiln-seasoned and air-seasoned. A short list showing the change in weight which takes place when green timber is seasoned is given on page 283. Reference may also be made to the number of cubic feet in 1 ton of timber in the green and seasoned conditions on p. 283.

In any particular timber there is a considerable range of weight owing to the natural variation which occurs in different trees and in different parts of the same tree. It must be emphasised that the figures given are averages only.

(*Forest Products Research Laboratory.*)

SPECIFIC GRAVITIES OF GRANULAR MATERIALS.

The values given in the table on p. 122 for granular materials, such as cement, are those of the solid material, and should be distinguished from the density of the material in bulk, which is affected by the voids between the particles.

Bulk Densities of Granular Materials.

Material.	Bulk Density Lbs. per Cub. Ft.	Material.	Bulk Density Lbs. per Cub. Ft.
Cement, Portland	90	Gunpowder	60
,, Roman	97	Gravel	109
Earth, loose	93	Sand, coarse	112
,, rammed	100	,, damp, loose	87
,, with moist sand	128	,, dried, loose	97
		,, siliceous	106

Hydrometer Scales.

Liquids lighter than Water.

The Tagliabue formula, adopted by the American Petroleum Institution and in fairly general use, is

$$\text{Degrees A.P.I.} = \frac{141\cdot5}{\text{Sp. Gr.}} - 131\cdot5$$

The Bureau of Standards formula, less generally used, is

$$\text{Degrees Baumé} = \frac{144}{\text{Sp. Gr.}} - 144$$

(See page 126 for liquids heavier than water.)

Specific Gravities of Gases and Vapours.

(Vapours at the saturation (boiling) point.)

Fluid.	Specific Gravity compared with Air.	Fluid.	Specific Gravity compared with Air.
Atmospheric air	1	Sulphuretted hydrogen	1·19
Ammonia	0·589	Sulphur dioxide	2·21
Carbon dioxide	1·52	Steam, 212°	0·4863
,, monoxide	0·972	Smoke of bituminous coal	0·102
Carburetted hydrogen (Methane)	0·559	,, coke	0·105
Chlorine	2·47	,, wood	0·09
Carbonyl chloride (Phosgene)	3·389	Vapour of alcohol	1·613
Cyanogen	1·815	,, bisulphide of carbon	2·64
Gas, coal	0·4 0·752	,, bromine	5·5
		,, chloric ether	3·44
Hydrogen	0·069	,, ether	2·586
Hydrochloric acid	1·278	,, hydrochloric ether	2·255
Hydrocyanic ,,	0·942	,, iodine	8·675
Nitrogen	0·972	,, nitric acid	2·18
Nitric oxide	1·036	,, spirits of turpentine	4·763
Nitrous acid	2·638	,, sulphuric acid	3·39
,, oxide	1·527	,, ,, ether	2·586
Oxygen	1·102	,, sulphur	2·214
Phosphoretted hydrogen (Phosphine)	1·17	,, water	0·622

Hydrometer Scales.

Liquids heavier than water.

(*Compiled by the Kestner Evaporator and Engineering Co., Ltd.*)

Degrees Twaddell.	Sp. Gr.	Degrees Baumé, 'Rational.'	Approximate Degrees Brix at 15° C.	Degrees Twaddell.	Sp. Gr.	Degrees Baumé, 'Rational.'	Approximate Degrees Brix at 15° C.
1	1·005	0·7	1·3	52	1·260	29·7	55·2
2	1·010	1·4	2·5	54	1·270	30·6	56·6
3	1·015	2·1	3·7	56	1·280	31·5	58·5
4	1·020	2·7	4·9	58	1·290	32·4	60·2
5	1·025	3·4	5·9	60	1·300	33·3	62·0
6	1·030	4·1	7·2	62	1·310	34·2	63·8
7	1·035	4·7	8·3	64	1·320	35·0	65·4
8	1·040	5·4	9·5	66	1·330	35·8	66·8
9	1·045	6·0	10·6	68	1·340	36·6	68·6
10	1·050	6·7	12·0	70	1·350	37·4	70·0
11	1·055	7·4	13·4	72	1·360	38·2	71·8
12	1·060	8·0	14·5	74	1·370	39·0	73·5
13	1·065	8·7	15·8	76	1·380	39·8	75·0
14	1·070	9·4	17·0	78	1·390	40·5	76·4
15	1·075	10·0	18·2	80	1·400	41·2	77·8
16	1·080	10·6	19·3	82	1·410	42·0	79·5
17	1·085	11·2	20·4	84	1·420	42·7	81·0
18	1·090	11·9	21·6	86	1·430	43·4	82·5
19	1·095	12·4	22·5	88	1·440	44·1	83·8
20	1·100	13·0	23·6	90	1·450	44·8	85·4
21	1·105	13·6	24·6	92	1·460	45·4	86·6
22	1·110	14·2	25·6	94	1·470	46·1	88·0
23	1·115	14·9	26·8	96	1·480	46·8	89·5
24	1·120	15·4	27·6	98	1·490	47·4	90·8
25	1·125	16·0	28·6	100	1·500	48·1	92·2
26	1·130	16·5	29·5	102	1·510	48·7	93·8
27	1·135	17·1	30·5	104	1·520	49·4	95·2
28	1·140	17·7	31·5	106	1·530	50·0	96·8
29	1·145	18·3	32·5	108	1·540	50·6	—
30	1·150	18·8	33·4	110	1·550	51·2	—
31	1·155	19·3	34·4	112	1·560	51·8	—
32	1·160	19·8	35·4	114	1·570	52·4	—
33	1·165	20·3	36·2	116	1·580	53·0	—
34	1·170	20·9	37·4	118	1·590	53·6	—
35	1·175	21·4	38·4	120	1·600	54·1	—
36	1·180	22·0	39·4	122	1·610	54·7	—
37	1·185	22·5	40·4	124	1·620	55·2	—
38	1·190	23·0	41·4	126	1·630	55·1	—
39	1·195	23·5	42·2	128	1·640	56·3	—
40	1·200	24·0	43·2	130	1·650	56·9	—
41	1·205	24·5	44·2	132	1·660	57·4	—
42	1·210	25·0	45·2	134	1·670	57·9	—
43	1·215	25·5	46·2	136	1·680	58·4	—
44	1·220	26·0	47·0	138	1·690	58·9	—
45	1·225	26·4	47·8	140	1·700	59·5	—
46	1·230	26·9	48·8	142	1·710	60·0	—
47	1·235	27·4	49·8	144	1·720	60·4	—
48	1·240	27·9	50·6	146	1·730	60·9	—
49	1·245	28·4	52·0	148	1·740	61·4	—
50	1·250	28·8	53·4	150	1·750	61·8	—

B.—WEIGHT OF ENGINEERING MATERIALS AND FITTINGS.

Weight of Sheet, Bar, and Ball Metals.

WEIGHT OF SHEET, BAR, AND BALL CAST IRON.

(1 cubic foot weighs 450 lbs.)

Sheet.			Bar.		Ball.	Sheet.			Bar.		Ball.
Thickness or Diameter.		Weight of a Square Foot.	Weight of a *Square* Bar, 1 Foot long.	Weight of a *Round* Bar, 1 Foot long.	Weight of a Ball.	Thickness or Diameter.		Weight of a Square Foot.	Weight of a *Square* Bar, 1 Foot long.	Weight of a *Round* Bar, 1 Foot long.	Weight of a Ball.
Ins.	Ft.	Lbs.	Lbs.	Lbs.	Lbs.	Ins.	Ft.	Lbs.	Lbs.	Lbs.	Lbs.
1/32	·0026	1·173	·003	·002	—	3 1/8	·2604	117·3	30·52	23·97	4·162
1/16	·0052	2·344	·012	·010	—	1/4	·2708	121·8	33·01	25·93	4·681
3/32	·0078	3·516	·027	·021	·0001	3/8	·2813	126·5	35·60	27·95	5·243
1/8	·0104	4·687	·048	·038	·0003	1/2	·2917	131·2	38·28	30·07	5·846
5/32	·0130	5·861	·076	·060	·0005	5/8	·3021	135·9	41·07	32·25	6·498
3/16	·0156	7·032	·110	·086	·0009	3/4	·3125	140·6	43·95	34·51	7·193
7/32	·0182	8·203	·150	·118	·0014	7/8	·3229	145·3	46·93	36·85	7·934
1/4	·0208	9·375	·195	·154	·0021	4	·3333	150·0	50·01	39·27	8·726
9/32	·0234	10·54	·247	·194	·0030	1/8	·3438	154·7	53·18	41·77	9·572
5/16	·0260	11·73	·305	·240	·0042	1/4	·3542	159·3	56·46	44·33	10·47
11/32	·0287	12·89	·370	·290	·0056	3/8	·3646	164·0	59·82	46·99	11·42
3/8	·0313	14·06	·440	·346	·0072	1/2	·3750	168·7	63·33	49·71	12·43
13/32	·0339	15·24	·516	·400	·0092	5/8	·3854	173·4	66·86	52·52	13·49
7/16	·0365	16·41	·598	·470	·0114	3/4	·3958	178·1	70·52	55·39	14·62
15/32	·0391	17·56	·687	·540	·0140	7/8	·4063	182·8	74·28	58·34	15·81
1/2	·0417	18·75	·781	·610	·0170	5	·4167	187·5	78·12	61·37	17·05
9/16	·0469	21·10	·989	·777	·0243	1/8	·4271	192·2	82·10	64·47	18·35
5/8	·0521	23·44	1·221	·959	·0334	1/4	·4375	196·9	86·14	67·65	19·73
11/16	·0573	25·79	1·478	1·161	·0444	3/8	·4479	201·6	90·29	70·52	21·18
3/4	·0625	28·12	1·758	1·381	·0575	1/2	·4583	206·2	94·54	74·26	22·68
13/16	·0677	30·47	2·064	1·621	·0732	5/8	·4688	210·9	98·89	77·66	24·27
7/8	·0729	32·81	2·393	1·880	·0913	3/4	·4792	215·6	103·3	81·16	25·93
15/16	·0781	35·16	2·747	2·158	·1124	7/8	·4896	220·3	107·9	84·72	27·41
1	·0833	37·50	3·125	2·455	·1363	6	·5000	225·0	112·5	88·36	29·44
1/16	·0885	39·84	3·528	2·771	·1636	1/4	·5208	234·4	122·1	95·89	33·28
1/8	·0938	42·19	3·955	3·107	·1942	1/2	·5417	243·8	132·0	103·7	37·44
3/16	·0990	44·53	4·407	3·461	·2284	3/4	·5625	253·1	142·4	111·9	41·94
1/4	·1042	46·87	4·883	3·835	·2664	7	·5833	262·5	153·2	120·2	46·77
5/16	·1094	49·22	5·384	4·229	·3084	1/4	·6042	271·9	164·2	129·0	51·97
3/8	·1146	51·57	5·909	4·640	·3546	1/2	·6250	281·3	175·8	138·1	57·54
7/16	·1198	53·91	6·461	5·073	·4058	3/4	·6458	290·7	187·7	147·4	63·47
1/2	·1250	56·26	7·033	5·523	·4603	8	·6667	300·0	200·1	157·0	69·82
9/16	·1302	58·60	7·632	5·993	·5204	1/4	·6875	309·4	212·7	167·0	76·58
5/8	·1354	60·94	8·253	6·484	·5852	1/2	·7083	318·8	225·8	177·3	83·74
11/16	·1406	63·28	8·900	6·991	·6555	3/4	·7292	328·2	239·3	187·9	91·35
3/4	·1458	65·63	9·572	7·518	·7310	9	·7500	337·4	253·1	198·8	99·42
13/16	·1510	67·97	10·27	8·064	·8122	1/4	·7708	346·8	267·4	210·0	107·9
7/8	·1563	70·32	10·99	8·630	·8991	1/2	·7917	356·2	282·1	221·5	116·8
15/16	·1615	72·66	11·73	9·215	·9920	3/4	·8125	365·6	297·0	233·3	126·3
2	·1667	75·01	12·50	9·821	1·073	10	·8333	375·0	312·5	245·5	136·3
1/8	·1771	79·70	14·11	11·09	1·308	1/4	·8542	384·4	328·4	257·8	146·8
1/4	·1875	84·40	15·83	12·43	1·554	1/2	·8750	393·7	344·5	270·6	157·9
3/8	·1979	89·07	17·63	13·85	1·827	3/4	·8958	403·1	361·2	283·7	169·3
1/2	·2083	93·75	19·54	15·34	2·131	11	·9167	412·5	378·2	297·0	181·6
5/8	·2188	98·44	21·54	16·56	2·467	1/4	·9375	421·9	395·5	310·6	194·2
3/4	·2292	103·2	23·64	18·56	2·835	1/2	·9583	431·2	413·3	324·6	207·3
7/8	·2396	107·8	25·84	20·29	3·241	3/4	·9792	440·6	431·4	338·8	219·2
3	·2500	112·6	28·13	22·10	3·682	12	1 Foot	450·0	450·0	353·4	235·6

Weight of Wrought Iron and Steel.

(The Table gives the weights for wrought iron, 1 cubic foot of wrought iron being taken to weigh 485 lbs. For steel an addition of about 1 lb. in 100 lbs. must be made to the tabular amounts.)

Thickness or Diameter in Inches.	Thickness or Diameter in Decimals of a Foot.	Weight of a *Square Foot* in Pounds.	Weight of a *Square* Bar, 1 Foot long, in Pounds.	Weight of a *Round* Bar, 1 Foot long, in Pounds.	Weight of *Balls* in Pounds.	Thickness or Diameter in Inches.	Thickness or Diameter in Decimals of a Foot.	Weight of a *Square Foot* in Pounds.	Weight of a *Square* Bar, 1 Foot long, in Pounds.	Weight of a *Round* Bar, 1 Foot long, in Pounds.	Weight of *Balls* in Pounds.
1/32	·0026	1·263	·0033	·0026	—	3 1/8	·2604	126·3	32·89	25·83	4·484
1/16	·0052	2·526	·0132	·0104	—	1/4	·2708	131·4	35·57	27·94	5·045
3/32	·0078	3·789	·0296	·0233	·0001	3/8	·2813	136·4	38·37	30·13	5·649
1/8	·0104	5·052	·0526	·0414	·0003	1/2	·2917	141·5	41·26	32·41	6·301
5/32	·0130	6·315	·0823	·0646	·0005	5/8	·3021	146·5	44·26	34·76	7·000
3/16	·0156	7·578	·1184	·0930	·0009	3/4	·3125	151·6	47·37	37·20	7·750
7/32	·0182	8·841	·1612	·1266	·0015	7/8	·3229	156·6	50·57	39·72	8·550
1/4	·0208	10·10	·2105	·1653	·0023	4	·3333	161·7	53·89	42·33	9·405
9/32	·0234	11·37	·2665	·2093	·0033	1/8	·3438	166·7	57·31	45·01	10·32
5/16	·0260	12·63	·3290	·2583	·0045	1/4	·3542	171·8	60·84	47·78	11·28
11/32	·0287	13·89	·3980	·3126	·0060	3/8	·3646	176·8	64·47	50·63	12·31
3/8	·0313	15·16	·4736	·3720	·0078	1/2	·3750	181·9	68·20	53·57	13·39
13/32	·0339	16·42	·5558	·4365	·0098	5/8	·3854	186·9	72·05	56·59	14·54
7/16	·0365	17·68	·6446	·5063	·0123	3/4	·3958	192·0	75·99	59·69	15·75
15/32	·0391	18·95	·7400	·5813	·0151	7/8	·4063	197·0	80·05	62·87	17·03
1/2	·0417	20·21	·8420	·6613	·0184	5	·4167	202·1	84·20	66·13	18·37
9/16	·0469	22·73	1·066	·8370	·0262	1/8	·4271	207·1	88·47	69·48	19·78
5/8	·0521	25·26	1·316	1·033	·0359	1/4	·4375	212·2	92·83	72·91	21·26
11/16	·0573	27·79	1·592	1·250	·0478	3/8	·4479	217·2	97·31	76·43	22·82
3/4	·0625	30·31	1·895	1·488	·0620	1/2	·4583	222·3	101·9	80·02	24·45
13/16	·0677	32·84	2·223	1·746	·0788	5/8	·4688	227·3	106·6	83·70	26·16
7/8	·0729	35·37	2·579	2·025	·0985	3/4	·4792	232·4	111·4	87·46	27·94
15/16	·0781	37·89	2·960	2·325	·1211	7/8	·4896	237·5	116·3	91·31	29·80
1	·0833	40·42	3·368	2·645	·1470	6	·5000	242·5	121·3	95·23	31·74
1/16	·0885	42·94	3·803	2·986	·1763	1/4	·5208	252·6	131·6	103·3	35·88
1/8	·0938	45·47	4·263	3·348	·2093	1/2	·5417	262·7	142·3	111·8	40·36
3/16	·0990	48·00	4·750	3·730	·2461	3/4	·5625	272·8	153·5	120·5	45·19
1/4	·1042	50·52	5·263	4·133	·2870	7	·5833	282·9	165·0	129·6	50·40
5/16	·1094	53·05	5·802	4·557	·3323	1/4	·6042	293·0	177·0	139·0	56·00
3/8	·1146	55·57	6·368	5·001	·3820	1/2	·6250	303·1	189·5	148·8	62·00
7/16	·1198	58·10	6·960	5·466	·4365	3/4	·6458	313·2	202·3	158·9	68·40
1/2	·1250	60·63	7·578	5·952	·4960	8	·6667	323·3	215·6	169·3	75·24
9/16	·1302	63·15	8·223	6·458	·5606	1/4	·6875	333·4	229·3	180·1	82·52
5/8	·1354	65·68	8·893	6·985	·6306	1/2	·7083	343·5	243·4	191·1	90·25
11/16	·1406	68·20	9·591	7·533	·7062	3/4	·7292	353·6	247·9	202·5	98·45
3/4	·1458	70·73	10·31	8·101	·7876	9	·7500	363·8	272·8	214·3	107·1
13/16	·1510	73·26	11·07	8·690	·8750	1/4	·7708	373·9	288·2	226·3	116·3
7/8	·1563	75·78	11·84	9·300	·9688	1/2	·7917	384·0	304·0	238·7	126·0
15/16	·1615	78·31	12·64	9·930	1·069	3/4	·8125	394·1	320·2	251·5	136·2
2	·1667	80·83	13·47	10·58	1·176	10	·8333	404·2	336·8	264·5	146·9
1/8	·1771	85·89	15·21	11·95	1·410	1/4	·8542	414·3	353·9	277·9	158·2
1/4	·1875	90·94	17·05	13·39	1·674	1/2	·8750	424·4	371·3	291·6	170·1
3/8	·1979	95·99	19·00	14·92	1·969	3/4	·8958	434·5	389·2	305·7	182·6
1/2	·2083	101·0	21·05	16·53	2·296	11	·9167	444·6	407·5	320·1	195·6
5/8	·2188	106·1	23·21	18·23	2·658	1/4	·9375	454·7	426·3	334·8	209·2
3/4	·2292	111·2	25·47	20·01	3·056	1/2	·9583	464·8	445·4	349·8	223·5
7/8	·2396	116·2	27·84	21·87	3·492	3/4	·9792	474·9	465·0	365·2	238·4
3	·2500	121·3	30·31	23·81	3·968	12	1 Foot	485·0	485·0	380·9	253·9

WEIGHT OF A SQUARE FOOT OF SHEET METALS.

Thickness.	Weight in Lbs.			Thickness.	Weight in Lbs.		
Inch.	Iron.	Copper.	Brass.	Inch.	Iron.	Copper.	Brass.
½	20·21	23·12	21·87	$\frac{1}{16}$	2·52	2·89	2·73
·450	18·19	20·80	19·68	·055	2·22	2·54	2·40
·437	17·66	20·20	19·11	·048	1·94	2·22	2·10
⅜	15·15	17·34	16·40	·042	1·69	1·94	1·83
·340	13·74	15·72	14·87	·035	1·41	1·62	1·53
$\frac{5}{16}$	12·63	14·45	13·67	·033	1·33	1·52	1·44
·284	11·47	13·12	12·42	·029	1·17	1·34	1·27
·261	10·55	12·06	11·41	·028	1·13	1·29	1·22
¼	10·10	11·56	10·94	·025	1·01	1·15	1·09
·217	8·77	10·03	9·49	·021	·85	·97	·92
·208	8·40	9·62	9·10	·020	·81	·92	·87
$\frac{3}{16}$	7·57	8·67	8·20	·018	·73	·83	·79
·166	6·71	7·67	7·26	·015	·60	·69	·65
·158	6·38	7·30	6·91	·013	·52	·60	·57
·137	5·54	6·33	5·99	·012	·48	·55	·52
⅛	5·05	5·78	5·47	·010	·40	·46	·43
·109	4·40	5·04	4·77	·009	·36	·41	·40
·094	3·80	4·34	4·11	·008	·32	·37	·35
·080	3·23	3·70	3·50	·007	·28	·32	·30
·072	2·91	3·32	3·15	·005	·20	·23	·22

WEIGHT OF ROUND, OCTAGONAL, AND SQUARE STEEL PER FOOT RUN.

Diam.	Round.	Octag.	Square.	Diam.	Round.	Octag.	Square.	Diam.	Round.	Octag.	Square.
In.	Lb.	Lb.	Lb.	Ins.	Lbs.	Lbs.	Lbs.	Ins.	Lbs.	Lbs.	Lbs.
$\frac{1}{16}$	·010	·011	·013	⅝	1·044	1·101	1·329	1⅜	5·052	5·330	6·433
⅛	·041	·044	·053	$\frac{11}{16}$	1·263	1·332	1·608	1½	6·013	6·343	7·656
$\frac{3}{16}$	·094	·099	·119	¾	1·503	1·586	1·914	1⅝	7·057	7·444	8·985
¼	·167	·176	·213	$\frac{13}{16}$	1·764	1·861	2·246	1¾	8·185	8·633	10·421
$\frac{5}{16}$	·261	·275	·332	⅞	2·046	2·158	2·605	1⅞	9·396	9·910	11·963
⅜	·376	·396	·478	$\frac{15}{16}$	2·349	2·478	2·991	2	10·690	11·276	13·611
$\frac{7}{16}$	·511	·539	·651	1	2·673	2·819	3·403	2¼	13·530	14·271	17·227
½	·668	·706	·851	1⅛	3·382	3·568	4·307	2½	16·703	17·618	21·267
$\frac{9}{16}$	·845	·892	1·077	1¼	4·176	4·405	5·317	2¾	20·211	21·318	25·734

(Diameter of octagon steel is measured across sides.)

WEIGHT OF FLAT ROLLED IRON PER FOOT RUN.

Width.	Thickness of Metal in Parts of an Inch.											
	$\frac{1}{16}$	⅛	$\frac{3}{16}$	¼	$\frac{5}{16}$	⅜	$\frac{7}{16}$	½	⅝	¾	⅞	1
Ins.	Lb.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
1	·2083	·4166	·6250	·8333	1·042	1·250	1·458	1·666	2·083	2·500	2·916	3·333
⅛	·2344	·4688	·7033	·9375	1·172	1·406	1·640	1·875	2·344	2·812	3·280	3·75
¼	·2605	·5210	·7810	1·042	1·303	1·563	1·823	2·083	2·605	3·125	3·646	4·166
⅜	·2865	·5730	·8595	1·146	1·432	1·719	2·006	2·292	2·864	3·438	4·012	4·583
½	·3125	·6250	·9375	1·250	1·562	1·875	2·188	2·500	3·125	3·750	4·375	5·000
⅝	·3385	·6771	1·015	1·354	1·692	2·031	2·370	2·708	3·384	4·062	4·740	5·416
¾	·3646	·7292	1·094	1·458	1·823	2·188	2·550	2·916	3·646	4·375	5·105	5·833
⅞	·3906	·7812	1·172	1·562	1·953	2·344	2·735	3·125	3·906	4·688	5·470	6·25
2	·4166	·8333	1·25	1·666	2·083	2·500	2·916	3·333	4·166	5·000	5·833	6·666
⅛	·4427	·8855	1·328	1·771	2·214	2·656	3·098	3·542	4·428	5·312	6·198	7·083
¼	·4686	·9375	1·406	1·875	2·344	2·812	3·281	3·750	4·688	5·624	6·562	7·500
⅜	·4948	·9895	1·484	1·979	2·474	2·968	3·463	3·958	4·948	5·936	6·926	7·916
½	·5210	1·042	1·562	2·083	2·605	3·125	3·646	4·166	5·210	6·250	7·291	8·333
⅝	·5470	1·094	1·641	2·187	2·735	3·282	3·829	4·375	5·470	6·564	7·658	8·750
¾	·5730	1·146	1·719	2·292	2·865	3·438	4·011	4·583	5·730	6·876	8·022	9·166
⅞	·5990	1·198	1·797	2·396	2·995	3·594	4·193	4·792	5·990	7·188	8·386	9·583
3	·625	1·250	1·875	2·500	3·125	3·750	4·375	5·000	6·250	7·500	8·750	10·00
⅛	·6515	1·303	1·954	2·605	3·257	3·908	4·560	5·210	6·514	7·816	9·120	10·42
¼	·6770	1·354	2·031	2·708	3·385	4·062	4·739	5·416	6·770	8·124	9·478	10·83
⅜	·7031	1·406	2·109	2·812	3·516	4·218	4·921	5·625	7·032	8·436	9·842	11·25
½	·7291	1·458	2·188	2·916	3·646	4·375	5·105	5·833	7·291	8·750	10·21	11·66
⅝	·7555	1·511	2·266	3·021	3·777	4·533	5·288	6·042	7·554	9·066	10·58	12·08
¾	·7812	1·562	2·343	3·125	3·906	4·686	5·468	6·25	7·812	9·372	10·94	12·50
⅞	·8070	1·614	2·421	3·229	4·035	4·842	5·65	6·458	8·070	9·684	11·30	12·92

WEIGHT OF FLAT ROLLED IRON PER FOOT RUN (*continued*).

Width.	Thickness of Metal in Parts of an Inch.											
	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
4	·8333	1·666	2·500	3·333	4·166	5·000	5·833	6·666	8·333	10·00	11·66	13·33
$\frac{1}{8}$	·8595	1·719	2·578	3·438	4·297	5·166	6·016	6·875	8·594	10·31	12·03	13·75
$\frac{1}{4}$	·8855	1·771	2·656	3·542	4·427	5·312	6·198	7·083	8·854	10·62	12·40	14·16
$\frac{3}{8}$	·9115	1·823	2·734	3·646	4·557	5·468	6·380	7·291	9·114	10·94	12·76	14·58
$\frac{1}{2}$	·9375	1·875	2·812	3·750	4·687	5·624	6·562	7·500	9·374	11·25	13·12	15·00
$\frac{5}{8}$	·9636	1·927	2·891	3·854	4·818	5·782	6·745	7·708	9·636	11·56	13·49	15·42
$\frac{3}{4}$	·9895	1·979	2·968	3·958	4·947	5·936	6·926	7·917	9·894	11·87	13·85	15·83
$\frac{7}{8}$	1·016	2·031	3·048	4·062	5·080	6·096	7·112	8·125	10·16	12·19	14·22	16·25
5	1·042	2·083	3·125	4·166	5·210	6·25	7·291	8·333	10·42	12·50	14·58	16·66
$\frac{1}{8}$	1·068	2·136	3·204	4·271	5·340	6·408	7·476	8·542	10·68	12·81	14·95	17·08
$\frac{1}{4}$	1·094	2·188	3·282	4·375	5·470	6·564	7·658	8·750	10·94	13·13	15·31	17·50
$\frac{3}{8}$	1·120	2·240	3·360	4·479	5·600	6·720	7·840	8·958	11·20	13·44	15·68	17·92
$\frac{1}{2}$	1·146	2·292	3·438	4·584	5·730	6·876	8·022	9·167	11·46	13·75	16·04	18·33
$\frac{5}{8}$	1·172	2·344	3·516	4·687	5·860	7·032	8·204	9·375	11·72	14·06	16·40	18·75
$\frac{3}{4}$	1·198	2·396	3·594	4·791	5·990	7·188	8·386	9·583	11·98	14·37	16·77	19·16
$\frac{7}{8}$	1·224	2·448	3·672	4·896	6·120	7·344	8·568	9·792	12·24	14·69	17·13	19·58
6	1·250	2·500	3·750	5·000	6·250	7·500	8·750	10·00	12·50	15·00	17·50	20·00
$\frac{1}{8}$	1·276	2·552	3·828	5·104	6·380	7·656	8·932	10·21	12·76	15·31	17·86	20·42
$\frac{1}{4}$	1·302	2·604	3·906	5·208	6·510	7·812	9·114	10·42	13·02	15·62	18·23	20·83
$\frac{3}{8}$	1·328	2·657	3·984	5·312	6·640	7·968	9·297	10·63	13·28	15·93	18·59	21·25
$\frac{1}{2}$	1·354	2·708	4·063	5·417	6·770	8·126	9·480	10·83	13·54	16·25	18·96	21·66
$\frac{5}{8}$	1·381	2·761	4·143	5·521	6·906	8·286	9·668	11·04	13·81	16·57	19·33	22·08
$\frac{3}{4}$	1·406	2·813	4·218	5·625	7·030	8·436	9·843	11·25	14·06	16·87	19·69	22·50
$\frac{7}{8}$	1·432	2·864	4·296	5·729	7·160	8·592	10·02	11·46	14·32	17·18	20·04	22·92
7	1·458	2·916	4·375	5·833	7·291	8·750	10·20	11·66	14·58	17·50	20·42	23·33
$\frac{1}{8}$	1·484	2·969	4·452	5·938	7·420	8·904	10·39	11·87	14·84	17·81	20·78	23·75
$\frac{1}{4}$	1·511	3·021	4·533	6·042	7·555	9·066	10·58	12·08	15·11	18·13	21·16	24·16
$\frac{3}{8}$	1·536	3·073	4·608	6·146	7·680	9·216	10·75	12·29	15·36	18·43	21·50	24·58
$\frac{1}{2}$	1·562	3·125	4·686	6·250	7·810	9·372	10·93	12·50	15·62	18·74	21·86	25·00
$\frac{5}{8}$	1·588	3·177	4·764	6·354	7·940	9·528	11·12	12·71	15·88	19·05	22·24	25·42
$\frac{3}{4}$	1·615	3·229	4·845	6·458	8·075	9·690	11·31	12·92	16·15	19·38	22·62	25·88
$\frac{7}{8}$	1·641	3·281	4·923	6·562	8·205	9·846	11·48	13·13	16·41	19·69	22·96	26·25
8	1·666	3·333	5·000	6·666	8·333	10·00	11·66	13·33	16·66	20·00	23·33	26·66
$\frac{1}{8}$	1·693	3·386	5·079	6·771	8·455	10·15	11·85	13·54	16·91	20·30	23·70	27·08
$\frac{1}{4}$	1·719	3·438	5·157	6·875	8·595	10·31	12·03	13·75	17·19	20·61	24·06	27·50
$\frac{3}{8}$	1·745	3·489	5·235	6·979	8·725	10·47	12·21	13·96	17·45	20·94	24·42	27·92
$\frac{1}{2}$	1·771	3·542	5·313	7·083	8·855	10·63	12·40	14·17	17·71	21·26	24·80	28·33
$\frac{5}{8}$	1·797	3·594	5·391	7·188	8·985	10·78	12·58	14·37	17·97	21·56	25·16	28·75
$\frac{3}{4}$	1·823	3·646	5·469	7·292	9·115	10·94	12·76	14·58	18·23	21·88	25·52	29·17
$\frac{7}{8}$	1·849	3·698	5·547	7·396	9·245	11·09	12·94	14·79	18·49	22·18	25·88	29·58
9	1·875	3·750	5·625	7·500	9·375	11·25	13·12	15·00	18·75	22·50	26·24	30·00
$\frac{1}{8}$	1·901	3·802	5·703	7·604	9·505	11·41	13·31	15·21	19·00	22·81	26·62	30·42
$\frac{1}{4}$	1·927	3·854	5·781	7·708	9·635	11·56	13·49	15·42	19·27	23·12	26·98	30·83
$\frac{3}{8}$	1·953	3·906	5·859	7·812	9·765	11·72	13·67	15·62	19·53	23·44	27·34	31·25
$\frac{1}{2}$	1·979	3·958	5·937	7·916	9·895	11·87	13·85	15·84	19·79	23·74	27·70	31·67
$\frac{5}{8}$	2·005	4·010	6·015	8·021	10·02	12·03	14·04	16·04	20·04	24·06	28·08	32·08
$\frac{3}{4}$	2·031	4·062	6·093	8·125	10·16	12·18	14·21	16·25	20·32	24·36	28·42	32·50
$\frac{7}{8}$	2·057	4·114	6·171	8·229	10·29	12·31	14·40	16·46	20·58	24·68	28·80	32·92
10	2·083	4·166	6·250	8·333	10·41	12·50	14·58	16·66	20·82	25·00	29·16	33·33
$\frac{1}{8}$	2·109	4·219	6·327	8·438	10·55	12·65	14·76	16·87	21·10	25·30	29·52	33·75
$\frac{1}{4}$	2·135	4·270	6·405	8·541	10·67	12·81	14·94	17·08	21·34	25·62	29·88	34·17
$\frac{3}{8}$	2·162	4·323	6·486	8·646	10·81	12·97	15·13	17·29	21·62	25·94	30·26	34·58
$\frac{1}{2}$	2·188	4·375	6·564	8·750	10·94	13·13	15·31	17·50	21·88	26·26	30·62	35·00
$\frac{5}{8}$	2·214	4·427	6·642	8·854	11·07	13·28	15·50	17·71	22·14	26·56	31·00	35·42
$\frac{3}{4}$	2·239	4·479	6·717	8·958	11·20	13·13	15·67	17·92	22·40	26·86	31·34	35·83
$\frac{7}{8}$	2·266	4·531	6·798	9·062	11·33	13·59	15·86	18·12	22·66	27·18	31·72	36·25
11	2·291	4·583	6·873	9·166	11·46	13·75	16·04	18·33	22·90	27·50	32·08	36·66
$\frac{1}{8}$	2·318	4·636	6·954	9·271	11·59	13·91	16·22	18·54	23·18	27·82	32·44	37·08
$\frac{1}{4}$	2·344	4·688	7·032	9·375	11·72	14·06	16·40	18·75	23·44	28·12	32·80	37·50
$\frac{3}{8}$	2·370	4·740	7·110	9·479	11·85	14·22	16·59	18·96	23·70	28·44	33·18	37·92
$\frac{1}{2}$	2·395	4·791	7·185	9·582	11·97	14·37	16·76	19·16	23·94	28·74	33·52	38·33
$\frac{5}{8}$	2·422	4·844	7·266	9·688	12·11	14·53	16·95	19·37	24·22	29·00	33·90	38·75
$\frac{3}{4}$	2·448	4·896	7·344	9·792	12·24	14·68	17·13	19·58	24·48	29·36	34·26	39·16
$\frac{7}{8}$	2·474	4·948	7·422	9·896	12·37	14·84	17·32	19·79	24·74	29·68	34·64	39·58
12	2·500	5·000	7·500	10·00	12·50	15·00	17·50	20·00	25·00	30·00	35·00	40·00

Weight of Flat Steel per Foot Run.

Width.	Thickness of Metal in Parts of an Inch. ⅛	¼	⅜	½	⅝	¾	⅞	1
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
⅛	·0532	·1063	·1595	·2127	·2658	·3190	·3722	·4253
½	·2127	·4253	·6380	·8507	1·063	1·276	1·489	1·701
⅝	·2658	·5317	·7975	1·063	1·329	1·595	1·861	2·127
¾	·3190	·6380	·9570	1·276	1·595	1·914	2·233	2·552
⅞	·3722	·7443	1·117	1·489	1·861	2·233	2·605	2·977
1	·4253	·8507	1·276	1·701	2·127	2·552	2·976	3·403
1⅛	·4785	·9570	1·436	1·914	2·393	2·871	3·350	3·828
1¼	·5317	1·063	1·595	2·127	2·658	3·190	3·722	4·253
1⅜	·5849	1·170	1·755	2·339	2·924	3·509	4·094	4·679
1½	·6380	1·276	1·914	2·552	3·190	3·828	4·466	5·104
1¾	·7444	1·489	2·233	2·977	3·722	4·466	5·211	5·955
2	·8507	1·701	2·552	**3·403**	4·253	5·104	5·955	6·806
2¼	·9570	1·914	2·871	3·828	4·785	5·742	6·699	7·656
2½	1·063	2·127	3·190	4·253	5·317	6·380	7·444	8·507
2¾	1·170	2·339	3·509	4·679	5·849	7·018	8·188	9·358
3	1·276	2·552	3·828	5·104	6·380	7·656	8·932	10·208
3¼	1·382	2·765	4·147	5·530	6·912	8·294	9·677	11·059
3½	1·489	2·977	4·466	5·955	7·444	8·932	10·421	11·910
3¾	1·595	3·190	4·785	6·380	7·975	9·570	11·165	12·760
4	1·701	3·403	5·104	6·806	8·507	10·208	11·910	13·611
4¼	1·808	3·615	5·423	7·231	9·039	10·846	12·654	14·462
4½	1·914	3·828	5·742	7·656	9·570	11·484	13·398	15·313
4¾	2·020	4·041	6·061	[illegible]	10·102	12·122	14·143	16·163
5	2·127	4·254	6·380	8·507	10·634	12·760	14·887	17·014
5½	2·339	4·679	7·018	9·358	11·697	14·037	16·376	18·715
6	2·552	5·104	7·656	10·208	12·761	15·313	17·865	20·417

Weight of Steel Wire Ropes.

The weight per fathom (6 ft.) in lbs. of the usual engineering wire rope is approximately the square of the circumference, *i.e.*, for example, a 3-in. rope would weigh 9 lbs. per fathom, approximately.

Weight of Hexagon Rod Metal.

Weight of hexagon rod metal, measured across sides = weight of square metal × 0·86602.

Weight of Octagon Rod Metal.

Weight of octagon rod metal, measured across sides = weight of square metal × 0·82843.

Weight of Rolled Lead, Copper, and Brass: Sheets and Bars.

Thickness or Diameter, or Side.	Lead. Sheets, per Square Foot.	Lead. Square Bars, 1 Foot long.	Lead. Round Bars, 1 Foot long.	Copper. Sheets, per Square Foot.	Copper. Square Bars, 1 Foot long.	Copper. Round Bars, 1 Foot long.	Brass. Sheets, per Square Foot.	Brass. Square Bars, 1 Foot long.	Brass. Round Bars, 1 Foot long.	Thickness or Diameter, or Side.
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Ins.
1/32	1·86	·005	·004	1·44	·004	·003	1·36	·004	·003	1/32
1/16	3·72	·019	·015	2·89	·015	·012	2·71	·014	·011	1/16
3/32	5·58	·044	·034	4·33	·034	·027	4·06	·032	·025	3/32
⅛	7·44	·078	·061	5·77	·060	·047	5·12	·056	·044	⅛
5/32	9·30	·121	·095	7·20	·094	·074	6·75	·088	·069	5/32
3/16	11·2	·174	·137	8·66	·135	·106	8·13	·127	·109	3/16
7/32	13·0	·237	·187	10·1	·184	·144	9·50	·173	·136	7/32
¼	14·9	·310	·244	11·5	·240	·189	10·8	·226	·177	¼
5/16	18·6	·485	·381	14·4	·376	·295	13·5	·353	·277	5/16
⅜	22·3	·698	·548	17·3	·541	·425	16·3	·508	·399	⅜

Weight of Rolled Lead, Copper, and Brass: Sheets and Bars—*continued.*

Thickness or Diameter, or Side.	Lead.			Copper.			Brass.			Thickness or Diameter, or Side.
	Sheets, per Square Foot.	Square Bars, 1 Foot long.	Round Bars, 1 Foot long.	Sheets, per Square Foot.	Square Bars, 1 Foot long.	Round Bars, 1 Foot long.	Sheets, per Square Foot.	Square Bars, 1 Foot long.	Round Bars, 1 Foot long.	
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Ins.
7/16	26·0	·950	·746	20·2	·736	·578	19·0	·691	·543	7/16
1/2	29·8	1·24	·974	23·1	·962	·755	21·7	·903	·709	1/2
9/16	33·5	1·57	1·23	26·0	1·22	·955	24·3	1·14	·900	9/16
5/8	37·2	1·94	1·52	28·9	1·50	1·18	27·1	1·41	1·11	5/8
11/16	40·9	2·34	1·84	31·7	1·82	1·43	29·8	1·70	1·34	11/16
3/4	44·6	2·79	2·19	34·6	2·16	1·70	32·5	2·03	1·60	3/4
13/16	48·3	3·27	2·57	37·5	2·55	1·99	35·2	2·38	1·87	13/16
7/8	52·1	3·80	2·98	40·4	2·94	2·31	37·9	2·76	2·17	7/8
15/16	56·0	4·37	3·42	43·3	3·38	2·65	40·6	3·18	2·49	15/16
1	59·5	4·96	3·90	46·2	3·85	3·02	43·3	3·61	2·84	1
1 1/8	66·9	6·27	4·92	52·0	4·87	3·82	48·7	4·57	3·60	1 1/8
1 1/4	74·4	7·75	6·09	57·7	6·01	4·72	54·2	5·64	4·43	1 1/4
1 3/8	81·8	9·37	7·37	63·5	7·28	5·72	59·6	6·82	5·37	1 3/8
1 1/2	89·3	11·2	8·77	69·3	8·65	6·80	65·0	8·12	6·38	1 1/2
1 3/4	104	15·2	11·9	80·8	11·8	9·25	75·9	11·1	8·68	1 3/4
2	119	19·8	15·6	92·3	15·4	12·1	86·7	14·4	11·3	2

Weight of Sheet Lead.

Lbs. per Sq. Ft.	Thickness. Inch.	Lbs. per Sq. Ft.	Thickness. Inch.	Lbs. per Sq. Ft.	Thickness. Inch.	Lbs. per Sq. Ft.	Thickness. Inch.	Lbs. per Sq. Ft.	Thickness. Inch.	Lbs. per Sq. Ft.	Thickness. Inch.
2·5	·042	4	·068	6	·102	8	·136	10	·170	14	·237
3	·051	5	·085	7	·119	9	·153	12	·203	16	·271

Weight of Sheet Zinc.

Gauge No.	Approximate Thickness.		Approximate weight per square foot.			Approximate weight of Sheets. 36 ins. × 72 ins.			36 ins. × 84 ins.			36 ins. × 96 ins.		
	In.	Mm.	lbs.	oz.	drms.	lbs.	oz.	drms.	lbs.	oz.	drms.	lbs.	oz.	drms.
3	0·007	0·171	...	3	15	4	6	14	5	2	11	5	14	8
4	0·008	0·209	...	4	13	5	6	10	6	5	1	7	3	8
5	0·010	0·247	...	5	11	6	6	6	7	7	7	8	8	8
6	0·011	0·291	...	6	11	7	8	6	8	12	7	10	0	8
7	0·013	0·337	...	7	12	8	11	8	10	2	12	12	10	0
8	0·015	0·386	...	8	14	9	15	12	11	10	6	13	5	0
9	0·018	0·450	...	10	5	11	9	10	13	8	9	15	7	8
10	0·020	0·500	...	11	7	12	13	14	15	0	3	17	2	8
12	0·026	0·660	...	15	2	17	0	4	19	13	10	22	11	0
14	0·032	0·820	1	2	12	21	1	8	24	9	12	28	2	0
16	0·043	1·080	1	8	12	27	13	8	32	7	12	37	2	0
18	0·053	1·340	1	14	11	34	8	6	40	4	7	46	0	8
20	0·063	1·600	2	4	10	41	3	4	48	1	2	54	15	0
22	0·077	1·960	2	12	14	50	7	12	58	14	6	67	5	0
24	0·091	2·320	3	5	3	59	13	6	69	12	15	79	12	8
26	0·105	2·680	3	13	7	69	1	14	80	10	3	92	2	8

Specification of Sheet Metals.

When specifying for sheet metals, care should be taken that the nature of the gauge is stated, *i.e.* whether it is S.W.G., B.G., or Trade Gauge. In the case of 'Sheet Zinc,' for example, the Trade Gauge No. 20 is 0·063 in. (see above, and Index, 'Sheet Zinc Trade Gauge'), but this if expressed in S.W.G. is 0·36 in. (see Index, 'Imperial Standard Wire Gauge').

Weight and Electrical Characteristics of Aluminium Bar and Strip.

DIMENSIONS		SECTIONAL AREA	WEIGHT	RESISTANCE AT 20°C. (1)	CURRENT CARRYING CAPACITY (2) AMPERES FOR 40°C. RISE	
Width ins.	*Thickness ins.*	*Sq. ins.*	*lbs./ft.*	*Microhms per ft.*	*Bright*	*Dull Black*
6	¾	4·500	5·27	2·98	2780	3400
6	⅝	3·750	4·40	3·58	2540	3110
6	½	3·000	3·516	4·48	2280	2790
6	⅜	2·250	2·637	5·97	1980	2425
6	¼	1·500	1·758	8·96	1635	2000
5	½	2·500	2·930	5·38	1930	2365
5	⅜	1·875	2·198	7·17	1675	2050
5	¼	1·250	1·465	10·8	1380	1690
5	3/16	0·938	1·099	14·3	1205	1475
4½	½	2·250	2·637	5·97	1755	2150
4½	⅜	1·688	1·978	7·96	1520	1860
4½	¼	1·125	1·319	11·9	1250	1530
4½	3/16	0·844	0·989	15·9	1090	1335
4	¾	3·000	3·516	4·48	1945	2380
4	½	2·000	2·344	6·72	1575	1930
4	⅜	1·500	1·578	8·96	1370	1680
4	¼	1·000	1·172	13·4	1125	1380
4	3/16	0·750	0·879	17·9	980	1200
4	⅛	0·500	0·586	26·9	810	992
3½	½	1·750	2·051	7·68	1400	1715
3½	¼	0·875	1·026	15·4	995	1220
3¼	½	1·625	1·905	8·27	1310	1605
3¼	¼	0·813	0·953	16·5	925	1135
3	⅜	1·125	1·319	11·9	1055	1290
3	¼	0·750	0·879	17·9	862	1055
3	3/16	0·563	0·660	23·9	750	919
2¾	¼	0·688	0·806	19·5	797	977
2 9/16	⅜	0·961	1·126	14·0	915	1120
2½	½	1·250	1·465	10·8	1040	1275
2½	⅜	0·938	1·099	14·3	896	1100

(*The British Aluminium Co. Ltd.*)

(1) Based on an average resistivity of 2·845 microhms per cm. cube.

(2) Where a rise of 50° C. is permissible these capacities should be multiplied by 1·135.

Weight and Electrical Characteristics of Aluminium Bar and Strip.

DIMENSIONS		SECTIONAL AREA	WEIGHT	RESISTANCE AT 20°C. (1)	CURRENT CARRYING CAPACITY AMPERES FOR 40°C. RISE	
Width ins.	*Thickness ins.*	*Sq. ins.*	*lbs./ft.*	*Microhms per ft.*	*Bright*	*Dull Black*
2½	5/16	0·781	0·915	17·2	818	1000
2½	¼	0·625	0·733	21·5	730	895
2¼	¾	1·688	1·978	7·96	1195	1465
2¼	½	1·125	1·319	11·9	952	1165
2¼	⅜	0·844	0·989	15·9	818	1000
2¼	5/16	0·703	0·824	19·1	745	913
2¼	¼	0·563	0·660	23·9	665	815
2¼	3/16	0·422	0·495	31·8	578	708
2	¾	1·500	1·758	8·96	1085	1330
2	⅝	1·250	1·465	10·8	975	1195
2	½	1·000	1·172	13·4	862	1055
2	⅜	0·750	0·879	17·9	738	905
2	5/16	0·625	0·733	21·5	671	822
2	¼	0·500	0·586	26·9	598	733
2	3/16	0·375	0·440	35·8	519	635
2	⅛	0·250	0·293	53·8	425	520
1 15/16	⅛	0·242	0·284	55·5	413	506
1 13/16	5/16	0·566	0·663	23·7	615	753
1¾	⅝	1·094	1·282	12·3	875	1070
1¾	½	0·875	1·026	15·4	770	943
1¾	¼	0·438	0·513	30·7	532	651
1¾	3/16	0·328	0·384	41·0	459	562
1¾	⅛	0·219	0·257	61·4	377	462
1⅝	¼	0·406	0·476	33·1	497	608
1⅝	3/16	0·305	0·357	44·1	430	527
1⅝	⅛	0·203	0·238	66·2	351	430
1½	½	0·750	0·879	17·9	676	828
1½	⅜	0·563	0·660	23·9	575	704
1½	¼	0·375	0·440	35·8	463	567

(The British Aluminium Co. Ltd.)

(1) Based on an average resistivity of 2·485 microhms per cm. cube.

Weights per Square Foot.

Gauge and Thickness				Weight	Gauge and Thickness			Weight
S.W.G.	Fractions of an Inch	In.	Mm.	Lb. per sq. ft.	S.W.G.	In.	Mm.	Lb. per sq. ft.
...	3/8	0·375	9·53	5·29	22	0·028	0·711	0·395
3/0	...	0·372	9·45	5·24	23	0·024	0·610	0·338
2/0	...	0·348	8·84	4·91	24	0·022	0·559	0·310
1/0	...	0·324	8·23	4·57	25	0·020	0·508	0·282
...	5/16	0·312	7·93	4·40	26	0·018	0·457	0·254
1	...	0·300	7·62	4·23	27	0·0164	0·417	0·231
2	...	0·276	7·01	3·89	28	0·0148	0·376	0·209
3	...	0·252	6·40	3·55	29	0·0136	0·346	0·192
...	1/4	0·250	6·35	3·52	30	0·0124	0·315	0·175
4	...	0·232	5·89	3·27	31	0·0116	0·294	0·164
5	...	0·212	5·38	2·99	32	0·0108	0·274	0·152
6	...	0·192	4·88	2·71	33	0·0105	0·267	0·148
...	3/16	0·187	4·75	2·64	34	0·0092	0·233	0·130
7	...	0·176	4·47	2·48	35	0·0084	0·213	0·118
8	...	0·160	4·06	2·26	36	0·0076	0·193	0·107
9	...	0·144	3·66	2·03	37	0·0068	0·172	0·0959
10	...	0·128	3·25	1·80	38	0·0060	0·152	0·0846
...	1/8	0·125	3·18	1·76	39	0·0052	0·132	0·0733
11	...	0·116	2·95	1·64	40	0·0048	0·122	0·0677
12	...	0·104	2·64	1·47	41	0·0044	0·112	0·0620
13	...	0·092	2·34	1·30	42	0·0040	0·102	0·0564
14	...	0·080	2·03	1·13	43	0·0036	0·0915	0·0508
15	...	0·072	1·83	1·02	44	0·0032	0·0813	0·0451
16	...	0·064	1·63	0·902	45	0·0028	0·0711	0·0395
...	1/16	0·062	1·58	0·874	46	0·0024	0·0610	0·0338
17	...	0·056	1·42	0·790	47	0·0020	0·0508	0·0282
18	...	0·048	1·22	0·677	48	0·0016	0·0406	0·0226
19	...	0·040	1·02	0·564	49	0·0012	0·0305	0·0169
20	...	0·036	0·914	0·508	50	0·0010	0·0254	0·0141
21	...	0·032	0·813	0·451				

(The British Aluminium Co. Ltd.)

Aluminium Alloy Sheet.

	BA/60A	BA/40D	BA/20	MG/7	Birmabright
Specific Gravity . .	2·73	2·67	2·72	2·63	2·68

WEIGHT OF ANGLE AND TEE IRON PER FOOT RUN.

Breadth (Flanges Added).	Thickness in Fractions of an Inch.													
	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
2	1·13	1·46	1·76	2·03										
2¼	1·29	1·67	2·02	2·34										
2½	1·45	1·88	2·28	2·66	3·01									
2¾	1·60	2·08	2·54	2·97	3·37									
3	1·76	2·29	2·80	3·28	3·74	4·17								
3¼	1·91	2·50	3·06	3·59	4·10	4·58								
3½	2·07	2·71	3·32	3·91	4·47	5·00	5·51							
3¾	2·23	2·92	3·58	4·22	4·83	5·42	5·98							
4	2·38	3·13	3·84	4·53	5·20	5·83	6·45	7·03						
4¼	2·54	3·33	4·10	4·84	5·56	6·25	6·91	7·55						
4½	2·70	3·54	4·36	5·16	5·92	6·67	7·38	8·07	8·74					
4¾	2·85	3·75	4·62	5·47	6·29	7·08	7·85	8·59	9·31					
5	3·01	3·96	4·88	5·78	6·65	7·50	8·32	9·11	9·88	10·63				
5¼	3·16	4·17	5·14	6·09	7·02	7·92	8·79	9·64	10·46	11·25				
5½	3·32	4·38	5·40	6·41	7·38	8·33	9·26	10·16	11·03	11·88	12·70			
5¾	3·48	4·58	5·66	6·72	7·75	8·75	9·73	10·68	11·60	12·50	13·37			
6	3·63	4·79	5·92	7·03	8·11	9·17	10·20	11·20	12·17	13·13	14·05	14·95		
6¼	3·79	5·00	6·18	7·34	8·48	9·58	10·66	11·72	12·75	13·75	14·73	15·68		
6½	3·95	5·21	6·45	7·66	8·84	10·00	11·13	12·24	13·32	14·38	15·40	16·41	17·38	
6¾	4·10	5·42	6·71	7·97	9·21	10·42	11·60	12·76	13·89	15·00	16·08	17·13	18·16	
7	4·26	5·63	6·97	8·28	9·57	10·83	12·07	13·28	14·47	15·63	16·76	17·86	18·95	20·00
7¼	4·41	5·83	7·23	8·59	9·93	11·25	12·54	13·80	15·04	16·25	17·43	18·59	19·73	20·83
7½	4·57	6·04	7·49	8·91	10·30	11·67	13·01	14·32	15·61	16·88	18·11	19·32	20·51	21·67
7¾	4·73	6·25	7·75	9·22	10·66	12·08	13·48	14·84	16·18	17·50	18·79	20·05	21·29	22·50
8	4·88	6·46	8·01	9·53	11·03	12·50	13·95	15·36	16·76	18·13	19·47	20·78	22·07	23·33
8¼	5·04	6·67	8·27	9·84	11·39	12·92	14·41	15·89	17·33	18·75	20·14	21·51	22·85	24·17
8½	5·20	6·88	8·53	10·16	11·76	13·33	14·88	16·41	17·90	19·38	20·82	22·24	23·63	25·00
8¾	5·35	7·08	8·79	10·47	12·12	13·75	15·35	16·93	18·48	20·00	21·50	22·97	24·41	25·83
9	5·51	7·29	9·05	10·78	12·49	14·17	15·82	17·45	19·05	20·63	22·17	23·70	25·20	26·67
9¼	5·66	7·50	9·31	11·09	12·85	14·58	16·29	17·97	19·62	21·25	22·85	24·43	25·98	27·50
9½	5·82	7·71	9·57	11·41	13·22	15·00	16·76	18·49	20·19	21·88	23·53	25·16	26·76	28·33
9¾	5·98	7·92	9·83	11·72	13·58	15·42	17·23	19·01	20·77	22·50	24·20	25·88	27·54	29·17

WEIGHT OF ANGLE AND TEE IRON PER FOOT RUN (*continued*).

Breadth (Flanges Added).	Thickness in Fractions of an Inch.													
	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
10	6·13	8·13	10·09	12·03	13·95	15·83	17·70	19·53	21·34	23·13	24·88	26·61	28·32	30·00
10¼	6·29	8·33	10·35	12·34	14·31	16·25	18·16	20·05	21·91	23·75	25·56	27·34	29·10	30·83
10½	6·45	8·54	10·61	12·66	14·67	16·67	18·63	20·57	22·49	24·38	26·24	28·07	29·88	31·67
10¾	6·60	8·75	10·87	12·97	15·04	17·08	19·10	21·09	23·06	25·00	26·91	28·80	30·66	32·50
11	6·76	8·96	11·13	13·28	15·40	17·50	19·57	21·61	23·62	25·63	27·59	29·53	31·45	33·33
11¼	..	9·17	11·39	13·59	15·77	17·92	20·04	22·13	24·21	26·25	28·27	30·26	32·23	34·17
11½	..	9·38	11·65	13·91	16·13	18·33	20·51	22·66	24·78	26·88	28·95	30·99	33·01	35·00
11¾	..	..	11·91	14·22	16·50	18·75	20·98	23·18	25·35	27·50	29·62	31·72	33·79	35·83
12	..	..	12·17	14·53	16·86	19·17	21·45	23·70	25·92	28·13	30·30	32·45	34·57	36·67
12¼	..	..	..	14·84	17·23	19·58	21·91	24·22	26·50	28·75	30·98	33·18	35·35	37·50
12½	..	..	..	15·16	17·59	20·00	22·38	24·74	27·07	29·38	31·65	33·91	36·13	38·33
12¾	..	..	..	..	17·96	20·42	22·85	25·26	27·64	30·00	32·33	34·64	36·91	39·17
13	..	..	..	..	18·32	20·83	23·32	25·78	28·22	30·63	33·01	35·36	37·70	40·00
13½	..	..	..	..	..	21·67	24·26	26·82	29·36	31·88	34·36	36·82	39·26	41·67
14	..	..	..	..	..	22·50	25·20	27·86	30·51	33·13	35·72	38·28	40·82	43·33
14½	..	..	..	..	..	23·33	26·13	28·91	31·65	34·38	37·07	39·74	42·38	45·00
15	..	..	..	..	..	24·17	27·07	29·95	32·80	35·63	38·42	41·20	43·95	46·67

(*Henderson & Glass.*)

WEIGHT OF ANGLE AND TEE STEEL PER FOOT RUN.

Breadth (Flanges added.)	Thickness in Fractions of an Inch.											
	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
2	1·16	1·49	1·79	2·07								
2¼	1·32	1·70	2·06	2·39								
2½	1·47	1·91	2·32	2·71	3·07							
2¾	1·63	2·12	2·59	3·03	3·44							
3	1·79	2·34	2·85	3·35	3·81	4·25						
3¼	1·96	2·55	3·12	3·67	4·18	4·68						
3½	2·11	2·76	3·39	3·98	4·56	5·10	5·62					
3¾	2·27	2·98	3·65	4·30	4·93	5·53	6·10					
4	2·43	3·19	3·92	4·62	5·30	5·95	6·57	7·17				
4¼	2·59	3·40	4·18	4·94	5·67	6·38	7·05	7·70				

WEIGHT OF ANGLE AND TEE STEEL PER FOOT RUN—*continued.*

Breadth (Flanges added).	Thickness in Fractions of an Inch.											
	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
4½	2·75	3·61	4·45	5·26	6·05	6·80	7·53	8·23	8·91			
4¾	2·91	3·83	4·72	5·58	6·42	7·23	8·01	8·77	9·50			
5	3·07	4·04	4·98	5·90	6·78	7·65	8·49	9·30	10·08	10·84		
5¼	3·23	4·25	5·25	6·22	7·16	8·08	8·97	9·83	10·67	11·48		
5½	3·39	4·46	5·51	6·54	7·53	8·50	9·44	10·34	11·20	12·11	12·95	
5¾	3·55	4·68	5·78	6·85	7·90	8·93	9·92	10·89	11·83	12·75	13·64	
6	3·71	4·89	6·05	7·17	8·28	9·36	10·40	11·42	12·42	13·39	14·33	15·26
6¼	3·87	5·10	6·31	7·49	8·65	9·78	10·88	11·95	13·00	14·03	15·02	15·99
6½	4·02	5·31	6·57	7·81	9·02	10·20	11·34	12·48	13·59	14·66	15·71	16·73
6¾	4·18	5·53	6·84	8·13	9·39	10·63	11·83	13·02	14·17	15·30	16·40	17·48
7	4·34	5·74	7·11	8·45	9·76	11·05	12·31	13·55	14·76	15·94	17·09	18·22
7¼	4·50	5·95	7·37	8·77	10·13	11·48	12·79	14·08	15·34	16·58	17·78	18·97
7½	4·66	6·16	7·64	9·08	10·51	11·90	13·27	14·61	15·92	17·21	18·47	19·71
7¾	4·82	6·38	7·90	9·40	10·88	12·33	13·75	15·14	16·51	17·85	19·17	20·45
8	4·98	6·59	8·17	9·72	11·25	12·75	14·22	15·67	17·09	18·49	19·86	21·20
8¼	5·14	6·80	8·43	10·04	11·62	13·18	14·70	16·20	17·68	19·13	20·55	21·94
8½	5·30	7·01	8·70	10·36	11·99	13·60	15·18	16·73	18·26	19·76	21·24	22·68
8¾	5·46	7·23	8·97	10·68	12·37	14·03	15·66	17·27	18·85	20·40	21·93	23·43
9	5·62	7·44	9·23	11·00	12·74	14·45	16·14	17·80	19·43	21·04	22·62	24·17
9¼	5·78	7·65	9·50	11·32	13·11	14·88	16·62	18·33	20·02	21·68	23·31	24·92
9½	5·94	7·86	9·76	11·63	13·48	15·30	17·09	18·86	20·60	22·31	24·00	25·66
9¾	6·10	8·08	10·03	11·95	13·85	15·73	17·57	19·39	21·18	22·95	24·69	26·40
10	6·26	8·29	10·30	12·28	14·22	16·16	18·05	19·93	21·77	23·59	25·38	27·15
10¼	6·42	8·50	10·56	12·59	14·60	16·58	18·53	20·45	22·35	24·23	26·07	27·89
10½	6·57	8·71	10·82	12·91	14·97	17·00	19·01	20·98	22·94	24·86	26·76	28·63
10¾	6·73	8·93	11·09	13·23	15·34	17·43	19·48	21·52	23·52	25·50	27·45	29·38
11	6·89	9·14	11·36	13·55	15·71	17·85	19·96	22·05	24·11	26·14	28·14	30·10
11¼	—	9·35	11·62	13·87	16·08	18·38	20·44	22·58	24·69	26·78	28·83	30·87
11½	—	9·56	11·89	14·18	16·46	18·70	20·92	23·11	25·27	27·41	29·52	31·61
11¾	—	—	12·15	14·50	16·83	19·13	21·40	23·64	25·86	28·05	30·21	32·36
12	—	—	12·42	14·82	17·20	19·55	21·87	24·17	26·44	28·69	30·91	33·10

(*Henderson & Glass.*)

EBONITE SHEET.

Thickness.			Approx. Weight per Sheet.					
			36 × 18 ins. (914 × 457 m/m.)		24 × 24 ins. (610 × 610 m/m.)		36 × 24 ins. (914 × 610 m/m.)	
Ins.	Ins.	m/m.	lbs.	kgs.	lbs.	kgs.	lbs.	kgs.
...	0·015	0·38	0·42	0·19	0·37	0·17	0·56	0·25
...	0·025	0·64	0·70	0·32	0·63	0·29	0·94	0·43
...	0·040	1·02	1·12	0·51	1·00	0·45	1·50	0·68
...	0·660	1·52	1·68	0·76	1·50	0·68	2·25	1·02
...	0·090	2·29	2·52	1·14	2·25	1·02	3·37	1·53
1/8	0·125	3·18	3·51	1·59	3·12	1·42	4·69	2·13
3/16	0·187	4·75	5·26	2·39	4·67	2·12	7·01	3·18
1/4	0·250	6·35	7·02	3·18	6·25	2·84	9·38	4·26
5/16	0·312	7·93	8·77	3·98	7·80	3·54	11·70	5·31
3/8	0·375	9·53	10·55	4·79	9·37	4·25	14·06	6·38
7/16	0·437	11·10	12·29	5·58	10·92	4·95	16·40	7·44
1/2	0·500	12·70	14·04	6·37	12·50	5·67	18·75	8·51
5/8	0·625	15·88	17·55	7·96	15·62	7·09	23·43	10·63
3/4	0·750	19·05	21·06	9·55	18·74	8·50	28·12	12·76
7/8	0·875	22·23	24·60	11·16	21·86	9·91	32·80	14·88
1	1·000	25·40	28·12	12·76	25·00	11·34	37·50	17·01

EBONITE ROD.

Diameter.			Approx. Weight per 100 metres.	Diameter			Approx. Weight per 100 metres.
Ins.	Ins.	m/m.	Lbs.	Ins.	Ins.	m/m.	Lbs.
3/32	0·0937		1·18	5/8	0·625		52·35
	0·0984	2½	1·29		0·630	16	53·17
1/8	0·125		2·09	21/32	0·656		57·67
	0·137	3½	2·54		0·669	17	60·00
5/32	0·157	4	3·23	11/16	0·687		63·25
3/16	0·187		4·69		0·709	18	67·30
7/32	0·217	5½	6·28	23/32	0·719		69·27
1/4	0·250		8·37		0·728	18½	71·08
	0·256	6½	8·77	3/4	0·750		75·37
9/32	0·282		10·65		0·768	19½	78·98
	0·295	7½	11·68	25/32	0·781		81·72
5/16	0·312		13·05		0·787	20	83·08
	0·315	8	13·29	13/16	0·812		88·35
11/32	0·344		15·86		0·827	21	91·60
3/8	0·375	9½	18·85	27/32	0·844		95·45
13/32	0·406		22·09		0·866	22	100·5
	0·413	10½	22·90	7/8	0·875		102·6
7/16	0·437		25·60	29/32	0·905	23	110·0
	0·453	11½	27·47	15/16	0·937		117·6
15/32	0·468		29·35		0·945	24	120·0
	0·472	12	29·90	1	1·000	25·4	134·0
1/2	0·500		33·50	1⅛	1·125	28·5	170
17/32	0·532	13½	37·85	1¼	1·25	31·8	210
9/16	0·562		42·33	1⅜	1·375	35·0	253
	0·571	14½	43·66	1½	1·5	38·0	302
19/32	0·594		47·28	1¾	1·75	44·5	410
	0·610	15½	49·90	2	2	50·8	536

(Siemens Bros. & Co.)

Weight of Pipes and Tubes.

D = outer diameter; b = bore; t = thickness; W = weight of 1 ft. length; k = ·02182 × weight of 1 cubic foot of metal.

$$W = t(D - t)k = t(b + t)k.$$

Approximate Weights of Cold Drawn Weldless Steel Tubes in Lb. per Foot. Plain at Ends.

Outside Dia. in Ins.	Thicknesses in Fractions of an Inch. 1/16	3/32	1/8	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8
1	·630	·914	1·176	1·638	2·017						
1 1/8	·714	1·04	1·344	1·89	2·353	2·731					
1 1/4	·798	1·166	1·512	2·142	2·689	3·151	3·529				
1 3/8	·883	1·292	1·68	2·394	3·025	3·571	4·033	4·411			
1 1/2	·967	1·418	1·848	2·646	3·361	5·991	4·537	5·000	5·378		
1 5/8	1·051	1·544	2·016	2·898	3·697	4·411	5·041	5·588	6·05		
1 3/4	1·135	1·67	2·184	3·15	4·034	4·831	5·545	6·176	6·723	7·184	
1 7/8	1·219	1·796	2·352	3·402	4·37	5·252	6·050	6·764	7·395	7·941	
2	1·303	1·922	2·52	3·654	4·706	5·672	6·554	7·352	8·067	8·697	9·243
2 1/8	1·387	2·048	2·688	3·906	5·042	6·092	7·058	7·941	8·739	9·453	10·083
2 1/4	1·471	2·174	2·856	4·158	5·378	6·512	7·562	8·529	9·411	10·209	10·923
2 3/8	1·555	2·3	3·024	4·41	5·714	6·932	8·066	9·117	10·084	10·966	11·763
2 1/2	1·639	2·426	3·192	4·662	6·050	7·352	8·57	9·705	10·756	11·722	12·603
2 5/8	1·723	2·552	3·36	4·914	6·386	7·772	9·074	10·293	11·428	12·478	13·444
2 3/4	1·807	2·678	3·528	5·166	6·723	8·192	9·578	10·881	12·100	13·234	14·284
2 7/8	1·891	2·804	3·696	5·418	7·059	8·613	10·083	11·470	12·773	13·991	15·125
3	1·975	2·93	3·864	5·67	7·395	9·033	10·587	12·058	13·445	14·747	15·965
3 1/8		3·056	4·032	5·922	7·731	9·453	11·091	12·646	14·117	15·503	16·805
3 1/4		3·182	4·20	6·174	8·067	9·873	11·595	13·234	14·79	16·259	17·645
3 3/8		3·308	4·368	6·426	8·403	10·293	12·099	13·822	15·462	17·016	18·486
3 1/2		3·434	4·536	6·678	8·739	10·713	12·603	14·410	16·134	17·772	19·326
3 5/8			4·704	6·93	9·075	11·133	13·107	14·999	16·806	18·528	20·166
3 3/4			4·872	7·182	9·412	11·553	13·611	15·587	17·478	19·284	21·006
3 7/8			5·04	7·434	9·748	11·974	14·116	16·175	18·151	20·041	21·847
4			5·208	7·686	10·084	12·394	14·620	16·763	18·823	20·797	22·687
4 1/8			5·376	7·938	10·420	12·814	15·124	17·351	19·495	21·553	23·527
4 1/4			5·544	8·19	10·756	13·234	15·628	17·940	20·167	22·309	24·367
4 3/8			5·712	8·442	11·092	13·654	16·132	18·528	20·84	23·066	25·208
4 1/2			5·880	8·694	11·428	14·074	16·636	19·116	21·512	23·822	26·048

Standard Weights and Dimensions of Cast-iron Water-main Pipes.

Internal Diam. of Pipe.	Length of each Pipe.		Depth of Socket.	Thickness of Metal.	Mean Weight of each Pipe.			Internal Diam. of Pipe.	Length of each Pipe.		Depth of Socket.	Thickness of Metal.	Mean Weight of each Pipe.		
Ins.	Ft.	Ins.	Ins.	Ins.	cwt.	qrs.	lbs.	Ins.	Ft.	Ins.	Ins.	Ins.	cwt.	qrs.	lbs.
2	6	3	3	5/16	0	2	0	18	12	4 1/2	4 1/2	3/4	14	2	16
3	9	4	3	3/8	1	0	14	19	12	4 1/2	4 1/2	—		—	
4	9	4	3	3/8	1	2	0	20	12	4 1/2	4 1/2	3/4	20	0	0
5	9	4	3 1/2	7/16	2	0	0	21	12	4 1/2	4 1/2	—		—	
6	9	4	3 1/2	7/16	2	2	0	22	12	4 1/2	4 1/2	3/4	22	3	14
7	9	4	3 1/2	1/2	3	1	14	23	12	4	5	—		—	
8	9	4	4	1/2	3	2	0	24	12	4	5	7/8	24	0	0
9	9	4	4	1/2	4	2	0	26	12	5	5	—		—	
10	9	4	4	9/16	5	1	0	28	12	5	5	—		—	
11	12	4	4 1/4	5/8		—		30	12	5	5	1	35	3	0
12	12	4 1/2	4 1/4	5/8	8	2	0	32	12	5	5	—		—	
13	12	4 1/2	4 1/4	—		—		34	12	5	5	—		—	
14	12	4 1/2	4 1/4	5/8	10	2	0	36	12	5	5	1	41	0	0
15	12	4 1/2	4 1/2	5/8	11	2	0	40	12	5	5	1 1/8	48	2	0
16	12	4 1/2	4 1/2	5/8	13	2	0	42	12	5	5	1 1/8	50	2	0
17	12	4 1/2	4 1/2	—		—		48	12	5 1/2	5 1/2	1 1/8	64	0	0

Weight of Ordinary Brass Pipes per Foot Run

$k = 12\cdot105.$

Bore. (b)	Thickness of Metal in Parts of an Inch (t). $\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{1}{4}$	·22	·53	·94	1·43	2·01	2·68	3·44
$\frac{1}{2}$	·40	·89	1·47	2·15	2·91	3·75	4·70
$\frac{3}{4}$	·58	1·25	2·01	2·86	3·80	4·83	5·95
1	·76	1·61	2·55	3·58	4·70	5·92	7·25
$1\frac{1}{4}$	·94	1·96	3·09	4·31	5·64	6·98	8·46
$1\frac{1}{2}$	1·12	2·34	3·67	5·01	6·49	8·05	9·71
$1\frac{3}{4}$	1·33	2·66	4·14	5·70	7·36	9·11	10·94
2	1·48	3·04	4·69	6·44	8·27	10·20	12·21
$2\frac{1}{4}$	1·65	3·40	5·23	7·16	9·17	11·27	13·46
$2\frac{1}{2}$	1·83	3·75	5·77	7·87	10·06	12·35	14·72
$2\frac{3}{4}$	2·01	4·11	6·31	8·59	10·96	13·42	15·97
3	2·19	4·47	6·84	9·31	11·85	14·69	17·42

Weight of Ordinary Copper Pipes per Foot Run.

$k = 11\cdot452.$

Bore. (b)	Thickness of Metal in Parts of an Inch (t). $\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{1}{4}$	·23	·56	·99	1·51	2·13	2·83	3·64
$\frac{1}{2}$	·42	·94	1·56	2·27	3·07	3·97	4·96
$\frac{3}{4}$	·61	1·32	2·13	3·02	4·02	5·11	6·29
1	·80	1·70	2·69	3·78	4·96	6·24	7·61
$1\frac{1}{4}$	·99	2·08	3·26	4·54	5·91	7·38	8·94
$1\frac{1}{2}$	1·18	2·46	3·83	5·29	6·85	8·51	10·26
$1\frac{3}{4}$	1·37	2·84	4·40	6·05	7·80	9·64	11·58
2	1·56	3·22	4·96	6·81	8·75	10·78	12·91
$2\frac{1}{4}$	1·75	3·59	5·53	7·56	9·69	11·92	14·23
$2\frac{1}{2}$	1·94	3·97	6·10	8·32	10·64	13·06	15·56
$2\frac{3}{4}$	2·13	4·35	6·67	9·08	11·59	14·19	16·88
3	2·31	4·73	7·24	9·83	12·53	15·32	18·21

Weight of Lead Pipes, Sizes and Weights usually Manufactured.

Bore.	Length.	Weight and Thickness of each Length. Common.		Middling.		Strong.	
		Thickness.	Weight.	Thickness.	Weight.	Thickness.	Weight.
Ins.	Ft.	In.	Lbs.	In.	Lbs.	In.	Lbs.
$\frac{1}{2}$	15	·11	16	·15	22	·17	26
$\frac{3}{4}$	15	·12	24	·13	27	·15	30
1	15	·12	30	·15	39	·16	42
$1\frac{1}{4}$	12	·14	36	·16	44	·20	53
$1\frac{1}{2}$	12	·16	48	·18	56	·21	67
2	10	·15	50	·18	60	·21	70
$2\frac{1}{2}$	10	·17	70	·21	86	·24	100

Formulæ for Weight of Lead Piping.

$15\cdot517\,(D-t)\,t$ lbs. per foot; $81930\,(D-t)\,t$ lbs. per mile; $36\cdot576\,(D-t)\,t$ tons per mile; where D = external diameter in inches, t = thickness in inches.

APPROXIMATE WEIGHTS
(Based on the Imperial

Thickness, S.W.G.	15	14	13	12	11	10	9	8	7	6	5
Ins. (*t*)	·072	·080	·092	·104	·116	·128	·144	·160	·176	·192	·212
m/m.	1·829	2·032	2·337	2·642	2·946	3·251	3·658	4·064	4·470	4·877	5·385
External Diam. Ins. (*D*).											WEIGHT PER
1	0·700	0·771	0·875	0·976	1·074	1·169	1·291	1·407	1·519	1·624	1·749
1⅛	0·794	0·875	0·995	1·112	1·226	1·336	1·479	1·617	1·749	1·876	2·027
1¼	0·888	0·980	1·116	1·248	1·377	1·504	1·668	1·826	1·979	2·127	2·304
1⅜	0·982	1·085	1·236	1·384	1·529	1·671	1·856	2·036	2·210	2·378	2·582
1½	1·077	1·190	1·356	1·520	1·681	1·839	2·045	2·245	2·440	2·630	2·859
1⅝	1·171	1·294	1·477	1·656	1·833	2·007	2·233	2·455	2·671	2·881	3·137
1¾	1·265	1·399	1·597	1·793	1·985	2·174	2·422	2·664	2·901	3·132	3·414
1⅞	1·359	1·504	1·718	1·929	2·137	2·342	2·610	2·873	3·131	3·384	3·692
2	—	1·608	1·838	2·065	2·288	2·509	2·799	3·083	3·362	3·635	3·969
2⅛	—	1·713	1·959	2·201	2·440	2·677	2·987	3·292	3·592	3·886	4·247
2¼	—	1·818	2·079	2·337	2·592	2·844	3·176	3·502	3·822	4·138	4·524
2⅜	—	—	2·199	2·473	2·744	3·012	3·364	3·711	4·053	4·389	4·802
2½	—	—	2·320	2·609	2·896	3·179	3·553	3·921	4·283	4·640	5·079
2⅝	—	—	2·440	2·745	3·048	3·347	3·741	4·130	4·514	4·892	5·357
2¾	—	—	2·561	2·882	3·200	3·514	3·930	4·339	4·744	5·143	5·634
2⅞	—	—	2·681	3·018	3·351	3·682	4·118	4·549	4·974	5·394	5·912
3	—	—	2·802	3·154	3·503	3·850	4·307	4·758	5·205	5·646	6·189
3⅛	—	—	—	3·290	3·655	4·017	4·495	4·968	5·435	5·897	6·467
3¼	—	—	—	3·426	3·807	4·185	4·684	5·177	5·665	6·148	6·744
3⅜	—	—	—	3·562	3·959	4·352	4·872	5·387	5·896	6·400	7·022
3½	—	—	—	3·698	4·111	4·520	5·061	5·596	6·126	6·651	7·299
3⅝	—	—	—	3·835	4·262	4·687	5·249	5·806	6·357	6·902	7·577
3¾	—	—	—	3·971	4·414	4·855	5·438	6·015	6·587	7·154	7·854
3⅞	—	—	—	4·107	4·566	5·022	5·626	6·224	6·817	7·405	8·132
4	—	—	—	4·247	4·718	5·190	5·815	6·434	7·048	7·656	8·410
4¼	—	—	—	4·506	5·022	5·525	6·192	6·853	7·509	8·159	8·965
4½	—	—	—	4·776	5·325	5·860	6·569	7·272	7·969	8·662	9·520
4¾	—	—	—	5·056	5·631	6·195	6·946	7·690	8·430	9·164	10·075
5	—	—	—	5·336	5·937	6·530	7·323	8·109	8·891	9·667	10·630
5¼	—	—	—	5·600	6·243	6·866	7·700	8·528	9·352	10·170	11·185
5½	—	—	—	5·876	6·549	7·201	8·077	8·947	9·812	10·672	11·740
5¾	—	—	—	6·152	6·857	7·536	8·454	9·366	10·273	11·175	12·295
6	—	—	—	6·432	7·164	7·871	8·831	9·785	10·734	11·678	12·850
6¼	—	—	—	6·715	7·471	8·213	9·208	10·204	11·195	12·180	13·405
6½	—	—	—	7·000	7·778	8·555	9·585	10·623	11·655	12·683	13·960
6¾	—	—	—	7·273	8·087	8·897	9·962	11·042	12·116	13·186	14·515
7	—	—	—	7·556	8·393	9·239	10·339	11·460	12·577	13·688	15·070
7¼	—	—	—	7·834	8·700	9·581	10·716	11·880	13·038	14·191	15·625
7½	—	—	—	8·115	9·[illegible]	9·923	11·083	12·300	13·499	14·694	16·180
7¾	—	—	—	8·383	9·314	10·265	11·460	12·720	13·959	15·196	16·735
8	—	—	—	8·664	9·621	10·6[illegible]7	11·847	13·135	14·420	15·699	17·290
8¼	—	—	—	8·943	9·928	10·949	12·224	13·554	14·880	16·204	17·845
8½	—	—	—	9·227	10·235	11·291	12·601	13·973	15·340	16·710	18·400
8¾	—	—	—	9·516	10·542	11·633	12·978	14·400	15·800	17·216	18·955
9	—	—	—	9·807	10·850	11·985	13·355	14·818	16·260	17·722	19·510
9¼	—	—	—	10·073	11·158	12·327	13·732	15·230	16·720	18·230	20·065
9½	—	—	—	10·342	11·466	12·669	14·109	16·656	17·180	18·738	20·620
9¾	—	—	—	10·604	11·774	13·011	14·486	16·070	17·640	19·230	21·175
10	—	—	—	10·878	12·082	13·353	14·863	16·507	18·100	19·735	21·730
10¼	—	—	—	11·156	12·390	13·695	15·240	16·922	18·563	20·242	22·285
10½	—	—	—	11·428	12·698	14·042	15·617	17·344	19·024	20·753	22·840
10¾	—	—	—	11·687	13·006	14·384	15·994	17·768	19·482	21·256	23·395
11	—	—	—	11·952	13·314	14·726	16·371	18·185	19·940	21·756	23·950
11½	—	—	—	12·516	13·930	15·410	17·125	19·016	20·860	22·758	25·053
12	—	—	—	13·050	14·550	16·100	17·878	19·855	21·780	23·760	26·200

OF WROUGHT IRON TUBES. (For Steel, add 2 %.)
Standard Wire Gauge.) $W = t(D - t)\,10{\cdot}472.$

FOOT IN POUNDS (W).

4 ·232 5·893	3 ·252 6·401	2 ·276 7·010	1 ·300 7·620	1/8 ·125 3·175	3/16 ·187 4·762	1/4 ·250 6·350	5/16 ·313 7·937	3/8 ·375 9·525	7/16 ·437 11·112	1/2 ·500 12·700	Thickness, S.W.G. Ins. (t) m/m. External Dia. Ins. (D).
1·866	1·974	2·092	2·199	1·145	1·595	1·963	2·250	2·454	2·577	2·618	1
2·169	2·304	2·451	2·592	1·309	1·841	2·291	2·659	2·945	3·150	3·272	1 1/8
2·473	2·634	2·815	2·984	1·473	2·086	2·618	3·068	3·436	3·723	3·927	1 1/4
2·777	2·963	3·176	3·377	1·636	2·332	2·945	3·477	3·927	4·295	4·581	1 3/8
3·081	3·293	3·538	3·770	1·800	2·577	3·272	3·886	4·418	4·868	5·236	1 1/2
3·384	3·623	3·899	4·163	1·963	2·822	3·600	4·295	4·909	5·440	5·890	1 5/8
3·688	3·953	4·260	4·555	2·127	3·068	3·927	4·704	5·400	6·013	6·545	1 3/4
3·992	4·283	4·621	4·948	2·291	3·313	4·254	5·113	5·890	6·586	7·199	1 7/8
4·295	4·613	4·983	5·341	2·454	3·559	4·581	5·522	6·381	7·159	7·854	2
4·599	4·943	5·344	5·733	2·618	3·804	4·909	5·931	6·872	7·731	8·508	2 1/8
4·903	5·273	5·705	6·126	2·782	4·050	5·236	6·340	7·363	8·304	9·163	2 1/4
5·206	5·602	6·067	6·519	2·945	4·295	5·563	6·749	7·854	8·877	9·817	2 3/8
5·510	5·932	6·428	6·911	3·109	4·541	5·890	7·159	8·345	9·449	10·472	2 1/2
5·814	6·262	6·789	7·304	3·272	4·786	6·218	7·568	8·836	10·022	11·126	2 5/8
6·117	6·592	7·150	7·697	3·436	5·031	6·545	7·977	9·327	10·595	11·781	2 3/4
6·421	6·922	7·512	8·090	3·600	5·277	6·872	8·386	9·818	11·167	12·435	2 7/8
6·725	7·252	7·873	8·482	3·763	5·522	7·200	8·795	10·308	11·740	13·090	3
7·028	7·582	8·234	8·875	3·927	5·768	7·527	9·204	10·799	12·313	13·744	3 1/8
7·332	7·911	8·596	9·268	4·091	6·013	7·854	9·613	11·290	12·885	14·399	3 1/4
7·636	8·241	8·957	9·660	4·254	6·259	8·181	10·022	11·781	13·458	15·053	3 3/8
7·940	8·571	9·318	10·053	4·418	6·504	8·508	10·431	12·272	14·031	15·708	3 1/2
8·243	8·901	9·679	10·446	4·581	6·750	8·836	10·840	12·763	14·604	16·362	3 5/8
8·547	9·231	10·041	10·838	4·745	6·995	9·163	11·249	13·254	15·176	17·017	3 3/4
8·851	9·561	10·402	11·231	4·908	7·240	9·490	11·658	13·745	15·749	17·671	3 7/8
9·154	9·891	10·763	11·624	5·072	7·486	9·817	12·067	14·235	16·322	18·326	4
9·762	10·550	11·486	12·409	5·400	7·977	10·472	12·885	15·217	17·467	19·635	4 1/4
10·369	11·210	12·208	13·195	5·727	8·468	11·126	13·704	16·199	18·612	20·944	4 1/2
10·976	11·870	12·931	13·980	6·054	8·959	11·781	14·522	17·181	19·758	22·253	4 3/4
11·584	12·530	13·654	14·765	6·381	9·450	12·435	15·340	18·162	20·903	23·562	5
12·191	13·189	14·376	15·551	6·709	9·940	13·090	16·158	19·144	22·048	24·871	5 1/4
12·798	13·849	15·099	16·336	7·036	10·431	13·744	16·976	20·126	23·194	26·180	5 1/2
13·406	14·509	15·821	17·122	7·363	10·922	14·399	17·794	21·108	24·339	27·489	5 3/4
14·013	15·169	16·544	17·907	7·690	11·413	15·053	18·612	22·090	25·485	28·798	6
14·621	15·828	17·266	18·692	8·017	11·904	15·708	19·430	23·071	26·630	30·107	6 1/4
15·228	16·488	17·989	19·478	8·345	12·395	16·362	20·249	24·053	27·775	31·416	6 1/2
15·835	17·184	18·712	20·263	8·672	12·886	17·017	21·067	25·035	28·921	32·725	6 3/4
16·443	17·807	19·434	21·048	8·999	13·376	17·671	21·885	26·017	30·066	34·034	7
17·050	18·467	20·157	21·834	9·327	13·867	18·326	22·703	26·998	31·212	35·343	7 1/4
17·658	19·127	20·879	22·619	9·654	14·358	18·980	23·521	27·980	32·357	36·652	7 1/2
18·265	19·787	21·602	23·405	9·981	14·849	19·635	24·339	28·962	33·502	37·961	7 3/4
18·872	20·446	22·324	24·190	10·308	15·340	20·289	25·157	29·944	34·648	39·270	8
19·480	21·106	23·047	24·976	10·636	15·831	20·944	25·975	30·925	35·793	40·579	8 1/4
20·087	21·766	23·769	25·761	10·963	16·322	21·598	26·794	31·907	36·938	41·888	8 1/2
20·694	22·426	24·492	26·546	11·290	16·813	22·253	27·612	32·889	38·084	43·197	8 3/4
21·302	23·085	25·215	27·332	11·617	17·303	22·907	28·430	33·871	39·229	44·506	9
21·909	23·745	25·937	28·117	11·945	17·794	23·562	29·248	34·852	40·375	45·815	9 1/4
22·517	24·405	26·660	28·903	12·272	18·285	24·216	30·066	35·834	41·520	47·124	9 1/2
23·124	25·065	27·382	29·688	12·599	18·776	24·871	30·884	36·816	42·665	48·433	9 3/4
23·731	25·724	28·105	30·473	12·926	19·267	25·525	31·702	37·708	43·811	49·742	10
24·338	26·383	28·828	31·258	13·253	19·758	26·179	32·520	38·780	45·957	51·052	10 1/4
24·945	27·042	29·551	32·043	13·580	20·249	26·833	33·338	39·762	46·103	52·361	10 1/2
25·535	27·702	30·274	32·828	13·910	20·740	27·487	34·156	40·744	47·250	53·670	10 3/4
26·162	28·362	31·002	33·613	14·237	21·231	28·141	34·974	41·726	48·396	54·980	11
27·376	29·682	32·445	35·183	14·897	22·213	29·450	36·610	43·690	50·690	57·600	11 1/2
28·590	31·002	33·891	36·753	15·550	23·195	30·758	38·246	45·654	52·984	60·215	12

(John Spencer, Ltd.)

Weight of Cast-Iron Pipes per Foot Run.
Wt = Thickness × (Bore + Thickness) × 9·85.

Inner Diam. or Bore in Inches.	Thickness of Metal in Inches.												
	¼	⅜	½	⅝	¾	⅞	1	1⅛	1¼	1⅜	1½	1¾	2
	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.	Wt. in Lbs.
1	3·07	5·07	7·38	9·99	12·9	16·2	19·7	23·5	27·7	32·1	36·9	47·4	59·1
¼	3·69	6·00	8·61	11·5	14·8	18·3	22·2	26·3	30·8	35·5	40·6	51·7	64·0
½	4·30	6·92	9·84	13·1	16·6	20·5	24·6	29·1	33·8	38·9	44·3	56·0	68·9
¾	4·92	7·84	11·1	14·6	18·5	22·6	27·1	31·8	36·9	42·3	48·0	60·3	73·8
2	5·53	8·76	12·3	16·2	20·3	24·8	29·5	34·6	40·0	45·7	51·7	64·6	78·7
¼	6·15	9·69	13·5	17·7	22·2	26·9	32·0	37·4	43·1	49·0	55·4	68·9	86·7
½	6·76	10·6	14·8	19·2	24·0	29·1	34·5	40·1	46·1	52·4	59·1	73·2	88·6
¾	7·37	11·5	16·0	20·8	25·9	31·2	36·9	42·9	49·2	55·8	62·7	77·5	93·5
3	7·98	12·5	17·2	22·3	27·7	33·4	39·4	45·7	52·3	59·2	66·4	81·8	98·4
¼	8·60	13·4	18·5	23·8	29·5	35·5	41·8	48·3	55·4	62·3	70·1	86·1	103
½	9·21	14·3	19·7	25·4	31·4	37·7	44·3	51·2	58·4	65·9	73·8	90·4	108
¾	9·83	15·2	20·9	26·9	33·2	39·8	46·8	54·0	61·5	69·3	77·5	94·7	113
4	10·3	16·1	22·2	28·5	35·1	42·0	49·2	56·7	64·6	72·7	81·2	99·0	118
¼	11·1	17·1	23·4	30·0	36·9	44·1	51·7	59·5	67·7	76·1	84·9	103	123
½	11·7	18·0	24·6	31·5	38·8	46·3	54·1	62·3	70·7	79·5	88·6	108	128
¾	12·3	18·9	25·8	33·1	40·6	48·5	56·6	65·0	73·8	83·9	92·3	112	133
5	12·9	19·8	27·1	34·6	42·5	50·6	59·1	67·8	76·9	87·2	96·0	116	138
¼	13·5	20·8	28·3	36·1	44·3	52·8	61·5	70·6	80·0	90·6	99·6	121	143
½	14·2	21·7	29·5	37·7	46·1	54·9	64·0	73·3	83·0	94·0	103	125	148
¾	14·8	22·6	30·8	39·2	48·0	57·1	66·4	76·1	86·1	97·4	107	129	153
6	15·4	23·5	32·0	40·8	49·8	59·2	68·9	78·9	89·2	99·8	111	134	158
½	16·6	25·4	34·5	43·8	53·5	63·5	73·8	84·4	95·3	107	118	142	167
7	17·8	27·2	36·9	46·9	57·2	67·8	78·7	89·4	102	113	126	151	177
½	19·1	29·1	39·4	50·0	60·9	72·1	83·7	95·5	108	120	133	159	187
8	20·3	30·9	41·8	53·1	64·6	76·4	88·6	101	114	127	140	168	197
½	21·5	32·8	44·3	56·1	68·3	80·7	93·5	107	120	134	148	177	207
9	22·8	34·6	46·8	59·2	72·0	85·1	98·4	112	126	140	155	185	217
½	24·0	36·4	49·2	62·3	75·7	89·3	103	118	132	147	163	194	226
10	25·1	38·3	51·7	65·3	79·4	93·6	108	123	138	154	170	202	235
½	26·4	40·1	54·1	68·4	83·0	97·9	113·2	129	145	161	177	211	245
11	27·6	42·0	56·6	71·5	86·7	102	118	134	151	168	185	220	255
½	28·8	43·8	59·1	74·6	90·4	107	123	140	157	174	192	228	265
12	30·0	45·7	61·5	77·7	94·1	111	128	145	163	181	199	237	275
13	32·5	49·4	66·4	83·8	102	120	138	156	175	195	214	254	294
14	35·0	53·1	71·4	89·4	109	128	148	168	188	208	229	271	314
15	37·4	56·7	76·3	96·1	116	137	158	179	200	222	244	289	334
16	39·1	60·4	81·2	102	124	145	167	190	212	235	258	306	353
17	42·3	64·1	86·1	108	131	154	177	201	225	249	273	323	373
18	44·8	67·8	91·0	115	139	163	187	212	237	262	288	340	393
19	47·3	71·5	96·0	121	146	171	197	223	249	276	303	357	412
20	49·7	75·2	101	127	153	180	207	234	261	289	317	375	432
21	52·2	78·9	106	133	161	188	217	245	274	303	332	392	452
22	54·6	82·6	111	139	168	196	227	256	286	316	347	409	471
23	57·1	86·3	116	145	175	206	236	267	298	330	362	426	491
24	59·6	89·9	121	152	183	214	246	278	311	343	375	444	511
25	62·0	93·6	126	158	190	223	256	289	323	357	391	461	531
26	64·5	97·3	131	164	198	231	266	300	335	370	406	478	550
27	66·9	101	135	170	205	240	276	311	348	384	421	495	570
28	69·4	105	140	176	212	249	286	323	360	397	436	512	590
29	71·8	109	145	182	220	257	295	334	372	411	450	530	609
30	74·2	112	150	188	227	266	305	345	384	424	465	547	629
31	76·7	116	155	195	234	275	315	356	397	438	480	564	649
32	79·1	120	160	201	242	283	325	367	409	451	495	581	668
33	81·6	123	165	207	249	292	335	378	421	465	509	598	688
34	84·1	127	170	213	257	300	345	389	434	479	524	616	709
35	86·3	131	175	219	264	509	354	400	446	492	539	633	726
36	89·0	134	180	225	271	318	364	411	458	506	554	650	746
42	104	156	210	262	315	370	423	478	532	588	644	753	864
48	119	178	239	298	359	422	482	544	605	669	733	856	982

WEIGHTS AND DIMENSIONS OF CAST-IRON PIPES, FOR GAS, WATER, AND STEAM.

Dimensions.			Spigot and Faucet Joints.			Turned and Bored Joints.			Flanged Joints.					
Inside Diameter.	(A) Thickness of Metal.	Length exclusive of Faucet.	Average Weight per Pipe.			Average Weight per Pipe.			Inside Diameter.	(B) Thickness of Metal.	Length over Flanges.	Average Weight per Pipe.		
Ins.	Ins.	Ft.	Cwts.	Qrs.	Lbs.	Cwts.	Qrs.	Lbs.	Ins.	Ins.	Ft.	Cwts.	Qrs.	Lbs.
3/4	7/32	4½			14½				2	5/16	6		1	21
1	1/4	6			22		1	0	2½	5/16	9		3	2
1¼	1/4	6			26		1	5	2½	1/2	9	1	1	7
1½	1/4	6		1	2		1	7	3	5/16	9	1	0	0
1½	9/32	6		1	7		1	13	3	3/8	9	1	0	13
2	1/4	6		1	14		1	20	3	1/2	9	1	2	0
2	5/16	6		1	22		2	0	3	5/8	9	2	0	0
2	11/32	6		2	0		2	8	3½	3/8	9	1	1	3
2½	5/16	6		2	0		2	7	3½	1/2	9	1	3	0
2½	5/16	9		3	0		3	10	4	3/8	9	1	1	21
2½	11/32	9		3	14		3	21	4	7/16	9	1	2	24
3	5/16	9		3	14		3	21	4	1/2	9	2	0	0
3	11/32	9	1	0	0	1	0	7	4	5/8	9	2	2	4
3	3/8	9	1	0	10	1	0	17	4½	1/2	9	2	0	21
3½	5/16	9	1	0	14	1	0	21	4½	9/16	9	2	2	0
4	11/32	9	1	1	14	1	1	21	5	1/2	9	2	1	21
4	3/8	9	1	2	0	1	2	7	5	5/8	9	3	0	14
4½	11/32	9	1	2	14	1	2	21	5½	1/2	9	2	2	14
5	3/8	9	1	3	14	1	3	21	6	7/16	9	2	2	0
5	13/32	9	2	0	0	2	0	7	6	1/2	9	2	3	14
5½	3/8	9	2	0	11	2	0	18	6	5/8	9	3	2	14
6	13/32	9	2	1	14	2	1	14	7	1/2	9	3	1	7
6	7/16	9	2	2	0	2	2	9						
7	7/16	9	2	3	14	2	3	24						
7	1/2	9	3	1	4	3	1	14						

(*Macfarlane, Strang & Co., Ltd.*)

STANDARD THICKNESSES AND WEIGHTS OF SCREWED AND SOCKETED TUBES TO BRITISH STANDARD SPECIFICATION Nos. 788 and 789

Nom. Bore	Approx. Outside Diameter	Thickness S.W.G.			Weight per foot in lb.					
					Steel to B.S.S. 789			Wrought Iron to B.S.S. 788		
		Gas	Water	Steam	Gas	Water	Steam	Gas	Water	Steam
$\frac{1}{8}''$	$\frac{13}{32}''$	14	13	12	·281	·311	·338	·276	·305	·331
$\frac{1}{4}''$	$\frac{17}{32}''$	14	13	12	·389	·435	·477	·382	·426	·468
$\frac{3}{8}''$	$\frac{11}{16}''$	13	12	11	·590	·653	·713	·579	·641	·700
$\frac{1}{2}''$	$\frac{27}{32}''$	12	11	10	·828	·908	·984	·812	·891	·966
$\frac{3}{4}''$	$1\frac{1}{16}''$	11	10	9	1·182	1·287	1·421	1·159	1·262	1·393
1"	$1\frac{11}{32}''$	10	9	8	1·678	1·860	2·037	1·646	1·825	1·997
$1\frac{1}{4}''$	$1\frac{11}{16}''$	9	8	7	2·398	2·633	2·863	2·350	2·581	2·807
$1\frac{1}{2}''$	$1\frac{29}{32}''$	8	7	6	3·012	3·281	3·541	2·954	3·215	3·472
2"	$2\frac{3}{8}''$	8	7	6	3·828	4·175	4·514	3·753	4·093	4·427
$2\frac{1}{2}''$	3"	7	6	5	5·372	5·819	6·369	5·267	5·705	6·245
3"	$3\frac{1}{2}''$	7	6	5	6·344	6·875	7·532	6·220	6·741	7·384
$3\frac{1}{2}''$	4"	7	6	5	7·309	7·923	8·685	7·165	7·768	8·515
4"	$4\frac{1}{2}''$	7	6	5	8·297	8·996	9·863	8·135	8·820	9·670
5"	$5\frac{1}{2}''$	7	6	5	10·256	11·123	12·201	10·056	10·906	11·962
6"	$6\frac{1}{2}''$	7	6	5	12·254	13·290	14·579	12·014	13·030	14·294

The dimensions and weights are subject to the tolerances given in the above specifications. The weights are based on a length (measured from end of tube to end of socket) of 14 feet for $\frac{1}{8}''$ to $\frac{3}{8}''$ nom. bore inclusive and 20 feet for $\frac{1}{2}''$ to 6" nom. bore inclusive.

STANDARD WEIGHTS OF CAST-IRON WATER-PIPE FITTINGS.

Internal Diameter of Pipes and Fittings in Inches.

Description.	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.
Pipes	0 2 0	1 0 14	1 2 0	2 0 0	2 2 0	3 1 14	3 2 0	4 1 0	5 3 0		8 2 0		10 2 0	11 2 0	13 2 0	
Bends $\frac{1}{16}$-in.	0 1 6	0 2 0	0 3 4	1 0 0	1 1 7	1 3 7	2 2 0	3 0 4	3 2 0		4 1 9		5 3 0	6 0 0	7 0 0	
Bends $1\frac{1}{8}$ in.	0 1 7	0 2 2	0 3 0	0 3 27	1 1 7	1 3 25	2 2 7	3 0 4	3 1 14		4 1 7		5 2 0	6 1 0	6 3 0	
Bends $\frac{1}{4}$ in.	0 1 7	0 2 7	0 3 7	1 0 16	1 2 0	1 3 14	2 2 7	3 0 21	3 1 21		4 1 0		5 3 0	6 2 0	7 1 0	
Cross-pipe	0 2 21	0 3 18	1 0 14	1 1 7	1 3 21	2 1 21	2 3 0	4 0 0	4 3 0		6 2 0		6 2 0	10 0 0	10 2 0	
S bend	0 1 21	0 2 0	0 3 21	1 0 4	1 3 0	2 0 0	2 3 0	3 1 0	3 3 0		5 2 0					
Bevel arm-piece or Y-piece spigots	0 2 0	0 3 0	1 0 7	1 1 14	1 2 7	2 0 9	2 3 0	3 1 0	4 1 0		6 2 0	Made to Drawings.				
Bevel arm-piece or Y-piece sockets	0 2 7	0 3 14	1 0 21	1 2 0	1 3 0	2 1 7	2 3 0	3 1 0	4 3 0		6 2 0					
Arm-pipe or branch	0 3 0	0 3 21	1 1 0	1 1 21	1 3 7	2 1 14	3 0 0	3 2 0	4 3 14		5 2 0		8 0 0	9 1 0	9 2 14	
Tee-pipe	0 1 7	0 3 0	1 0 0	1 1 13	1 2 4	2 1 0	3 1 0	3 2 14	4 0 0		5 1 7		7 2 14	9 0 0	9 1 0	
Wash-out pipe	0 2 14	0 2 21	0 3 21	1 0 16	1 1 14	1 2 14	1 3 20	2 2 0	3 0 0		1 2 0		1 3 0	2 0 0	2 3 0	
Collar or thimble	0 0 20	0 0 24	0 1 4	0 1 14	0 2 0	0 2 21	0 3 14	1 0 0	1 1 10		1 2 2		1 3 14	1 3 25	2 0 18	
Cap	0 0 12	0 0 18	0 0 24	0 1 2	0 1 12	0 2 0	0 2 14	0 3 0	0 3 9		1 0 4		1 1 21	1 2 7	1 3 0	
Reducers	3″–2″	4″–3″	5″–4″	6″–5″	7″–6″	8″–7″	9″–8″	10″–9″	12″–10″		14″–12″	Made to Drawings.				
	0 1 0	0 2 0	0 2 7	0 3 14	1 0 0	1 1 10	1 3 21	2 0 0	2 3 21		3 3 18					

Description.	18	19	20	21	22	23	24	26	28	30	32	34	36	40	42	48
	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.
Pipes	14 2 16		20 0 0		22 3 14		21 0 0			35 3 0			41 0 0	48 2 0	50 2 0	64 0 0
Bends $\frac{1}{16}$ in.	7 3 7		9 1 21													
Bends $\frac{1}{8}$ in.	7 2 7		11 1 14				13 0 7			18 3 0						
Bends $\frac{1}{4}$ in.	7 3 16		10 3 21				13 1 6									
Cross-pipe	11 2 0		14 0 0				17 0 0									
S bend																
Bevel arm-piece or Y-piece spigots																
Bevel arm-piece or Y-piece sockets								Made Specially to Drawings.								
Arm-pipe or branch	10 2 0		12 0 0				15 0 0									
Tee-pipe	10 1 0		11 2 0				14 1 10									
Wash-out pipe	3 1 0		4 0 0													
Collar or thimble	3 0 25		3 2 10				5 2 10									
Cap	2 [illegible] 0		3 0 0				4 2 10									
Reducers																

Pipes 2-in. bore in 6 ft. 3 ins. lengths: 3-in. bore to 9 ins., 9 ft. 4 ins. lengths: 9-in. bore to 48 ins., 12 ft. 4 ins. lengths.

The pipes are vertically cast in dry sand, socket down, and are tested to a pressure of 500 feet head (216 lbs. per sq. in.) of water; **the fittings are tested to the same pressure.**

(The Staveley Coal and Iron Co.)

APPROXIMATE WEIGHT OF SOCKET PIPES AND FITTINGS.

Socket Pipes.

Inside Diam.	Length ex. Socket.	Depth of Socket.	Diam. of Socket.	Weight per Length.	Inside Diam.	Length ex. Socket.	Depth of Socket.	Diam. of Socket.	Weight per Length.
ins.	ft. ins.	ins.	ins.	c. q. lbs.	ins.	ft. ins.	ins.	ins.	c. q. lbs.
1½	6 0	2¾	2¾	1 0 to 1 10	6	9 0	4½	7½	2 1 14 2 1 21 2 2 0
2	6 0	3	3¼	1 14 1 21 2 0	7	9 0	5	8¾	2 3 14 to 3 0 0
2½	6 0	3½	3¾	2 0 to 2 14	8	9 0	5	9¾	3 1 14 to 3 3 0
3	9 0	3½	4¼	3 14 3 21 1 0 0	9	9 0	5	10¾	3 3 14 to 4 2 0
3½	9 0	4	1¾	1 1 0	10	9 0	5	11¾	4 2 0 5 0 0
4	9 0	4	5¼	1 1 14 1 1 21 1 2 0	12	9 0	5	13¾	5 3 0 6 2 0
4½	9 0	4	6	1 3 2	12	12 0	5	13¾	8 3 0
5	9 0	4½	6½	1 3 14 1 3 21 2 0 0					

Connections.

Description.	1½ in.	2 in.	2½ in.	3 in.	3½ in.	4 in.	5 in.	6 in.	7 in.	8 in.	9 in.	10 in.	12 in.
	q. lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.	c.q.lbs.
Elbow	10	19	26	1 6	1 16	2 0	3 0	3 18	1 1 24	1 2 14	2 0 10	2 3 0	—
½ bend	18	26	1 6	1 10	1 22	2 8	1 0 0	1 1 14	2 0 20	2 3 21	3 0 14	4 1 12	4 2 0
⅛ ,, ...	18	24	25	1 7	—	2 0	3 26	1 1 8	1 3 22	2 3 20	3 1 14	3 2 0	3 3 0
1/16 ,, ...	—	20	25	1 8	—	1 27	2 25	1 1 0	—	—	3 0 0	3 1 7	4 0 14
Tee	1 7	1 18	2 5	2 24	3 6	3 19	1 0 21	1 2 10	2 2 7	3 0 18	4 0 16	4 1 0	5 2 0
Branch	1 9	1 24	2 12	3 0	—	1 0 10	1 1 14	1 2 14	3 0 0	3 1 6	3 3 24	4 2 16	5 2 21
Collar	11	16	20	26	1 2	1 16	1 20	2 10	3 0	3 17	3 10	1 0 14	1 2 0
Syphon bend . .													
,, . .													
Plug pipe . . .	—	1 10	1 20	1 25	—	2 12	2 27						
End plug pipe . .	—	—	—	2 20	—	2 26	3 12						
,, ,, New Style													
Ducksfoot bend .	—	1 16	—	2 7	—	3 16							
Syphon and cover .	3 26	1 2 0	—	2 0 0	—	2 1 0	2 2 14	3 1 21	5 0 21	5 1 14	6 0 14	7 0 0	8 1 0
,, ,, new style . . .	—	1 1 17	—	1 3 16	—	2 0 25	3 0 15	3 0 21	—	3 2 7	—	—	9 3 0
Cross	—	1 27	—	3 1[illegible]	—	1 0 16	1 2 12	2 0 4					
Double bend 4½ cen. .	—	—	—	1 16	—	2 7							
,, 6 ,, .	—	—	—	1 17	—	2 10							
,, 7 ,, .													
,, 9 ,, .	—	—	—	1 18	—	2 14							
,, 12 ,, .	—	—	—	1 23	—	3 0							
,, 13½ ,, .	—	—	—	2 0	—	3 0							
,, 15 ,, .													
,, 18 ,, .													
,, 21 ,, .													
Cap	8	8	9	15	18	23	1 1	1 8	1 24	2 10	2 22	3 12	1 1 20
Angle branch . .	—	1 24	—	3 0	—	3 24	1 1 14	1 3 14					

Reducing Pipes.

Size.	c. q. lb.	Size.	c. q. lb.	Size.	q. lb.	Size.	q. lb.	Size.	lbs.
12 in. to 10 in.	3 0 7	8 in. to 6 in.	1 3 14	5 in. to 4 in.	3 10	2 to 1½ in.	1 6	3 to 2 in., 2 in. socket	25
10 ,, 9	3 0 2	7 ,, 6	1 2 0	5 ,, 3	3 0	6 ,, 5 in., 5 in. socket	2 14		
10 ,, 8	2 2 25	7 ,, 5	1 1 21	4 ,, 3	2 18	6 ,, 4 4 ,,	1 25		
9 ,, 8	2 1 20	6 ,, 5	1 0 8	4 ,, 2	2 8	5 ,, 4 4 ,,	1 18		
9 ,, 7	2 0 8	6 ,, 4	1 0 4	3 ,, 2½	2 6	4 ,, 3 3 ,,	1 9		
8 ,, 7	2 0 0	6 ,, 3	1 0 0	3 ,, 2	1 24	4 ,, 2 2 ,,	1 8		

(Batley, Pegg & Co.)

WEIGHT OF SEAMLESS COPPER TUBES.

S.W.G.		0000	000	00	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
		Thickness of Copper.																							
Inches.		·400	·372	·348	·324	·300	·276	·252	·232	·212	·192	·176	·160	·144	·128	·116	·104	·092	·080	·072	·064	·056	·048	·040	·036
F=full. B=bare.		13/32 B	3/8 B	11/32 F	21/64 B	19/64 F	9/32 B	1/4 F	15/64 B	7/32 B	3/16 F	11/64 F	5/32 F	9/64 F	1/8 F	7/64 F	7/64 B	3/32	5/64 F	5/64 B	1/16 F	1/16 B	3/64 F	3/64 B	1/32 F
Millimètres.		10·160	9·449	8·839	8·229	7·620	7·010	6·401	5·893	5·385	4·877	4·470	4·064	3·658	3·251	2·946	2·262	2·337	2·032	1·829	1·626	1·422	1·219	1·016	0·914
Internal Diam.		Weight of a Lineal Foot in Pounds.																							
Inches.	Mm.																								
1/8	3·2	2·54	2·24	1·99	1·76	1·54	1·34	1·15	1·00	·86	·74	·64	·55	·47	·39	·34	·29	·24	·20	·17	·15	·12	·10	·08	·07
1/4	6·3	3·14	2·80	2·52	2·25	2·00	1·76	1·53	1·35	1·18	1·03	·91	·79	·69	·58	·51	·44	·38	·32	·28	·24	·21	·17	·14	·12
3/8	9·5	3·75	3·36	3·04	2·74	2·45	2·17	1·91	1·70	1·50	1·32	1·17	1·04	·90	·78	·69	·60	·52	·44	·39	·34	·29	·25	·20	·18
1/2	12·7	4·35	3·92	3·57	3·23	2·90	2·59	2·29	2·05	1·83	1·61	1·44	1·28	1·12	·97	·86	·76	·66	·56	·50	·44	·38	·32	·26	·23
5/8	15·9	4·96	4·49	4·10	3·72	3·36	3·01	2·67	2·40	2·15	1·90	1·70	1·52	1·34	1·17	1·04	·92	·80	·68	·61	·53	·46	·39	·32	·29
3/4	19·0	5·56	5·05	4·62	4·21	3·81	3·43	3·05	2·76	2·47	2·19	1·97	1·76	1·56	1·36	1·21	1·07	·94	·80	·72	·63	·55	·46	·38	·34
7/8	22·2	6·17	5·61	5·15	4·70	4·26	3·84	3·44	3·11	2·79	2·48	2·24	2·00	1·77	1·55	1·39	1·23	1·08	·92	·82	·73	·63	·54	·44	·40
1	25·4	6·77	6·17	5·67	5·19	4·72	4·26	3·82	3·46	3·11	2·77	2·50	2·24	1·99	1·75	1·57	1·39	1·21	1·04	·93	·82	·71	·61	·50	·45
1 1/8	28·6	7·38	6·74	6·20	5·68	5·17	4·68	4·20	3·81	3·43	3·06	2·77	2·49	2·21	1·94	1·74	1·55	1·35	1·17	1·04	·92	·80	·68	·56	·51
1 1/4	31·7	7·98	7·30	6·73	6·17	5·62	5·09	4·58	4·16	3·75	3·35	3·04	2·78	2·43	2·13	1·92	1·70	1·49	1·29	1·15	1·02	·88	·75	·62	·56
1 3/8	34·9	8·59	7·86	7·25	6·66	6·08	5·51	4·96	4·51	4·07	3·64	3·30	2·97	2·65	2·33	2·09	1·86	1·63	1·41	1·26	1·11	·97	·83	·68	·61
1 1/2	38·1	9·19	8·42	7·78	7·15	6·53	5·93	5·34	4·86	4·39	3·93	3·57	3·21	2·86	2·52	2·27	2·02	1·77	1·53	1·37	1·21	1·05	·90	·74	·67
1 5/8	41·3	9·80	8·99	8·31	7·64	6·99	6·35	5·72	5·21	4·71	4·22	3·83	3·45	3·08	2·71	2·44	2·17	1·91	1·65	1·48	1·31	1·14	·97	·81	·72
1 3/4	44·4	10·40	9·55	8·83	8·13	7·44	6·76	6·10	5·56	5·03	4·51	4·10	3·70	3·30	2·91	2·62	2·33	2·05	1·77	1·59	1·40	1·22	1·04	·87	·78
1 7/8	47·6	11·01	10·11	9·36	8·62	7·89	7·18	6·48	5·91	5·35	4·80	4·37	3·94	3·52	3·10	2·79	2·49	2·19	1·89	1·70	1·50	1·31	1·12	·93	·83
2	50·8	11·61	10·67	9·88	9·11	8·35	7·60	6·86	6·26	5·67	5·09	4·63	4·18	3·73	3·29	2·97	2·65	2·33	2·01	1·80	1·60	1·39	1·19	·99	·89
2 1/8	54·0	12·22	11·24	10·41	9·60	8·80	8·02	7·25	6·61	5·99	5·38	4·90	4·42	3·95	3·49	3·14	2·80	2·47	2·13	1·91	1·69	1·48	1·26	1·05	·94
2 1/4	57·1	12·82	11·80	10·94	10·09	9·25	8·48	7·63	6·97	6·31	5·67	5·16	4·66	4·17	3·68	3·32	2·96	2·61	2·25	2·02	1·79	1·56	1·33	1·11	1·00
2 3/8	60·3	13·43	12·36	11·46	10·58	9·71	8·85	8·01	7·32	6·63	5·96	5·48	4·91	4·39	3·88	3·50	3·12	2·75	2·38	2·13	1·89	1·65	1·41	1·17	1·05
2 1/2	63·5	14·03	12·92	11·99	11·07	10·16	9·27	8·39	7·67	6·95	6·25	5·70	5·15	4·61	4·07	3·67	3·28	2·88	2·50	2·24	1·98	1·73	1·48	1·23	1·10
2 5/8	66·7	14·64	13·49	12·52	11·56	10·62	9·69	8·77	8·02	7·28	6·54	5·96	5·39	4·82	4·26	3·85	3·43	3·02	2·62	2·35	2·08	1·82	1·55	1·29	1·16
2 3/4	69·8	15·24	14·05	13·04	12·05	11·07	10·10	9·15	8·37	7·60	6·83	6·23	5·63	5·04	4·46	4·02	3·59	3·16	2·74	2·46	2·18	1·90	1·62	1·35	1·21
2 7/8	73·0	15·85	14·61	13·57	12·54	11·52	10·52	9·53	8·72	7·92	7·12	6·50	5·87	5·26	4·65	4·20	3·75	3·30	2·86	2·57	2·27	1·99	1·70	1·41	1·27
3	76·2	16·45	15·17	14·09	13·08	11·98	10·94	9·91	9·07	8·24	7·41	6·76	6·12	5·48	4·84	4·37	3·90	3·44	2·98	2·68	2·37	2·07	1·77	1·47	1·32
3 1/4	82·5	17·66	16·30	15·15	14·01	12·88	11·77	10·68	9·77	8·88	7·99	7·29	6·60	5·91	5·23	4·72	4·22	3·72	3·22	2·89	2·57	2·24	1·91	1·59	1·43
3 1/2	88·9	18·87	17·42	16·20	14·99	13·79	12·61	11·44	10·47	9·52	8·58	7·83	7·08	6·35	5·62	5·07	4·53	4·00	3·46	3·11	2·76	2·41	2·06	1·71	1·54
3 3/4	95·2	20·08	18·55	17·25	15·97	14·70	13·44	12·20	11·18	10·16	9·16	8·36	7·57	6·78	6·00	5·42	4·85	4·28	3·71	3·33	2·95	2·58	2·20	1·83	1·65
4	101·6	21·29	19·67	18·30	16·95	15·61	14·28	12·96	11·88	10·80	9·74	8·89	8·05	7·22	6·39	5·78	5·16	4·55	3·95	3·55	3·15	2·75	2·35	1·95	1·76
4 1/4	107·9	22·50	20·80	19·36	17·93	16·51	15·11	13·72	12·58	11·44	10·32	9·42	8·54	7·65	6·78	6·13	5·48	4·83	4·19	3·76	3·34	2·92	2·50	2·08	1·87
4 1/2	114·3	23·71	21·93	20·41	18·91	17·42	15·95	14·49	13·28	12·08	10·90	9·96	9·02	8·09	7·17	6·48	5·79	5·11	4·48	3·98	3·53	3·09	2·64	2·20	1·97
4 3/4	120·6	24·92	23·05	21·46	19·89	18·33	16·78	15·25	13·98	12·73	11·48	10·49	9·50	8·53	7·55	6·83	6·11	5·39	4·67	4·20	3·73	3·26	2·79	2·32	2·08
5	127·0	26·13	24·18	22·51	20·87	19·23	17·62	16·01	14·68	13·37	12·06	11·03	9·99	8·96	7·94	7·18	6·42	5·67	4·92	4·42	3·92	3·42	2·93	2·44	2·19
5 1/4	133·3	27·34	25·30	23·57	21·85	20·14	18·45	16·77	15·39	14·01	12·64	11·55	10·47	9·40	8·33	7·53	6·74	5·95	5·16	4·64	4·11	3·59	3·08	2·56	
5 1/2	139·7	28·55	26·43	24·62	22·83	21·05	19·29	17·54	16·09	14·65	13·22	12·08	10·96	9·83	8·71	7·88	7·05	6·22	5·40	4·85	4·31	3·76	3·22	2·68	
5 3/4	146·0	29·76	27·55	25·67	23·81	21·96	20·12	18·30	16·79	15·29	13·80	12·62	11·44	10·27	9·10	8·23	7·36	6·50	5·64	5·07	4·50	3·93	3·37	2·80	
6	152·4	30·97	28·68	26·72	24·79	22·86	20·95	19·06	17·49	15·93	14·38	13·15	11·92	10·70	9·49	8·58	7·68	6·78	5·88	5·29	4·69	4·10	3·51	2·92	

WEIGHT OF SEAMLESS COPPER TUBES—(*continued*).

S.W.G.		0000	000	00	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
		Thickness of Copper.																							
Inches.		·400	·372	·348	·324	·300	·276	·252	·232	·212	·192	·176	·160	·144	·128	·116	·104	·092	·080	·072	·064	·056	·048	·040	·036
F=full. B=bare.		13/32B	3/8B	11/32F	21/64B	19/64F	9/32B	1/4F	15/64B	7/32B	3/16F	11/64F	5/32F	9/64F	1/8F	7/64F	7/64B	3/32	5/64F	5/64B	1/16F	1/16B	3/64F	3/64B	1/32F
Millimetres.		10·160	9·449	8·839	8·229	7·620	7·010	6·401	5·893	5·385	4·877	4·470	4·064	3·658	3·251	2·946	2·642	2·337	2·032	1·829	1·626	1·422	1·219	1·016	·914
Internal Diam.		Weight of a Lineal Foot in Pounds.																							
Inches.	Mm.																								
6¼	158·7	32·18	29·80	27·78	25·77	23·77	21·79	19·82	18·19	16·57	14·96	13·68	12·41	11·14	9·88	8·93	7·99	7·06	6·13	5·51	4·89	4·27	3·66		
6½	165·1	33·39	30·93	28·83	26·75	24·58	22·62	20·58	18·89	17·21	15·54	14·21	12·89	11·57	10·26	9·28	8·31	7·34	6·37	5·72	5·08	4·44	3·80		
6¾	171·4	34·60	32·05	29·88	27·73	25·59	23·46	21·85	19·60	17·85	16·12	14·75	13·37	12·01	10·65	9·63	8·62	7·61	6·61	5·94	5·28	4·61	3·95		
7	177·8	35·81	33·18	30·93	28·71	26·49	24·29	22·11	20·30	18·50	16·70	15·28	13·86	12·44	11·04	9·99	8·94	7·89	6·85	6·16	5·47	4·78	4·09		
7¼	184·1	37·02	34·30	31·99	29·69	27·40	25·13	22·87	21·00	19·14	17·29	15·81	14·34	12·88	11·42	10·34	9·25	8·17	7·09	6·38	5·66	4·95			
7½	190·5	38·23	35·43	33·04	30·67	28·31	25·96	23·63	21·70	19·78	17·87	16·34	14·83	13·32	11·81	10·69	9·57	8·45	7·34	6·60	5·86	5·12			
7¾	196·8	39·44	36·55	34·09	31·65	29·22	26·80	24·39	22·40	20·42	18·45	16·88	15·31	13·75	12·20	11·04	9·88	8·73	7·58	6·81	6·05	5·29			
8	203·2	40·65	37·68	35·14	32·68	30·12	27·63	25·16	23·10	21·06	19·03	17·41	15·79	14·19	12·59	11·39	10·20	9·01	7·82	7·03	6·24	5·46			
8¼	209·5	41·86	38·80	36·20	33·61	31·03	28·47	25·92	23·81	21·70	19·61	17·94	16·28	14·62	12·97	11·74	10·51	9·28	8·06	7·25	6·44				
8½	215·9	43·07	39·93	37·25	34·59	31·94	29·30	26·68	24·51	22·34	20·19	18·47	16·76	15·06	13·36	12·09	10·82	9·56	8·30	7·47	6·63				
8¾	222·2	44·28	41·06	38·30	35·57	32·84	30·14	27·44	25·21	22·98	20·77	19·00	17·25	15·49	13·75	12·44	11·14	9·84	8·55	7·68	6·82				
9	228·6	45·49	42·18	39·35	36·55	33·75	30·97	28·21	25·91	23·63	21·35	19·54	17·73	15·93	14·13	12·79	11·45	10·12	8·79	7·90	7·02				
9¼	234·9	46·70	43·30	40·41	37·53	34·66	31·81	28·97	26·61	24·27	21·93	20·07	18·21	16·36	14·52	13·14	11·77	10·40	9·03	8·12					
9½	241·3	47·91	44·43	41·46	38·51	35·57	32·64	29·73	27·31	24·91	22·51	20·60	18·70	16·80	14·91	13·49	12·08	10·68	9·27	8·34					
9¾	247·6	49·12	45·55	42·51	39·49	36·47	33·48	30·49	28·02	25·55	23·09	21·13	19·18	17·24	15·30	13·84	12·40	10·95	9·51	8·55					
10	254·0	50·33	46·68	43·56	40·47	37·38	34·31	31·25	28·72	26·19	23·67	21·67	19·67	17·67	15·68	14·20	12·71	11·23	9·76	8·77					
10¼	260·3	51·54	47·80	44·62	41·45	38·29	35·15	32·02	29·42	26·83	24·25	22·20	20·15	18·11	16·07	14·55	13·03	11·51	10·00						
10½	266·7	52·75	48·93	45·67	42·43	39·20	35·98	32·78	30·12	27·47	24·83	22·73	20·63	18·54	16·46	14·90	13·34	11·79	10·24						
10¾	273·0	53·96	50·05	46·72	43·41	40·10	36·81	33·54	30·82	28·11	25·42	23·26	21·12	18·98	16·84	15·25	13·66	12·07	10·48						
11	279·4	55·16	51·18	47·77	44·39	41·01	37·65	34·30	31·52	28·75	26·00	23·80	21·60	19·41	17·23	15·60	13·97	12·34	10·72						
11¼	285·7	56·37	52·30	48·83	45·37	41·92	38·48	35·06	32·23	29·40	26·58	24·33	22·08	19·85	17·62	15·95	14·28	12·62							
11½	292·1	57·58	53·43	49·88	46·35	42·83	39·32	35·83	32·93	30·04	27·16	24·86	22·57	20·28	18·01	16·30	14·60	12·90							
11¾	298·4	58·79	54·55	50·93	47·33	43·73	40·15	36·59	33·63	30·68	27·74	25·39	23·05	20·72	18·39	16·65	14·91	13·18							
12	304·8	60·00	55·68	51·98	48·31	44·64	40·99	37·35	34·33	31·32	28·32	25·92	23·54	21·16	18·78	17·00	15·23	13·46							
12¼	311·1	61·21	56·80	53·04	49·29	45·55	41·82	38·11	35·03	31·96	28·90	26·46	24·02	21·59	19·17	17·35	15·54								
12½	317·5	62·42	57·93	54·09	50·26	46·45	42·66	38·88	35·73	32·60	29·48	26·99	24·50	22·03	19·55	17·70	15·86								
12¾	323·8	63·63	59·05	55·14	51·24	47·36	43·49	39·64	36·44	33·24	30·06	27·52	24·99	22·46	19·94	18·05	16·17								
13	330·2	64·84	60·18	56·19	52·22	48·27	44·33	40·40	37·14	33·88	30·64	28·05	25·47	22·90	20·33	18·41	16·49								
13¼	336·5	66·05	61·30	57·25	53·20	49·18	45·16	41·16	37·84	34·53	31·22	28·59	25·96	23·33	20·72	18·76									
13½	342·9	67·26	62·43	58·30	54·18	50·08	46·00	41·92	38·54	35·17	31·80	29·12	26·44	23·77	21·10	19·11									
13¾	349·2	68·47	63·55	59·35	55·16	50·99	46·83	42·69	39·24	35·81	32·38	29·65	26·92	24·20	21·49	19·46									
14	355·6	69·68	64·68	60·40	56·14	51·90	47·67	43·45	39·94	36·45	32·96	30·18	27·41	24·64	21·88	19·81									
		3·87	**3·35**	**2·93**	**2·54**	**2·18**	**1·84**	**1·54**	**1·30**	**1·09**	**·89**	**·75**	**·62**	**·50**	**·40**	**·33**	**·26**	**·20**	**·15**	**·12**	**·10**	**·08**	**·06**	**·04**	**·03**

If the external diameter is given, subtract figure **in** *black type at bottom of column; for example, the weight per lineal foot of a copper tube 2 inches external diameter, 12 S.W.G., is 2·65 — 0·26 = 2·39 lbs.* The weights are calculated from 12 samples of tubes made from tough copper (specific gravity at 60° F. 8·8917, or ·3299 lb. per cubic inch). **Brazed copper tubes weigh more than seamless; but an exact multiple cannot be given. Mandril-drawn brazed tubes weigh the same as seamless tubes. In practice a slight deviation from the above theoretical weights must be expected. To ascertain the weight of a seamless tube of other metal, multiply the weight of a similar copper tube by ·96 for brass (70 and 30 alloy), by ·86 for wrought iron, by ·81 for cast iron, or by 1·28 for lead.** *(The Broughton Copper Co., Ltd.)*

WEIGHT OF SEAMLESS BRASS TUBES.

(Containing 70 % of copper.)

S.W.G.		Thickness of Brass															
		5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Inches		·212	·192	·176	·160	·144	·128	·116	·104	·092	·080	·072	·064	·056	·048	·040	·036
Millimètres		5·385	4·877	4·470	4·064	3·658	3·251	2·946	2·642	2·337	2·032	1·829	1·626	1·422	1·219	1·016	·914
External Diameter.		Weight of a Lineal Foot in Pounds.															
Ins.	Mm.																
3/8	9·5	..	..	..	..	..	..	..	..	..	..	..	..	·21	·18	·16	·14
7/16	11·1	..	..	..	..	..	..	..	..	..	..	..	·28	·25	·22	·18	·17
1/2	12·7	..	..	..	..	..	..	..	..	..	..	·36	·32	·29	·25	·21	·19
5/8	15·9	..	..	..	..	..	..	..	..	..	·51	·46	·42	·37	·32	·27	·25
3/4	19·0	..	..	..	..	..	..	..	..	·70	·62	·57	·51	·45	·39	·33	·30
7/8	22·2	..	..	..	..	..	..	..	·93	·84	·74	·67	·60	·53	·46	·39	·35
1	25·4	..	..	..	..	..	..	1·19	1·08	·97	·86	·78	·70	·62	·53	·45	·40
1 1/8	28·6	..	..	..	..	..	1·49	1·36	1·24	1·11	·97	·88	·79	·70	·60	·50	
1 1/4	31·7	..	..	..	..	..	1·67	1·53	1·39	1·24	1·09	·99	·88	·78	·67	·56	
1 3/8	34·9	..	..	..	..	2·06	1·86	1·70	1·54	1·37	1·21	1·09	·98	·86	·74	·62	
1 1/2	38·1	..	..	..	..	2·27	2·04	1·87	1·69	1·51	1·32	1·20	1·07	·94	·81	·68	
1 5/8	41·3	..	..	..	2·73	2·48	2·23	2·04	1·84	1·64	1·44	1·30	1·16	1·02	·88		
1 3/4	44·4	..	..	..	2·96	2·69	2·42	2·21	1·99	1·78	1·56	1·41	1·26	1·10	·95		
1 7/8	47·6	..	..	3·48	3·19	2·90	2·60	2·38	2·14	1·91	1·67	1·51	1·35	1·19	1·02		
2	50·8	..	..	3·74	3·43	3·11	2·79	2·54	2·30	2·04	1·79	1·62	1·44	1·27	1·09		
2 1/8	54·0	..	4·32	3·99	3·66	3·32	2·98	2·71	2·45	2·18	1·90	1·72	1·54	1·35			
2 1/4	57·1	..	4·60	4·25	3·89	3·53	3·16	2·88	2·60	2·31	2·02	1·83	1·63	1·43			
2 3/8	60·3	5·34	4·88	4·51	4·18	3·74	3·35	3·05	2·75	2·45	2·14	1·93	1·72	1·51			
2 1/2	63·5	5·65	5·16	4·76	4·36	3·95	3·54	3·22	2·90	2·58	2·25	2·04	1·81	1·59			
2 5/8	66·7	5·96	5·44	5·02	4·59	4·16	3·72	3·39	3·05	2·71	2·37	2·14	1·91	1·67			
2 3/4	69·8	6·27	5·72	5·28	4·83	4·37	3·91	3·56	3·20	2·85	2·49	2·24	2·00	1·76			
2 7/8	73·0	6·57	6·00	5·53	5·06	4·58	4·09	3·73	3·36	2·98	2·60	2·35	2·09	1·84			
3	76·2	6·88	6·28	5·79	5·29	4·79	4·28	3·90	3·51	3·11	2·72	2·45	2·19	1·92			
3 1/8	79·3	7·19	6·56	6·04	5·52	5·00	4·47	4·06	3·66	3·25	2·84	2·56	2·28				
3 1/4	82·5	7·50	6·84	6·30	5·76	5·21	4·65	4·23	3·81	3·38	2·95	2·66	2·37				
3 3/8	85·7	7·81	7·12	6·56	5·99	5·42	4·84	4·40	3·96	3·52	3·07	2·77	2·47				
3 1/2	88·9	8·12	7·40	6·81	6·22	5·63	5·03	4·57	4·11	3·65	3·19	2·87	2·56				
3 5/8	92·0	8·43	7·68	7·07	6·46	5·84	5·21	4·74	4·26	3·78	3·30	2·98	2·65				
3 3/4	95·2	8·73	7·95	7·32	6·69	6·05	5·40	4·91	4·42	3·92	3·42	3·08	2·75				
3 7/8	98·4	9·04	8·23	7·58	6·92	6·26	5·58	5·08	4·57	4·05	3·53	3·19	2·84				
4	101·6	9·35	8·51	7·84	7·15	6·47	5·77	5·25	4·72	4·19	3·65	3·29	2·93				
4 1/4	107·9	9·97	9·07	8·35	7·62	6·88	6·14	5·58	5·02	4·45	3·88	3·50					
4 1/2	114·3	10·59	9·63	8·86	8·09	7·30	6·52	5·92	5·32	4·72	4·12	3·71					
4 3/4	120·6	11·20	10·19	9·37	8·55	7·72	6·89	6·26	5·63	4·99	4·35						
5	127·0	11·82	10·75	9·89	9·02	8·14	7·26	6·60	5·93	5·26	4·58						
5 1/4	133·3	12·44	11·31	10·40	9·48	8·56	7·63	6·93	6·23	5·53							
5 1/2	139·7	13·05	11·87	10·91	9·95	8·98	8·01	7·27	6·53	5·79							
5 3/4	146·0	13·67	12·43	11·42	10·41	9·40	8·38	7·61	6·84								
6	152·4	14·29	12·99	11·94	10·88	9·82	8·75	7·95									
Per cent.		*103·4*	*105·2*	*102·1*	*102·9*	*102·6*	*104·4*	*103·3*	*104·6*	*103·1*	*103·6*	*100·0*	*101·5*	*103·5*	*102·0*	*104·8*	*97·3*

The figures in italics show the difference in percentage of weight of brass tubes made to the B.W.G. and the S.W.G., the latter being taken at 100.

The above weights, multiplied by ·994, give the weight of a brass tube 2 and 1 alloy.

These weights are calculated from the specific gravity of 16 samples of tubes, from various makers, each of the samples being 1 foot long × 1 3/8 inch external diameter × 14 B.W.G., which was found to be 8·558, or ·3089 lb. per cubic inch, for the alloy of 70 and 30, and 8·508 specific gravity, or ·3071 lb. per cubic inch, for the alloy of 2 and 1 at a temperature of 60° F.

To ascertain the weight of a seamless tube of copper, multiply the weight of a similar brass tube by 1·0388.

SIZES AND WEIGHTS OF ALUMINIUM TUBES.

S.W.G.N		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
Thickness	ins.	0·300	0·276	0·252	0·232	0·212	0·192	0·176	0·160	0·144	0·128	0·116	0·104	0·092	0·080	0·072	0·064	0·056	0·048	0·040	0·036	0·032	0·028	0·024
	mm.	7·62	7·00	6·40	5·89	5·38	4·87	4·46	4·06	3·66	3·25	2·94	2·64	2·34	2·03	1·83	1·62	1·42	1·22	1·02	0·914	0·812	0·711	0·610
Outside Diameter		Weight in lbs. per Linear foot.																						
ins.	mm.																							
3/16	4·76	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	0·029	0·028	0·025	0·022	0·020	0·018	0·016	0·014
1/4	6·35	—	—	—	—	—	—	—	—	—	—	—	—	—	0·050	0·047	0·044	0·040	0·036	0·031	0·028	0·026	0·023	0·020
3/8	9·52	—	—	—	—	—	—	—	—	—	—	0·110	0·104	0·096	0·087	0·080	0·073	0·066	0·058	0·049	0·045	0·040	0·036	0·031
1/2	12·7	—	—	—	—	—	—	—	—	0·189	0·175	0·164	0·152	0·138	0·124	0·113	0·103	0·091	0·080	0·068	0·061	0·055	0·049	0·042
5/8	15·9	—	—	—	—	—	0·308	0·291	0·274	0·255	0·234	0·217	0·199	0·180	0·160	0·147	0·132	0·117	0·102	0·086	0·078	0·070	0·061	0·053
3/4	19·0	—	—	—	—	0·419	0·394	0·372	0·347	0·321	0·293	0·270	0·247	0·223	0·197	0·180	0·162	0·143	0·124	0·104	0·095	0·085	0·074	0·064
7/8	22·2	—	—	0·578	0·549	0·517	0·483	0·453	0·421	0·387	0·352	0·324	0·295	0·265	0·234	0·213	0·191	0·169	0·146	0·123	0·111	0·099	0·087	0·075
1	25·4	0·773	0·735	0·694	0·656	0·615	0·571	0·533	0·494	0·453	0·411	0·377	0·343	0·308	0·271	0·246	0·220	0·195	0·168	0·141	0·128	0·114	0·100	0·086
1 1/8	28·6	0·910	0·862	0·809	0·762	0·713	0·659	0·615	0·568	0·520	0·469	0·431	0·391	0·350	0·308	0·279	0·251	0·220	0·190	0·160	0·144	0·129	0·113	0·097
1 1/4	31·8	1·05	0·989	0·926	0·870	0·810	0·748	0·696	0·642	0·586	0·529	0·484	0·438	0·392	0·345	0·312	0·279	0·246	0·213	0·178	0·161	0·143	0·126	0·108
1 3/8	34·9	1·19	1·11	1·04	0·975	0·907	0·836	0·777	0·716	0·653	0·587	0·538	0·487	0·435	0·381	0·345	0·309	0·272	0·235	0·196	0·178	0·158	0·139	0·120
1 1/2	38·1	1·32	1·24	1·16	1·08	1·01	0·924	0·858	0·789	0·719	0·646	0·591	0·534	0·476	0·418	0·378	0·338	0·298	0·257	0·215	0·194	0·173	0·152	0·130
1 5/8	41·3	1·46	1·37	1·27	1·19	1·10	1·01	0·939	0·863	0·785	0·705	0·645	0·582	0·519	0·455	0·411	0·368	0·323	0·279	0·234	0·211	0·188	—	—
1 3/4	44·5	1·60	1·50	1·39	1·30	1·20	1·10	1·02	0·936	0·851	0·764	0·699	0·630	0·560	0·491	0·444	0·397	0·349	0·301	0·252	0·227	0·202	—	—
1 7/8	47·6	1·74	1·62	1·50	1·40	1·30	1·19	1·10	1·01	0·917	0·822	0·751	0·678	0·603	0·528	0·478	0·426	0·375	0·323	—	—	—	—	—
2	50·8	1·88	1·75	1·62	1·51	1·39	1·28	1·18	1·08	0·982	0·882	0·805	0·726	0·645	0·565	0·510	0·455	0·400	0·345	—	—	—	—	—
2 1/8	53·9	2·01	1·88	1·74	1·62	1·49	1·36	1·26	1·16	1·05	0·941	0·858	0·773	0·687	0·602	0·543	0·485	0·426	0·370	—	—	—	—	—
2 1/4	57·1	2·15	2·00	1·85	1·72	1·59	1·45	1·34	1·23	1·12	1·00	0·910	0·821	0·730	0·639	0·577	0·515	0·452	0·389	—	—	—	—	—
2 3/8	60·3	2·29	2·13	1·97	1·83	1·69	1·54	1·42	1·30	1·18	1·06	0·965	0·866	0·775	0·676	0·610	0·544	0·478	0·411	—	—	—	—	—
2 1/2	63·5	2·43	2·26	2·08	1·94	1·78	1·63	1·51	1·38	1·25	1·12	1·02	0·916	0·815	0·712	0·643	0·573	0·504	0·433	—	—	—	—	—
2 5/8	66·6	2·57	2·39	2·20	2·04	1·88	1·72	1·59	1·45	1·32	1·18	1·07	0·965	0·857	0·750	0·675	0·603	0·530	0·455	—	—	—	—	—
2 3/4	69·8	2·70	2·51	2·32	2·15	1·98	1·81	1·67	1·53	1·38	1·24	1·12	1·01	0·899	0·785	0·708	0·632	0·555	0·477	—	—	—	—	—
2 7/8	73·0	2·84	2·64	2·43	2·26	2·08	1·90	1·75	1·60	1·44	1·29	1·18	1·06	0·942	0·822	0·743	0·662	0·581	0·500	—	—	—	—	—
3	76·2	2·98	2·77	2·55	2·37	2·17	1·98	1·83	1·67	1·51	1·35	1·23	1·11	0·985	0·860	0·776	0·690	—	—	—	—	—	—	—
3 1/4	82·6	3·26	3·02	2·78	2·57	2·37	2·16	1·99	1·82	1·65	1·47	1·34	1·20	1·07	0·932	0·842	0·750	—	—	—	—	—	—	—
3 1/2	88·9	3·53	3·27	3·02	2·78	2·57	2·34	2·15	1·97	1·78	1·59	1·45	1·30	1·15	1·01	0·909	0·809	—	—	—	—	—	—	—
3 3/4	95·2	3·81	3·53	3·24	3·00	2·76	2·51	2·32	2·12	1·91	1·71	1·55	1·40	1·24	1·08	—	—	—	—	—	—	—	—	—
4	102	4·08	3·78	3·48	3·23	2·96	2·69	2·48	2·26	2·04	1·82	—	—	—	—	—	—	—	—	—	—	—	—	—
4 1/4	108	4·36	4·03	3·70	3·43	3·15	2·87	2·64	2·41	2·18	1·94	—	—	—	—	—	—	—	—	—	—	—	—	—
4 1/2	114	4·63	4·28	3·93	3·64	3·34	3·04	2·80	2·56	2·31	2·06	—	—											
4 3/4	121	4·91	4·54	4·17	3·85	3·54	3·22	2·96	2·71	2·44	2·18	—	—											
5	127	5·19	4·79	4·40	4·06	3·73	3·40	3·12	2·85	2·57	2·30	—	—											
5 1/4	133	5·46	5·05	4·62	4·27	3·92	3·57	3·29	3·00	2·70	2·41	—	—											
5 1/2	140	5·75	5·30	4·86	4·49	4·12	3·74	3·45	3·15	2·84	2·53	—	—											
5 3/4	146	6·01	5·55	5·09	4·71	4·32	3·92	—	—	—	—	—	—											
6	152	6·30	5·81	5·33	—	—	—	—	—	—	—	—	—											

	Aluminium.	Brass.	Copper.	Steel.
Specific Gravity ...	2·71	8·30	8·89	7·85
Ratio of Weight ...	1·00	3·06	3·28	2·90

NOTE.—The above weights are subject to the usual manufacturing tolerances. (*The British Aluminium Co., Ltd.*)

Tinplates.

Principal Denominations and Sizes.

Mark.	Size.	Sheets per Box.	Weight per Box.	Thicknesses of Sheets.		Mark.	Size.	Sheets per Box.	Weight per Box.	Thicknesses of Sheets.	
	Ins.		Lbs.	Mm.	In.		Ins.		Lbs.	Mm.	In.
IC	14×10	225	108	0·313	0·0123	IC	28×20	56	108	0·315	0·0123
IX	,,	,,	136	0·395	0·0155	IX	,,	,,	136	0·396	0·0155
IXX	,,	,,	156	0·453	0·0179	IC	20×10	225	154	0·313	0·0123
IXXX	,,	,,	176	0·511	0·0201	IX	,,	,,	194	0·394	0·0155
IC	20×14	112	108	0·315	0·0123	IC	14×18¾	124	110	0·309	0·0122
ICL	,,	,,	100	0·292	0·0114	IC	14×19¼	120	110	0·311	0·0122
ICL	,,	,,	95	0·277	0·0109	IC	30×21	112	243	0·315	0·0124
ICL	,,	,,	90	0·262	0·0103	CL	,,	,,	224	0·290	0·0114
ICL	,,	,,	85	0·248	0·0097	CLL	,,	,,	190	0·246	0·0097
ICL	,,	,,	80	0·233	0·0091	CLLL	,,	,,	176	0·228	0·0090
IX	,,	,,	136	0·396	0·0155	CLLLL	,,	,,	160	0·207	0·0081
IXX	,,	,,	156	0·455	0·0179	DC	17×12½	100	94	0·404	0·0160
IXXX	,,	,,	176	0·513	0·0201	DX	,,	,,	122	0·525	0·0206
IXXXX	,,	,,	196	0·571	0·0223	DXX	,,	,,	143	0·615	0·0242
IC	28×20	,,	216	0·315	0·0124	DXXX	,,	,,	164	0·706	0·0278
IX	,,	,,	272	0·396	0·0156	DXXXX	,,	,,	185	0·796	0·0313

Weights in Lbs. per Foot Super and Equivalent S.W.G.

Mark.	Weight.	S.W.G.	Mark.	Weight.	S.W.G.
IC	0·514	30	DXXX	1·138	22
IX	0·643	27	DXXXX	1·281	21
IXX	0·739	26	SDC	0·733	26
IXXX	0·836	25	SDX	0·825	25
IXXXX	0·932	23	SDXX	0·916	24
DC	0·664	27	SDXXX	1·008	23
DX	0·854	24	SDXXXX	1·100	22
DXX	0·996	23			

Approximate Weight of Single Plates

Mark.	Weight per Plate.	Size of Plate, Ins.	Weight per Plate.	Size of Plate, Ins.	Weight per Plate.	Size of Plate, Ins.
	lb. oz.		lb. oz.		lb. oz.	
IC	0 8	14 × 10	1 0	14 × 20	2 0	28 × 20
IX	0 10		1 4		2 8	
IXX	0 11½		1 7		2 14	
IXXX	0 13		1 10		3 4	
IXXXX	0 14¼		1 13		3 10	
DC	0 15½	17 × 12½	1 15½	17 × 25	3 14½	34 × 25
DX	1 4½		2 8½		5 0½	
DXX	1 7¼		2 15		5 14	
DXXX	1 10¾		3 5½		6 11½	
DXXXX	1 14¼		3 12½		7 9	
SDC	0 13¾	15 × 11	1 10¾	15 × 22	—	—
SDX	0 15½		1 14¼			
SDXX	1 0¾		2 1½			
SDXXX	1 2½		2 5			
SDXXXX	1 4½		2 3½			

Weight and Dimensions of Hoop-Iron per Foot Run.

B. W. Gauge.	Width.	Weight.	B. W. Gauge.	Width.	Weight.	B. W. Gauge.	Width.	Weight.	B. W. Gauge.	Width.	Weight.
	Ins.	Lbs.		Ins.	Lb.		Ins.	Lb.		Ins.	Lb.
11	2¾	1·1577	13	2	·6332	15	1½	·3687	18	1	·1616
12	2½	·9178	13	1¾	·5540	15	1⅜	·3334	19	⅞	·1237
12	2	·7342	14	1⅝	·4715	16	1¼	·2652	20	¾	·0884
13	2¼	·7123	14	1½	·4042	17	1⅛	·2084	21	⅝	·0694

Weight of Wire per 100 Feet.

B. W. Gauge.	Diam. in Decimals of an Inch.	Iron.	Steel.	Brass.	Copper.	B. W. Gauge.	Diam. in Decimals of an Inch.	Iron.	Steel.	Brass.	Copper.
		Lbs.	Lbs.	Lbs.	Lbs.			Lbs.	Lbs.	Lbs.	Lbs.
0	·340	30·582	30·919	33·427	35·170	19	·042	·466	·471	·510	·536
1	·312	25·753	26·036	28·148	29·616	20	035	·324	·327	·344	·372
2	·284	21·338	21·573	23·322	24·539	21	·033	·288	·291	·314	·331
3	·261	18·022	18·220	19·698	20·725	22	·029	·222	·224	·243	·255
4	·239	15·112	15·278	16·517	17·378	23	·028	·207	·209	·226	·238
5	·217	12·458	12·595	13·616	14·326	24	·025	·164	·166	·179	·189
6	·208	11·446	11·572	12·510	13·173	25	·021	·116	·118	·127	·134
7	·187	9·251	9·353	10·112	10·639	26	·020	·105	·107	·115	·121
8	·166	7·290	7·370	7·968	8·384	27	·018	·085	·086	·093	·098
9	·158	6·604	6·677	7·219	7·595	28	·015	·059	·060	·065	·068
10	·137	4·965	5·020	5·427	5·710	29	·013	·044	·045	·048	·051
11	·125	4·134	4·179	4·518	4·754	30	·012	·038	·038	·041	·043
12	·109	3·143	3·178	3·435	3·615	31	·010	·026	·026	·028	·030
13	·094	2·338	2·363	2·555	2·688	32	·009	·021	·021	·023	·024
14	·080	1·693	1·712	1·851	1·947	33	·008	·016	·017	·018	·019
15	·072	1·371	1·387	1·499	1·577	34	·007	·013	·013	·014	·014
16	·063	1·050	1·062	1·148	1·208	35	·005	·006	·006	·007	·007
17	·055	·800	·809	·874	·920	36	·004	·004	·004	·004	·005
18	·048	·609	·616	·666	·701						

Table of Weight of Castings in Proportion to the Make of Patterns of Different Materials.

A Pattern weighing 1 lb.	Will Weigh when Cast in				
	Cast Iron.	Zinc.	Copper.	Yellow Brass.	Gun-metal.
	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
Mahogany	8	8	10	9·8	10
White pine	14	14·5	18	17·5	17·8
Yellow pine	13	12·6	16	15·5	16
Cedar	11·5	11·4	14·5	14	14·5
Maple	10	9·8	12·5	12	12·4

Number of Cold Punched Nuts per 100 Pounds.

Size.	Square.	Hexagon.	Size.	Square.	Hexagon.	Size.	Square.	Hexagon.
⅜ inch	2,694	2,944	¾ inch	412	496	1⅛ inch	126	156
½ „	1,200	1,440	⅞ „	258	314	1¼ „	104	121
⅝ „	758	900	1 „	171	201	1½ „	60	68

Weight of Nut and Bolt-heads.

Head and Nut.	Diameter of Bolt in Inches.												
	¼	⅜	½	⅝	¾	⅞	1	1¼	1½	1¾	2	2½	3
	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
Hexagon .	·017	·057	·128	·267	·43	·73	1·1	2·14	3·77	5·62	8·75	17·2	28·8
Square .	·021	·070	·164	·321	·553	·882	1·31	2·56	4·42	7·00	10·5	21·0	36·4

Weight of Washers per 100.

Size.	Weight.	Size.	Weight.	Size.	Weight.	Size.	Weight.
In.	Lbs.	In.	Lbs.	Ins.	Lbs.	Ins.	Lbs.
⅜	1¾	¾	6¾	1	10¾	1¼	24
½	2½	⅞	8½	1⅛	18¾	1½	30
⅝	4½						

Weight of Wrought-iron Nails per 1,000.

Name.	Lgth. Ins.	Weight per 1000. Lbs.	Ozs.	Name.	Lgth. Ins.	Weight per 1000. Lbs.	Ozs.	Name.	Lgth. Ins.	Weight per 1000. Lbs.	Ozs.
Spike . .	5	100	0	Clasp, strong .	2½	12	0	Tacks, clout .	¾	1	12
,, . .	6	200	0	,, ,, .	2¾	14	0	Tacks, cut .	1/16		14
,, . .	7	300	0	,, ,, .	3¼	20	0	,, ,, .	⅝		12
,, . .	8	450	0	,, ,, .	3½	25	0	,, ,, .	½		10
,, . .	9	600	0	,, ,, .	3¾	32	0	Trunk . .	¾	1	4
,, . .	10	750	0	,, ,, .	4	40	0	,, . .	1	1	12
Rose . .	1	3	0	Clasp, cut .	1¼	3	0	,, . .	1¼	2	12
,, . .	1¼	4	0	,, ,, .	1½	3	8	,, . .	1¾	5	0
,, . .	1½	7	0	,, ,, .	2	8	0	Brads, wrought	½		8
,, . .	2	10	0	,, ,, .	2½	12	0	,, ,, .	¾		12
,, . .	2½	16	0	Clout . .	¾	2	8	,, ,, .	1		14
,, . .	2¾	20	0	Clout, fine .	1	2	0	,, ,, .	1¼	1	12
,, . .	3¼	28	0	,, ,, .	1¼	3	0	,, ,, .	1½	2	12
,, . .	3¾	36	0	,, ,, .	1½	4	0	,, ,, .	2	4	0
,, . .	4	50	0	,, ,, .	1¾	5	0	Brads, flooring	2¼	10	0
,, . .	4½	70	0	Clout, strong .	1⅛	5	0	,, ,, .	2½	15	0
,, . .	5	90	0	,, ,, .	1½	9	0	,, ,, .	3¼	20	0
Rose, fine drawn	4	40	0	,, ,, .	1¾	16	0	Brads, cut .	½		5
,, ,, ,, .	5¼	80	0	,, ,, .	2	23	0	,, ,, .	¾		12
Rose, stamped .	1¼	4	0	,, ,, .	2¼	32	0	,, ,, .	1	1	0
,, ,, .	1½	5	0	,, ,, .	2½	40	0	,, ,, .	1¼	1	4
,, ,, .	2	7	0	,, ,, .	3	50	0	Glaziers' sprigs	½		4
,, ,, .	2½	11	0	,, ,, .	3¼	60	0	,, ,,	¾		14
,, ,, .	3	25	0	,, ,, .	3¾	80	0	Horseshoe .	2	5	8
Clasp, fine .	1	1	12	Clout, tinned .	1⅛	4	0	,, . .	2⅛	7	0
,, ,, .	1¼	2	8	,, ,, .	1¾	7	0	,, . .	2¼	8	8
,, ,, .	1½	4	0	Tacks, Flemish	¼		4	,, . .	2⅜	10	0
,, ,, .	2	6	0	,, ,, .	⅜		6	,, . .	2½	11	0
,, ,, .	2½	10	0	,, ,, .	½		8	,, . .	2¾	13	8
,, ,, .	3¼	18	0	,, ,, .	11/16		14	Cart-tyre .	3½	187	8
Clasp, strong .	1¾	7	0	Tacks, clout .	1⅛	3	8	,, ,, . .	4	218	12
,, ,, .	2¼	10	0	,, ,, .	15/16	2	12	Mop . .	5½	58	0

Weight of Wrought Copper Nails per 1,000.

Name.	Length. Ins.	Weight per 1000. Lbs.	Ozs.	Name.	Length. Ins.	Weight per 1000. Lbs.	Ozs.	Name.	Length. Ins.	Weight per 1000. Lbs.	Ozs.
Diehead .	5	80	6	Rose, strong	1	3	4	Rose, fine .	2¾	18	11
,, .	5	116	4	Rose, fine .	¾	1	12	,, ,, .	3	30	1
,, .	4	80	1	,, ,, .	1	2	5	,, ,, .	4	50	6
,, .	3	40	0	,, ,, .	1¼	4	7	Jagged .	1¾	3	4
,, .	2½	29	4	,, ,, .	1½	5	5	Brads .	2	6	5
,, .	2¼	14	12	,, ,, .	1¾	6	3	,, .	1½	3	0
Rose, strong	2	11	4	,, ,, .	2	10	2	Tacks .	¾	1	15
,, ,,	1½	9	9	,, ,, .	2¼	10	3	,, . .	½		13
,, ,,	1¼	5	3	,, ,, .	2½	14	6	,, . .	⅜		10

Weight of Screws per 1,000.

Name.	Length.	Weight per 1000.		Name.	Length.	Weight per 1000.		Name.	Length.	Weight per 1000.	
	Ins.	Lbs.	Ozs.		Ins.	Lbs.	Ozs.		Ins.	Lbs.	Ozs.
Strong .	3¾	28	0	Middling .	2¾	16	0	Fine . .	2¾	14	0
" .	3	24	0	" .	2½	16	0	" . .	2½	14	0
" .	2¾	24	0	" .	2	14	0	" . .	2¼	13	0
" .	2½	22	0	" .	1¾	14	0	" . .	2	13	0
" .	2	22	0	" .	1½	13	0	" . .	1¾	12	0
" .	1½	20	0	" .	1¼	13	0	" . .	1½	11	0
" .	1¼	18	0	" .	1	12	0	" . .	1¼	10	0
" .	1	16	0	" .	⅞	11	0	" . .	1	8	0
" .	⅞	14	0	" .	¾	10	0	" . .	⅞	8	0
" .	¾	13	0	" .	⅝	9	0	" . .	¾	7	0
" .	⅝	13	0	" .	½	9	0	" . .	⅝	6	0
Middling .	5	24	0	" .	⅜	7	0	" . .	½	6	0
" .	4	20	0	Fine . .	3½	18	0	" . .	⅜	3	0
" .	3½	20	0	, . .	3¼	16	0	" . .	¼	3	0
" .	3	16	0	" . .	3	16	0				

Composition Nails.

Gauge	Thick.	Length.	In 1 lb.	Gauge	Thick.	Length.	In 1 lb.	Gauge	Thick.	Length.	In 1 lb.
No.	In.	Ins.	Num.	No.	In.	Ins.	Num.	No.	In.	Ins.	Num.
1	·04	¾	290	7	·1	1⅛	184	12	·12	2	64
2	·05	⅞	260	8	·1	1¼	168	13	·13	2¼	59
3	·06	1	212	9	·11	1¼	110	14	·14	2½	51
4	·07	1⅛	201	10	·11	1⅝	101	15	·15	2¾	43
5	·08	1¼	199	11	·12	1¾	71	16	·16	3	35
6	·09	1	190								

Braziers' Copper Rivets.

Gauge	Diameter.	Length.	In 10 lbs.	Gauge	Diameter.	Length.	In 10 lbs.	Gauge	Diameter.	Length.	In 10 lbs.
No.	In.	In.	Num.	No.	In.	Ins.	Num.	No	In.	Ins.	Num.
1	3/16	½	2,384	7	⅜	13/16	334	12	11/16	1 9/16	53·8
2	¼	½	1,018	8	7/16	1	210	13	¾	1¾	41·6
3	¼	9/16	983	9	½	1 3/16	141	14	13/16	1 13/16	32·8
4	5/16	9/16	573	10	9/16	1 5/16	99·5	15	⅞	2 1/16	26·3
5	5/16	⅝	516	11	⅝	1 7/16	71·9	16	1	2⅜	16·7
6	⅜	⅞	357								

Iron Rivets.

Gauge	Diameter.	Length.	In 10 lbs.	Gauge	Diameter.	Length.	In 10 lbs.	Gauge	Diameter.	Length.	In 10 lbs.
No.	In.	In.	Num.	No.	In	Ins.	Num.	No.	In.	Ins.	Num.
0	3/16	½	3280	6	⅜	13/16	380	11	11/16	1 9/16	61·3
1	¼	½	1276	7	7/16	1	239	12	¾	1¾	47·3
2	¼	9/16	1130	8	½	1 3/16	160	13	13/16	1 13/16	37·3
3	5/16	9/16	654	9	9/16	1 5/16	112	14	⅞	2 1/16	30
4	5/16	⅝	589	10	⅝	1 7/16	81·7	15	1	2⅜	19
5	⅜	⅞	407								

APPROXIMATE WEIGHT OF PIPE JOINTING MATERIALS PER JOINT.

Spigot and Faucet Pipes.			Flanged Pipes.						
			India Rubber.			Bolts and Nuts.			
Inside Diam.*	Lead	Rope Yarn.	Inside Diam.*	Faced all over.	Faced on Strips.	No.	Diam.	Length.	Weight.
Ins.	Lbs.	Ozs.	Ins.	Ozs.	Ozs.		Ins.	Ins.	Lbs.
¾	½	7	2	1⅛	⅞	4	½	2⅜	1¼
1	¾	1	2½	1¾	1⅛	4	9/16	2½	1¼
1¼	1	1	2½	2	1⅛	4	9/16	2½	1¼
1⅜	1¼	1	3	2	2	4	9/16	2⅝	1¼
1½	1¼	1	3	2	2	4	9/16	2⅝	1¼
2	1⅜	1¼	3	2	2	4	9/16	2⅝	1¼
2	1⅝	1¼	3	2	2	4	9/16	2⅝	1¼
2	1¾	1¼	3½	2½	2¾	4	9/16	2⅝	1¼
2½	2	1½	3½	2½	2¾	4	9/16	2⅝	1½
2½	2	1½	4	2⅝	3¼	4	9/16	2¾	1⅝
2½	2	1½	4	2⅝	3¼	4	9/16	2¾	[illegible]
3	2¼	2	4	2⅝	3¼	4	9/16	2¾	1¾
3	2¼	2	4	2⅞	3¼	4	9/16	2¾	1¾
3	2¼	2	4½	3¼	4	4	⅝	3	2¼
3½	2¾	2½	4½	3¼	4	4	⅝	3	2¼
4	3⅜	2¾	5	3⅜	5	6	⅝	3	3¼
4	3⅜	2¾	5	3⅜	5	6	⅝	3	3¼
4½	3⅜	3½	5½	3⅞	5½	6	⅝	3	3½
5	6	4	6	4¼	6	6	⅝	3	3½
5	6	4	6	4½	[illegible]	6	⅝	3	3½
5½	7	4½	6	4¼	6	6	⅝	3	3½
6	8	5¼	7	5¼	8	6	⅝	3¼	[illegible]
6	8	5¼							
7	8½	6½							
7	8½	6½							

(*Macfarlane, Strang & Co., Ltd.*)

* For thickness of metal, see columns (A) and (B), page 145.

SECTION V

PART III

STRENGTH OF MATERIALS.

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

A. STRENGTH AND ELASTICITY OF METALS.

MODULI OF ELASTICITY IN TENSION (YOUNG'S MODULUS) AND SHEAR FOR METALS.
(Lbs. per sq. in. × 10^6.)

Metal.	Modulus in Tension (E).	Modulus in Shear (C).
Steel (0·14% Carbon)	29·6	12·0
„ (1·7% „)	30·2	11·0
„ (Nickel)	28·5	11·2
Iron (wrought)	30·3	10·9–11·8
„ (cast)	14·2–18·6	5·0–7·5
Invar	20·0	—
Aluminium	10·1	3·7
Duralumin	10·4	4·0
'Y' Alloy (Cu-Ni-Mg-Al)	10·6	—
Brass (66–34)	13·8–14·5	5·0
Bronze	11·4	4·9
Phosphor bronze	14·8	6·1
Copper (drawn)	18·0	5·6
„ (electrolytic)	17·3	6·5
Lead (pure cast)	2·3	0·7
Nickel	29·9	10·6
Silver	11·2	4·1
Tin	7·7	2·4
Zinc	10·5–14·9	5·5

TRANSVERSE STRENGTH OF MATERIALS.

If a beam is tested to destruction in bending, the maximum fibre stress f is calculated from the formula

$$= \frac{My}{I}$$

M = bending moment.
y = distance from neutral axis of beam to extreme fibre.
I = moment of inertia of cross-section.

This is not usually the same as the ultimate strength found by direct loading.

Definition.—f is known as the Modulus of Rupture, and is generally understood as applying to rectangular sections.

Strength Data on Metals at Normal Temperatures.

STRESS UNITS: TONS PER SQUARE INCH.

Material.	Static Tension.					Static Torsion.			± Half Endurance Range.			Hardness Number.		Impact Value ft.-lb.
	Prop. Limit.	Yield Point.	Ultimate.	Elong. %.	Reduc. Area %.	Prop. Limit.	Yield Point.	Modulus of Rupture.	Rotating Bar.	Direct Stress.	Torsional.	Brinell.	Scleroscope.	
PLAIN CARBON STEELS:														
Armco iron	5·2	6·8	18·7	50	80	4·4	5·8	25·3	12·6	9·9	7·0	—	—	—
0·02 % C ingot iron	7·2	8·5	18·9	48	76	5·6	6·1	—	11·6	7·6	5·6	69	18	19·3c
,, ,, ,, heat treated . .	15·6	16·2	22·3	36	76	—	—	—	14·7	—	—	109	20	53·7c
W.I. in direction of rolling . . .	9·0	10·7	20·9	35	29	—	—	—	10·3	7·1	—	105	15	17·5c
,, perpendicular thereto . . .	6·2	10·1	15·4	6	5	—	—	—	—	4·9	—	105	15	4·8c
Mild steel 0·13 % C	18·3	21·0	25·2	21	71	—	—	—	—	13·0	—	104	—	—
,, ,, 0·14 % C	14·0	14·0	26·0	39	64	7·6	—	27·3	11·1	—	—	119	19	50·3c
,, ,, 0·18 % C	17·1	18·0	27·5	41	68	—	—	—	12·5	—		—	—	—
,, ,, ,, reduced cold from 0·5 to 0·44 in. diam. .	31·1	None	32·8	14	60	—	—	—	18·3	—	—	—	—	—
Steel 0·24 % C.	16·8	17·1	27·0	39	64	11·3	—	31·8	11·4	—	6·3	—	—	23·6c
,, ,, ...	14·0	15·0	24·6	39	61	6·5	—	27·7	10·0	—	—	—	—	13·5c
,, ,, ...	21·0	21·8	35·3	29	67	16·2	—	35·3	14·1	—	—	—	—	38·3c
Steel 0·37 % C (in direction of rolling) .	15·4	15·6	32·1	29	54	9·1	10·0	—	14·7	9·4	7·1	132	18	15·8c
,, ,, ,, ,, ,, ,, .	36·0	39·0	45·8	23	65	23·0	26·9	—	25·4	14·7	14·5	209	26	53·2c
,, 0·49 % C	20·0	21·0	40·8	27	40	12·2	13·4	—	14·7	8·9	8·9	164	21	14·1c
,, ,, ...	30·2	31·1	43·3	24	58	16·4	None	—	21·4	—	11·6	197	23	22·5c
,, 0·65 % C	—	22·8	50·0	17	27	12·8	15·0	44·5	20·5	19·2	9·6	222	—	—
,, 0·93 C	26·9	30·2	51·3	23	40	18·8	None	—	25·0	16·1	12·9	227	31	3·3c
,, 1·20 % C	55·4	58·1	80·3	9	15	36·0	None	—	41·1	—	21·4	369	45	2·0c

Alloy Steels : *														
3 % nickel steel	30·8	41·7	49·5	24	64	28·5	—	—	21·5	—	—	228	—	50c
3½ % " "	56·0	58·0	68·8	17	49	40·4	—	57·1	28·4	—	20·1	—	—	20·5c
3½ % " "	31·0	31·4	50·7	22	44	18·5	—	46·4	22·8	—	10·0	—	—	17·9c
3½ % " "	51·6	51·8	58·9	19	55	34·2	—	51·6	32·1	—	20·8	—	—	25·8c
1½ % nickel-chromium steel	47·2	48·6	59·7	22	55	31·1	—	53·8	24·5	—	—	269	35	30·8c
3 % " "	46·9	48·6	59·5	21	58	29·7	—	52·6	26·8	—	—	260	36	29·8c
Air-hardening nickel-chromium steel	42·1	81·7	110·0	8	21	—	55·8	89·6	46·0	—	—	486	—	15i
" " " " "	51·6	—	106·0	17	57	36·3	None	87·5	46·5	—	—	464	—	—
Stainless iron	16·6	17·1	29·9	38	80	—	12·5	32·6	13·5	—	—	137	—	105i
" steel	30·0	39·8	49·0	22	56	—	26·1	43·8	21·5	—	—	217	—	40i
1½ % chrome-vanadium steel	50·9	53·8	65·4	18	53	36·1	—	57·1	29·8	—	—	292	38	13·4c
Non-Ferrous Metals :														
Copper, annealed	1·4	—	14·5	56	72	—	—	—	4·5	—	—	47	6	30·5c
" cold-drawn (electro.)	—	8·5*	18·1	27	67	—	—	13·8	5·6	—	—	—	—	—
Bronze, α alloy, cold rolled	6·9	20·1*	26·1	32	71	—	—	23·5	—	—	5·3	—	—	—
Phosphor bronze, rolled	17·8	20·3*	29·6	21	58	15·1	—	24·8	12·9	—	—	—	—	—
70/30 Brass, annealed	2·4	—	19·8	84	85	—	—	—	9·0	—	—	—	—	—
60/40 "	10·5	—	29·5	48	38	—	—	—	14·3	—	—	90	12	31·6c
Muntz Metal	—	15·0	33·3	30	31	—	—	—	—	12·5	—	—	—	—
Naval Brass	—	13·0	23·2	28	30	—	—	—	—	8·8	—	—	—	—
Manganese bronze, cast	5·8	—	31·2	33	41	5·0	—	27·6	7·6	—	—	93	17	21c
Aluminium bronze, cast	2·3	—	26·5	20	28	3·3	—	22·7	9·8	—	—	96	21	9·4i
" " rolled	—	13·6*	38·8	34	54	—	—	26·7	10·9	—	4·5	—	—	—
Duralumin	11·2	—	22·8	16	50	8·5	—	19·4	8·3	—	—	100	19	13·9c
"	14·1	21·4	29·9	16	21	8·6	—	15·8	9·5	—	—	118	—	18i
Y alloy, rolled	7·7	15·0	25·0	24	32	—	—	—	10·2	—	—	—	—	—
Magnesium-aluminium alloy	6·8	—	19·8	14	17	2·3	—	12·8	6·7	—	—	65	22	4·6i
Monel metal	22·1	24·4	40·1	40	69	—	—	—	14·7	—	8·9	169	21	74·3c
Copper-nickel alloy, cold rolled	—	13·4*	22·3	36	67	—	—	18·6	8·0	—	—	—	—	—

* The properties of these materials are modified by heat-treatment.
c = Charpy test. i = Izod test.

(By permission of Dr. E. H. Salmon and Longmans, Green & Co., Ltd., from 'Materials and Structures,' Vol. I, Appendix.)

Data on Transverse Strength of Metals.

Material.	Breaking Load. Applied at end of bar 12 ins. long, 1 in. sq.	Breaking Load. Applied at middle of bar 36 ins. long, 1 in. wide, 2 ins. deep.	Modulus of Rupture.
	lb.	cwt.	Tons per sq. in.
Cast iron for general castings	500–600	24·0–29·0	16·0–19·5
„ „ engine frames	700–800	33·5–38·0	22·5–25·5
„ „ „ cylinders	750–950	36·0–45·5	24·0–30·0
„ cold blast	—	—	—
„ hot blast	—	—	—
Mild steel	840–970	40·0–46·5	27·0–31·0
Wrought iron	500–850	24·0–41·0	16·0–27·0
Brass	about 250	12	8

Breaking Strength of Wire.

Sizes, Weights, Lengths, and Breaking Loads of Annealed and Bright Iron Wire (*Rylands*).

Standard Wire Gauge.	Diameter.		Sectional Area.	Weight of 100 Yards.	Weight of Mile.	Length of 1 Cwt.	Breaking Loads. Annealed.	Breaking Loads. Bright.	Standard Wire Gauge.
	In.	Mm.	Sq. In.	Lbs.	Lbs.	Yards.	Lbs.	Lbs.	
7/0	·500	12·7	·1963	193·4	3,404	58	10,470	15,700	7/0
6/0	·464	11·8	·1691	166·5	2.930	67	9,017	13.525	6/0
5/0	·432	11·0	·1466	144·4	2,541	78	7,814	11,725	5/0
4/0	·400	10·2	·1257	123·8	2,179	91	6,702	10,052	4/0
3/0	·372	9·4	·1087	107·1	1,885	105	5,796	8,694	3/0
2/0	·348	8·8	·0951	93·7	1,649	120	5,072	7,608	2/0
1/0	·324	8·2	·0824	81·2	1,429	138	4,397	6,595	1/0
1	·300	7·6	·0707	69·6	1,225	161	3,770	5,655	1
2	·276	7·0	·0598	58·9	1,037	190	3,190	4,785	2
3	·252	6·4	·0499	49·1	864	228	2,660	3,990	3
4	·232	5·9	·0423	41·6	732	269	2.251	3,381	4
5	·212	5·4	·0353	34·8	612	322	1,883	2,824	5
6	·192	4·9	·0290	28·5	502	393	1,544	2,316	6
7	·176	4·5	·0243	24·0	422	467	1,298	1,946	7
8	·160	4·1	·0201	19·8	348	566	1,072	1,608	8
9	·144	3·7	·0163	16·0	282	700	869	1,303	9
10	·128	3·3	·0129	12·7	223	882	687	1,030	10
11	·116	3·0	·0106	10·4	183	1,077	564	845	11
12	·104	2·6	·0085	8·4	148	1,333	451	680	12
13	·092	2·3	·0066	6·5	114	1,723	355	532	13
14	·080	2·0	·0050	5·0	88	2,240	268	402	14
15	·072	1·8	·0041	4·0	70	2,800	218	326	15
16	·064	1·6	·0032	3·2	56	3,500	172	257	16
17	·056	1·4	·0025	2·4	42	4,667	131	197	17
18	·048	1·2	·0018	1·8	32	6,222	97	145	18
19	·040	1·0	·0013	1·2	21	9,333	67	100	19
20	·036	0·9	·0010	1·0	18	11,200	55	82	20

Breaking Loads of Steel Wire (*Rylands'*).

S.W.G.	Annealed.	Bright.	S.W.G.	Annealed.	Bright.	S.W.G.	Annealed.	Bright.
	Lbs.	Lbs.		Lbs.	Lbs.		Lbs.	Lbs.
7/0	13,611	20,310	3	3,458	5,187	12	590	881
6/0	11,722	17,583	4	2,930	4,395	13	461	691
5/0	10,159	15,243	5	2,447	3,672	14	349	523
4/0	8,712	13,067	6	2,007	3,011	15	284	424
3/0	7,534	11,302	7	1,668	2,530	16	223	334
2/0	6,593	9,891	8	1,393	2,091	17	170	256
1/0	5,726	8,573	9	1,130	1,694	18	128	188
1	4,901	7,351	10	893	1,339	19	87	130
2	4,127	6,221	11	734	1,099	20	72	106

Breaking Loads of Hard-Drawn Copper Wire.

*(British Standard Specification No. 174—1938.)**

Weight per Statute Mile.			Diameter.			Minimum Breaking Load of Wire of Standard Weight.	Minimum Number of Twists.		Weight of each piece (or coil).	
Standard.	Minimum.	Maximum.	Standard.	Minimum.	Maximum.				Minimum.	Maximum.
lbs.	lbs.	lbs.	inch.	inch.	inch.	lbs.			lbs.	lbs.
800	784	816	0·2237	0·2215	0·2260	2,400	15	In 6 ins.	100	140
600	588	612	0·1938	0·1918	0·1957	1,800	20			
400	392	408	0·1582	0·1566	0·1598	1,240	25			
300	294	306	0·1370	0·1356	0·1384	945	30			
200	196	204	0·1119	0·1107	0·1130	645	20	In 3 ins.	75	140
150	147	153	0·0969	0·0959	0·0979	490	25			
100	98	102	0·0791	0·0783	0·0799	330	30			
70	68·6	71·4	0·0662	0·0655	0·0668	235	35		50	140

BRITISH STANDARD SPECIFICATION FOR STRUCTURAL STEEL FOR BRIDGES AND GENERAL BUILDING CONSTRUCTION.*

(No. 15—1948.) (*Abstract.*)

(See also No. 449—1937. Amended 1938–1946.)

No. 1 quality Steel shall be made by the Open Hearth Process (Acid or Basic) or Acid Bessemer Process unless one of these processes is specially required or specified, and shall not show on analysis more than 0·06 per cent. of sulphur or of phosphorus.

No. 2 quality (Copper Bearing) Steel shall conform to the above specification for No. 1 quality Steel but shall in addition contain copper within the following limiting values, as may be specified :

(*a*) 0·20 to 0·35 per cent., or (*b*) over 0·35 to 0·50 per cent.

The quality of the steel required shall be stated in the enquiry or order.

Plates, sections, and flat bars must show a tensile breaking strength of 28 to 33 tons per sq. in., with an elongation of not less than 20 per cent. for steel of ·375 in. thickness and upwards, and not less than 16 per cent. for steel below ·375 in. thickness

Round and square bars (other than rivet bars) must show a tensile breaking strength of 28 to 33 tons per sq. in.; the elongation must be not less than 20 or 24 per cent. (according to the form standard test piece used for making the test).

Rivet Bars, 25 to 30 tons per sq. in. breaking strength, and not less than 26 or 30 per cent. elongation (according to the test piece used).

The minimum yield stress is also specified for each shape.

BRITISH STANDARD SPECIFICATION FOR STRUCTURAL STEEL FOR MARINE BOILERS.*

(No. 14—1942.) (*Abstract.*)

***Plates.*—The tensile breaking strength of steel plates for shells, butt straps, and girders shall be between the limits of 26 and 35 tons per square inch. For plates intended for flanging or welding, and for combustion chambers and furnaces, the tensile breaking strength shall be between the limits of 26 and 30 tons per square inch. The elongation, measured on a Standard test piece having a gauge length of 8 ins., shall be not less than 20 per cent. for material of 0·375 in. in thickness and upwards required to have a tensile breaking strength of 28 to 35 tons per sq. in.; and not less than 23 per cent. for material of 0·375 in. in thickness and upwards required to have a tensile breaking strength of 26 to 30 tons per sq. in.**

***Angle and Tee Bars and Stays.*—The tensile breaking strength of angle and tee bars shall be between the limits of 28 and 32 tons per sq. in., with an elongation of not less than 20 per cent. measured on the Standard test piece. For bars for longitudinal stays the tensile breaking**

* By permission of the British Standards Institution.

strength shall be between 28 and 35 tons per sq. in., with an elongation of not less than 20 per cent. measured on the Standard test piece.

For material under 0·375 in. in thickness the elongation may be not more than 3 per cent. below the above-named elongations.

***Rivet Bars.*—The tensile breaking strength of rivet bars shall be between the limits of 26 and 30 tons per sq. in. of section, with an elongation of not less than 25 or 30 per cent. (according to the test piece used). The bars may be tested the full size as rolled.**

BRITISH STANDARD SPECIFICATION FOR STRUCTURAL STEEL FOR SHIPBUILDING.*

(No. 13—1942.) (*Abstract.*)

***Plates.*—The tensile breaking strength of steel plates, determined from Standard test pieces, shall be between the limits of 28 and 33 tons per sq. in. or 26 and 32 tons per sq. in. For plates specially intended for cold flanging the tensile breaking strength shall be between the limits of 26 and 30 tons per sq. in. The elongation, measured on a Standard test piece having a gauge length of 8 ins., shall be not less than 20 per cent. for material of ·375 in. in thickness and upwards, and not less than 16 per cent. for material below ·375 in. in thickness.**

***Angles, Bulb Angles, Channels, etc.*—The tensile breaking strength of sectional material, such as angles, bulb angles, channels, etc., shall be between the limits of 28 and 33 tons per sq. in. The elongation, measured on a Standard test piece having a gauge length of 8 ins., shall be not less than 20 per cent. for material of ·375 in. in thickness and upwards, and not less than 16 per cent. for material below ·375 in. in thickness.**

***Rivet Bars.*—The tensile breaking strength of rivet bars shall be between the limits of 25 and 30 tons per sq. in. of section, with an elongation of not less than 25 or 30 per cent.** (according to the test piece used).

BRITISH STANDARD TENSILE AND IMPACT TESTS.

For details, see British Standard Specifications No. 18—1938 (Tensile Tests), No. 131—1933 (Notched Bar Tests) and No. 1349—1947 (Impact Test for Cast Iron.)

ADMIRALTY TESTS FOR STEEL AND IRON.

(1925.)

TESTS FOR STEEL.

TESTS FOR STEEL ANGLES, ANGLE BULBS, TEES, TEE BULBS, ZED BARS, SHIP PLATES, AND SHIP SHEETS.

(*Abstract.*)

Strips cut lengthwise or crosswise to have an ultimate tensile strength of not less than 26 and not more than 30 tons per sq. in. of section, with an elongation of 20 per cent. in a length of 8 ins. For plates of 10 lbs. and sheets elongation may be not less than 18 per cent.

One bar, plate, or sheet is to be selected from every charge, provided the number of bars, plates, or sheets does not exceed 50. If above that number, one for every additional 50 or portion of 50. A sample may be selected from every section of bar, or every thickness of plate or sheet, from each charge, but bars may be received or rejected without a trial of every section, and plates or sheets without a trial of every thickness, on the invoice.

Strips cut crosswise or lengthwise 1¼ in. wide must stand bending cold in a press to a curve of which the inner radius is one and a half times the thickness of the steel tested.

The steel shall stand such forge tests (hot and cold for bars and hot for plates and sheets), and other general usage tests as may be considered necessary to prove the soundness of the material and its fitness to stand such treatment and bending as it may be subjected to in the shipyard.

The strips may all be cut in a planing machine, and have the sharp edges taken off.

The ductility of every bar or plate is to be ascertained by the application of such of the above tests as may be considered necessary.

The steel to be free from lamination and injurious surface defects.

* By permission of the British Standards Institution.

TESTS FOR STEEL RIVETS.

One tensile test shall be taken from each charge, but when the weight of the bars, as rolled from one charge exceeds 5 tons, an additional test shall be made for each further 5 tons or portion thereof.

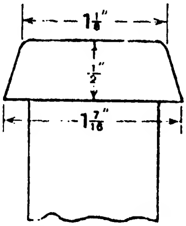


FIG. 1.—Pan Heads, Straight Necks.

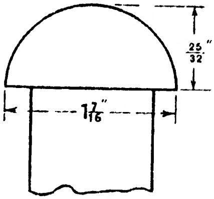


FIG. 2.—Snap Heads, Straight Necks.

SHIP RIVETS. ACID, BASIC, OR ELECTRIC FURNACE.

The rivets are to be made from British 'acid open-hearth' steel bars having an ultimate tensile strength, for 'ordinary and screw' rivets of not less than 26 and not more than 30 tons per square inch, with a minimum elongation of 25 per cent. in a length of 8 diameters of the test piece.

Rivets selected as considered necessary to stand the following tests:—

(a) Bending double when cold, and hammered till the two parts of the shank touch, in the manner shown in fig. 6, without fracture.

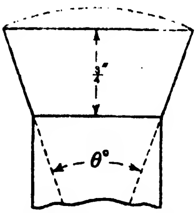


FIG. 3.—Countersunk Heads.

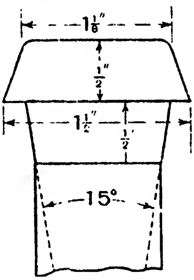


FIG. 4.—Pan Heads, Conical Necks.

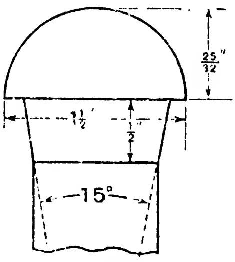


FIG. 5.—Snap Heads, Conical Necks.

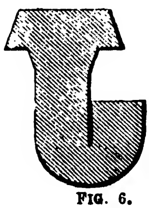


FIG. 6.

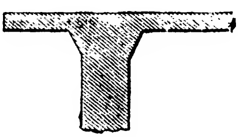


FIG. 7.

(b) Flattening of the rivet head, while hot, in the manner shown in fig. 7, without cracking at the edges. The head to be flattened until its diameter is twice the diameter of the shank.

(c) The shank of the rivet to be nicked on one side, and bent cold to show the quality of the material.

SCHEDULE OF DIMENSIONS.

Diam. of Rivet.	With Pan or Snap Heads and Straight Necks. Fig. 1 or fig. 2. Diam. of Head.	Depth of Head. Snap.	Depth of Head. Pan.	With Countersunk Heads. Fig. 3. Angle of countersink.	Depth of Head.	With Pan or Snap Heads and Conical Necks. Fig. 4 or fig. 5. Diam. of Head.	Depth of Head. Snap.	Depth of Head. Pan.	Depth of Cone.
Inch.	Inches.	Inch.	Inch.	Degrees.	Inch.	Inches.	Inch.	Inch.	Inch.
1 3/8	2	1 1/8	1 1/16	—	—	2	1 1/8	1 1/16	15/32
1 1/4	1 7/8	1	7/8	45	31/32	1 17/32	1	7/8	7/8
1 1/8	1 19/32	29/32	9/16	45	7/8	1 21/32	29/32	9/16	9/16
1	1 7/16	25/32	3/4	45	3/4	1 3/8	25/32	3/4	3/4
7/8	1 9/32	11/16	7/16	45	21/32	1 11/32	11/16	7/16	7/16
3/4	1 1/8	5/8	5/8	45	9/16	1 3/16	5/8	5/8	5/8
5/8	31/32	1/2	5/16	60	7/16	1	1/2	5/16	5/16
1/2	13/16	3/8	1/2	60	5/16	27/32	3/8	1/4	1/4
7/16	23/32	5/16	7/32	—	—	—	—	—	—
3/8	5/8	1/4	5/16	60	1/4	—	—	—	—
5/16	17/32	7/32	3/32	—	—	—	—	—	—
1/4	7/16	3/16	1/8	60	3/32	—	—	—	—
3/16	5/16	1/8		—	—	—	—	—	—

TESTS FOR STEEL BARS, FLAT, SEGMENTAL, ROUND, SQUARE, AND HEXAGONAL.

Test pieces are to have an ultimate strength of not less than 28 and not exceeding 32 tons per square inch of section, with an elongation of 20 per cent. in a length of 8 ins. The steel to stand such forge tests, both hot and cold, as may be sufficient to prove soundness of material and fitness for service.

Strips not less than 1 in. square or of the full thickness of section of the bar as rolled, must stand bending double in a press to a curve of which the inner radius is one and a half times the thickness of the steel tested. To be tested cold.

Hot and Cold Forge Test.

The steel shall stand such forge tests, both hot and cold, as may be sufficient in the opinion of the Overseer to prove the soundness of the material and fitness for the service for which it is intended.

Welding Test.

Whenever the Overseer may deem it necessary, sample pieces are to be taken for testing the welding qualities of the steel by welding two pieces together, and bending the resulting piece in way of the weld when cold. The result is to be satisfactory to the Overseer.

INSTRUCTIONS FOR TREATMENT OF MILD STEEL.

All plates or bars which can be bent *cold* are to be so treated, and if the whole length cannot be bent cold, the portion to be bent hot must be of a uniform temperature throughout; the varying temperature from hot to cold portion is to be extended so as to avoid an abrupt termination of the heat.

If plates or bars have been so severely cold worked as to raise any doubts as to their suitability for the service intended, subsequent normalising may be required at the discretion of the Overseer.

In cases where plates or bars have to be heated, the greatest care should be taken to prevent any work being done upon the material after it has fallen to the dangerous limit of temperature known as a 'blue heat'—say from 600° to 400° F. Should this limit be reached during working, the plates or bars should be reheated.

Where plates or bars have been heated throughout for bending, flanging, &c., and the work has been completed at one heat, subsequent normalising is unnecessary.

Where simple forge-work has been done, such as the formation of joggles, corners, and easy curves or bends, on portions of plates or bars, and the material has not been much distressed, subsequent normalising may be dispensed with.

Every care is to be taken, however, to allow the material to cool slowly and uniformly throughout the part which has been heated.

Plates or bars which have had a large amount of work put upon them while hot, and have had to be reheated, should be subsequently normalised. It is preferred that this normalising should be done simultaneously over the whole of each plate or bar when this can be done conveniently.

If it is inconvenient to perform the operation of normalising at one time for the whole of a plate or bar, portions may be normalised separately, proper care being taken to prevent an abrupt termination of the line of heat. If the severe working has been limited to a comparatively small part of a plate or bar, normalising may be limited to the parts which have been heated, the same care being taken to prevent an abrupt termination of the line of heat.

Normalising Mild Steel.

The material is to be evenly and gradually heated until it reaches a temperature of about 870° C. It is to soak at that temperature for about 30 minutes. It is then to be removed from the furnace and allowed to cool in air free from draughts.

In important cases where any bar or plate shows signs of failure or fracture in working, the details should be forwarded to the Admiralty, in order that instructions may be given as to the disposal of the bar or plate. Failure need not be reported in every case.

It is not generally necessary to anneal plates or bars after punching as a means of making good damage done in punching. For mild steel plating which forms an important feature in the general structural strength, such as the inner and outer bottom plating, deck plating, deck stringers, &c., the butt straps should have the holes drilled or punched $\frac{1}{8}$-in. small and rimed. In such plating the countersunk holes should be punched about $\frac{1}{8}$ in. less in diameter than the rivets which are used, the enlargement of the holes being made in the countersinking. All countersinking to be carefully done.

All steel and iron work is to be carefully scaled, scraped, and cleaned before being painted, and each portion of it as it is turned out of hand is to have a coat of linseed oil, thin red lead, or other substance, as may be required, as soon as it is sufficiently completed to receive it, in order to prevent as far as possible the work from becoming in any degree injuriously oxidised during the building of the ship. The plates of outer and inner bottom and sides, vertical keel and longitudinals, the lower plates of bulkheads, including those above the inner bottom, all the plates of oil fuel, water ballast and fresh water compartments, and the lower plates of frames, are to be treated as follows, with a view to removing the black oxide or scale:—The plates previous to their being taken in hand for working are to stand a few hours in a liquid consisting of 19 parts of water and one of hydrochloric acid. The plates should be pickled on edge and not laid flat. When plates are removed from the dilute acid bath both the surfaces are to be well brushed and washed to remove any scale which may still adhere to them. They should then be placed in another similar bath filled and kept well supplied with fresh water, or be thoroughly washed with a hose, as may be found necessary; the plates on removal from the fresh water should be placed on edge to dry.

TESTS FOR IRON.

TESTS FOR BAR IRON.

The whole of the iron supplied is to be of the description known as 'puddled' wrought iron.

The whole of the samples of every description of iron when tested with the grain are to realise a tensile strength of 22 tons to the square inch. Elongation 20 per cent. test piece 'B' for squares or rounds, and 20 per cent. test piece 'A' for flats.

Forge Test, Hot.

Bars are to be punched with a punch one-third the diameter of the bar, at distances of one and a half and three diameters from the end of the bar, the holes being at right angles to one

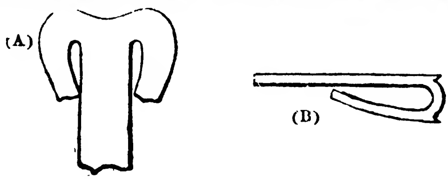


FIG. 8.

another. The holes are then to be drifted out to one and a quarter times the diameter of the bar. The sides of the holes are then to be split, and the ends must admit of turning back without fracture, as shown by (A).

Broad thick flats may be flattened on the edge to admit of being punched.

Thin flats $\frac{3}{8}$ in. thick and below are to be bent to a radius of two and a half times their thickness with the grain, without fracture.

Forge Tests, Cold.

Samples should be notched and bent, as shown by (B), to show the quality of the iron.

Bar Iron.

Ductility Test.

Test pieces as selected by the Overseer must withstand being bent cold through an angle of 90° by pressure, or by a succession of light blows, the inner radius of the bend to be not greater than three-fourths the diameter or thickness of the test piece, measured radially through the bend, without showing signs of cracking on the outside of the bent portion.

In all cases the samples are to be bent with that portion in tension that is nearest to the outside of the bar.

Welding Test.

Two pieces from each quantity of iron samples are to be scarphed and welded and subsequently bent whilst hot across the weld in order that the Overseer may be satisfied with the welding qualities of the iron.

Fire-Bars.

The bars are to be free from flaws and surface defects, and are to be subjected to such bending and other usage tests as may be considered necessary.

The standard weight of the bar iron is to be taken at 40 lb. per sq. ft. for bars of 1 in. thick, and in proportion for bars of all other thicknesses. No bar iron should exceed this prescribed weight, but a latitude of 5 per cent. below this weight will be allowed for thicknesses of ¼ an inch and upwards, and of 10 per cent. for less thicknesses. The weight of the iron ordered will be ascertained by weighing the parcel when its weight does not exceed 5 tons, and in ots of 5 tons when its weight exceeds 5 tons.

Strength of Tubes.

Let,

p = pressure in lb. per sq. in. inside tube;
D = outside diameter of tube in inches;
d = inside " " "
f = maximum hoop tensile stress in tube.

Then, up to the elastic limit,

$$p = \frac{D^2 - d^2}{D^2 + d^2} f.$$

If

t = thickness of tube, and $n = \frac{d}{t}$,

then,

$$p = \frac{2}{n + 1 + \frac{1}{n+1}} f$$

If

t is small compared with d,

then

$$p = \frac{2t}{d} f.$$

For a value of $\frac{D}{d} = 1 \cdot 1$, this 'thin cylinder' formula gives a value of p about 5 per cent. high.

Strength of Thick Hollow Cylinders.

For mild steel,

if p = pressure causing yield, in lb. per sq. in.
f = tensile stress at yield in lb. per sq. in.;
k = ratio of external to internal diameter;

then

$$p = 0 \cdot 6 f \frac{k^2 - 1}{k^2}$$

For cast iron,

if p = bursting pressure
and f = ultimate strength in tension

$$p = f \frac{k^2 - 1}{k^2 + 1}$$

(*Cook and Robertson, Engineering, Dec.* 15, 1911.

Strength of Lead Pipes.

Internal diameter in ins. . .	½	⅝	¾	1	1¼	1½	2
Thickness in ins.	·2	·2	·22	·2	·21	·24	·21
Weight per ft.	2·3	2·6	3·8	4·1	5·3	7·1	9·2
Bursting pressure in lb. per sq. in. .	1,579	1,349	1,191	911	683	734	498

(*Kirkald*)

Collapsing Strength of Steel Tubes.

Researches by Prof. R. T. Stewart (Amer. Soc. Mech.E., 1906) led to the following principal results:

1. The length of tube, between transverse joints tending to hold it to a circular form, has no practical influence upon the collapsing pressure of a commercial lap-welded steel tube so long as this length is not less than about six diameters of tube.

2. The formulæ, as based upon the present research, for the collapsing pressure of modern lap-welded Bessemer steel tubes are as follows:—

$$P = 1{,}000\left(1 - \sqrt{1 - 1{,}600\frac{t^2}{d^2}}\right) \quad . \quad . \quad (A) \qquad P = 86{,}670\frac{t}{d} - 1{,}386 \quad . \quad . \quad (B)$$

P = collapsing pressure, in pounds per square inch; d = outside diameter of tube in inches; t = thickness of wall in inches.

Formula (A) is for values of P less than 580 lbs., or for values of $\frac{t}{d}$ less than 0·023, while formula (B) is for values greater than these.

These formulæ, while strictly correct for tubes that are 20 ft. in length between transverse joints tending to hold them to a circular form, are at the same time substantially correct for all lengths greater than about six diameters. They have been tested for seven diameters, ranging from 3 ins. to 10 ins., in all obtainable commercial thicknesses of wall, and are known to be correct for this range.

3. For the most favourable practical conditions—viz. when the tube is subjected only to stress due to fluid pressure, and only the most trivial loss could result from its failure—a factor of safety of 3 would appear sufficient.

4. When only a moderate amount of loss could result from failure, use a factor of 4.

5. When considerable damage to property and loss of life might result from a failure of the tube, then use a factor of safety of 6.

6. When the conditions of service are such as to cause the tube to become less capable of resisting collapsing pressure, such as the thinning of wall due to corrosion, the weakening of the material due to overheating, the creating of internal stress in the wall of the tube due to unequal heating, vibration, etc., the above factors of safety should be increased in proportion to the severity of these actions.

Strength of Iron Chains.

(*Abstract, Safety Pamphlet No. 3; 'The use of Chains and other Lifting Gear.' Issued by the Home Office*, 1930.)*

Chains, hooks and other similar appliances for lifting purposes are generally manufactured from wrought iron or mild steel. The selection of a suitable quality of material is of the utmost importance. Material which is ductile and sufficiently elastic to recover from the strain produced by shocks should always be used. (See B.S. specification for 'Short Link Wrought Iron Crane Chain,' No. 394—1944.)

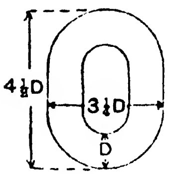


FIG. 9.—Straight-Sided Link.

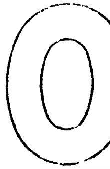


FIG. 10.—Oval Link.

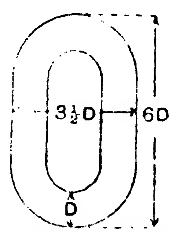


FIG. 11.—Long Link.

Ordinary short-link or close-link chains usually have links (fig. 9) with an overall length of four and a half times, and a breadth of three and a quarter times the diameter of the bar. For chains designed to stand severe shocks, links with slightly oval sides, as in fig. 10, instead of with straight sides, as in fig. 9, should be adopted; they give the chain greater elasticity.

* **His Majesty's Stationery Office, York House, Kingsway, London, W.C. 2.**

A cheaper but weaker form of link (fig. 11) is sometimes used for packing chains and for light foundry chains in cases where it is required to hitch a hook in any link. The links of such chains have an outside length of six times and an outside breadth of three and a half times the diameter of the bar.

The Ultimate Strength of Chain Iron.

The ultimate strength of chain iron varies from 21 to 25 tons per sq. in., but a reduction in strength results from welding and bending action in a link, so that the breaking stress in actual tests is about 15·5 tons per sq. in. for close-link crane chain. Based on this, the breaking strength of this class of chain is obtained by the formula $W = \frac{15 \cdot 5}{2} \pi d^2 = 24d^2$ tons (approximately).

This corresponds very closely with the standard adopted by the Admiralty, War Office, railway and dock companies, and with that specified in the Anchors and Chain Cables Act, 1899, for short-link chain cables, but most authorities agree that *short-link or close-link lifting chains of good quality should have a minimum strength of approximately $27d^2$ tons*, for chains up to 1½ in. diameter, but slightly less for chains exceeding this diameter (*see* Table A, below).

Safe Working Loads for Chains.

The tensile test loads, generally known as the Admiralty Proof Loads, are approximately equal to $12d^2$ tons for short-link cable where, d is the diameter in inches of the iron of the chain.

The safe working load generally adopted for close-link welded chains is one-half the proof load, i.e. *the safe working load* $S.L. = 6d^2$ *tons*. With chain having a breaking strength of $27d^2$ tons this allows a factor of safety * of 4·5, which may be regarded as satisfactory if the chain is not subjected to shock or any abnormal use.

Table A.

Breaking, Proof, and Maximum Working Loads for Short-Link Chains.

Size of Chain. Diameter of Iron in ins. (1)	Minimum Breaking Load. Tons. (2)	Proof Load. Tons. (3)	Maximum Working Load. Tons. (4)	Size of Chain. Diameter of Iron in ins. (1)	Minimum Breaking Load. Tons. (2)	Proof Load. Tons. (3)	Maximum Working Load. Tons. (4)
1/8	1/2	1/5	3/32	1 3/8	51	22 3/4	11 3/10
3/16	1	2/5	1/5	1 7/16	55 [illegible]	24 3/4	12 2/5
1/4	1 7/10	3/4	3/8	1 1/2	60 [illegible]	27	13 1/2
5/16	2 5/8	1 1/6	3/5	1 9/16	63 [illegible]	29 1/4	14 2/3
3/8	3 3/4	1 3/4	4/5	1 5/8	68 [illegible]	31 3/4	15 5/6
7/16	5 1/5	2 1/4	1 1/8	1 11/16	73 [illegible]	34 1/10	17
1/2	6 3/4	3	1 1/2	1 3/4	79 [illegible]	36 3/4	18 3/8
9/16	8 1/2	3 4/5	1 9/10	1 13/16	85 2/5	39 2/5	19 7/10
5/8	10 1/2	4 3/4	2 3/10	1 7/8	91 [illegible]	42 1/5	21
11/16	12 7/10	5 2/3	2 4/5	1 15/16	97 [illegible]	45	22 1/2
3/4	15 1/5	6 3/4	3 3/8	2	100	48	24
13/16	17 4/5	7 9/10	3 9/10	2 1/16	106	51	25 1/2
7/8	20 2/3	9 1/5	4 3/5	2 1/8	112 [illegible]	54	27
15/16	23 3/4	10 1/2	5 1/4	2 3/16	119 [illegible]	57 1/2	28 3/4
1	27	12	6	2 1/4	126 [illegible]	60 3/4	30 3/8
1 1/16	30 1/2	13 1/2	6 3/4	2 5/16	133 [illegible]	64	32
1 1/8	34 1/10	15 1/5	7 3/5	2 3/8	140 [illegible]	67 3/4	33 7/8
1 3/16	38	19 9/10	8 1/2	2 7/16	149 [illegible]	71 1/4	35 5/8
1 1/4	42 1/5	18 3/4	9 3/8	2 1/2	156	75	37 1/2
1 5/16	47 7/10	20 5/8	10 3/10				

* As a result of experiments on chains carried out at Illinois University, the National Safety Council of the United States of America has recommended that the safe working load for chains should not exceed $0 \cdot 4fd^2$ where f is the safe tensile stress of the material; for good wrought iron this may be taken as 7 tons per sq. in., hence the safe working load would be $2 \cdot 8d^2$ tons, which allows a factor of safety o fover 9, i.e. about twice the factor of safety generally adopted in this country.

Where there is any special risk to life or limb a maximum safe working load of $5d^2$ tons should be adopted, and where chains are subjected to shock or heavy wear, or work over pulleys, or are used for lifting vessels containing molten metal or heated liquids, the safe working load should not exceed $3 \cdot 5d^2$ tons.

The maximum safe working load for a long-link chain (fig. 11) *should not exceed two-thirds of that for a short-link chain of the same diameter.*

RINGS.

(1.)

The opening of a ring is determined, to some extent, by the size of the hook or shackle upon which it is to be placed, as this should be sufficient to allow the ring to hang freely and without jamming the end links of any attached chains.

Rings attached to sling chains are usually made with a mean diameter of four to five times the diameter of the iron forming the ring.

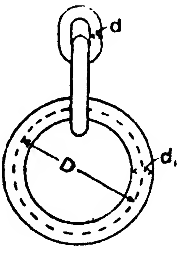


FIG. 12 —Ring

A ring must be sufficiently strong to carry safely a load equal to the sum of the safe loads of all the attached chains, and, moreover, must be capable of standing the total proof loads of these chains without deformation. When a proof load is applied to a chain, it causes a slight elongation, but this is not objectionable, because in use the load on the links is always applied in the same direction, but a ring is liable to be turned round, and hence deformation must be avoided.

If the safe working load for a ring (fig. 12) is regarded as one-half the proof load; then if D = mean diameter of ring and d_1 = diameter of the iron of the ring, the *maximum safe load can be calculated* by the formula S.L. $= 10 \dfrac{d_1^3}{D}$ tons. This should also be equal to the maximum safe load of the attached chain, i.e. $6d^2$ where d is the diameter of iron in the chain links.

EYEBOLTS.

Eyebolts should be used only for loads applied in the axial direction unless given adequate side support. The length of thread engaged with the article to be lifted should not be less than the diameter of the screwed portion. In a collared eyebolt, a neck equal in diameter to the core diameter of the screw and with a fillet under the collar should be provided. This is to prevent weakness due to cutting the thread right up to the collar.

If D is the diameter of the shank in inches, the safe load (direct) is $W = 1 \cdot 78\ D^2$ tons for wrought iron, or $W = 2 \cdot 2\ D^2$ tons for mild steel. The diameter C of the metal of the ring is usually $0 \cdot 9$ D, and the mean diameter of the ring 3 C.

SWIVELS.

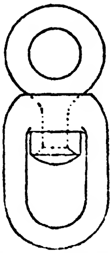


FIG. 13.—Swivel.

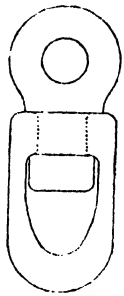


FIG. 14.—Weldless Swive..

The Board of Trade Regulations * for testing chain cables, etc., require that the various parts of a swivel shall be of the following proportions: 'The diameter of the iron in the common link being regarded as unity, the diameter of the iron in the eye of the swivel must be at least 1¼; the diameter of the iron in the pin at the bottom of the thread must be 1⅜; the diameter of the iron in the crown of the bow piece must be 1¼; the depth of the bow piece where bearing on the swivel nut must be 1¼; the diameter of the nut of the swivel pin must be 1⅜. The swivel pin should be screwed to the Whitworth standard and the nut secured thereto by welding it to the end of the pin or by fitting a tapered locking pin.'

The nut or collar on the shank of a swivel must be rigidly secured so that it cannot work loose, and if the shank is riveted at the end, an ample countersink must be allowed in the collar so that the metal cannot be sheared and the shank pulled through.

SLINGS.

The values of T for different values of W from ¼ to 20 tons, calculated for different angles between the legs, and the size of chain necessary to lift these loads safely, without the tension exceeding a safe working load of $6d^2$ tons, is given in Table B. Table C shows the actual loads calculated from the formula $W = 12d^2 \times \frac{AC}{AB}$ which can be safely lifted by slings of different sizes when used with two legs at various angles.

Thus it will be seen from Table B that a load of 5 tons can be safely lifted with a two-legged sling of $\frac{11}{16}$ in. chain if the angle between the legs does not exceed 30°, but when the angle is increased to 90° a $\frac{13}{16}$ in. chain is required, whilst with an angle of 150°, i.e. the sling nearly flat, a $1\frac{5}{16}$ in. chain is required if the tension in the chain is not to exceed the safe

* The Anchors and Chain Cables Act, 1899, with Regulations as to Testing, Scale of Maximum Charges, etc., issued by Board of Trade—to be obtained from H.M. Stationery Office, York House, Kingsway, London, W.C. 2.

oad $6d^2$ tons. Again, Table C shows that, whereas a two-legged sling made from good quality 1 in. chain can safely lift a load of 12 tons if the sling legs are vertical, the load should not exceed 10·35 tons if the legs are at an angle of 60°, nor more than 6 tons with an angle of 120°.

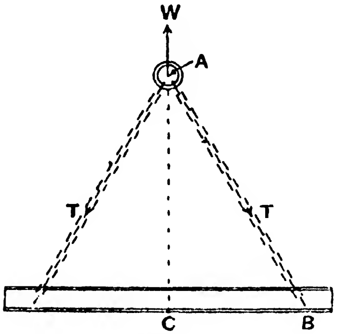


FIG. 15.

TABLE B.

SHOWING SIZES OF CHAIN REQUIRED IN TWO-LEGGED SLINGS WHEN TENSION DOES NOT EXCEED SAFE LOADS.

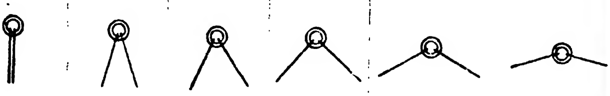
Total Load. Tons.	Diameter of Chains in Ins. for given Angles between Legs.					
	0°	30°	60°	90°	120°	150°
(1)	(2)	(3)	(4)	(5)	(6)	(7)
1/4	3/16	3/16	3/16	3/16	1/4	5/16
1/2	1/4	1/4	1/4	1/4	5/16	7/16
3/4	1/4	5/16	5/16	5/16	3/8	1/2
1	5/16	5/16	5/16	3/8	7/16	5/8
1 1/2	3/8	3/8	7/16	7/16	1/2	3/4
2	7/16	7/16	1/2	1/2	5/8	13/16
2 1/2	1/2	1/2	1/2	9/16	11/16	15/16
3	1/2	9/16	9/16	5/8	3/4	1
4	5/8	5/8	5/8	11/16	7/8	1 3/16
5	11/16	11/16	3/4	13/16	15/16	1 5/16
6	3/4	3/4	13/16	7/8	1	1 7/16
8	7/8	7/8	15/16	1	1 3/16	1 5/8
10	15/16	15/16	1	1 1/8	1 5/16	1 13/16
12	1	1 1/16	1 1/8	1 1/4	1 7/16	2
14	1 1/8	1 1/8	1 3/16	1 5/16	1 9/16	2 1/8
16	1 3/16	1 3/16	1 1/4	1 3/8	1 11/16	2 5/16
18	1 1/4	1 1/4	1 3/8	1 1/2	1 3/4	2 7/16
20	1 5/16	1 5/16	1 7/16	1 9/16	1 7/8	2 5/8

Many large users of chains issue to their employees printed rules containing instructions as to the use of multiple slings and the loads which can safely be lifted at various angles of the legs. In some cases a rule is adopted that the horizontal distance between the points of attachment of the load shall not exceed the length of sling leg; this means, in effect, that the angle between the legs must not exceed 60°.

This is a simple rule which can be readily followed by the slinger, but it makes it necessary always to provide chains which can safely lift any particular load at this opening of the sling legs. Thus, to lift a load of 12 tons, 1¼ in. chains should be used.

TABLE C.

SAFE WORKING LOADS FOR TWO-LEGGED SLINGS.



Chain Diameter. Ins.	Safe Working Loads in Tons and Cwts. for given Angles between Legs.					
	0°	30°	60°	90°	120°	150°
(1)	(2)	(3)	(4)	(5)	(6)	(7)
3/16	0–8	0–8	0–7	0–5	0–4	0–2
1/4	0–15	0–14	0–13	0–10	0–7	0–3
5/16	1–3	1–3	1–0	0–16	0–11	0–5
3/8	1–13	1–12	1–9	1–3	0–16	0–8
1/2	3–0	2–17	2–13	2–2	1–10	0–15
5/8	4–13	4–10	4–0	3–6	2–6	1–4
3/4	6–15	6–10	5–17	4–15	3–7	1–14
7/8	9–3	8–17	7–18	6–9	4–11	2–7
1	12–0	11–11	10–7	8–9	6–0	3–1
1 1/8	15–3	14–13	13–2	10–14	7–11	3–18
1 1/4	18–15	18–2	16–5	13–5	9–7	4–16
1 3/8	22–13	21–17	19–13	16–0	11–7	5–17
1 1/2	27–0	26–1	23–8	19–1	13–0	6–19
1 5/8	31–12	30–13	27–8	22–8	15–16	8–3
1 3/4	36–15	35–9	31–16	25–19	18–7	9–9
1 7/8	42–3	40–14	36–10	29–16	21–1	10–17
2	48–0	46–8	41–11	33–18	24–0	12–7
2 1/8	54–3	52–6	46–18	38–5	27–1	13–19
2 1/4	60–15	58–12	52–12	42–19	30–7	15–13
2 3/8	67–13	65–6	58–13	47–16	33–16	17–8
2 1/2	75–0	72–9	65–0	53–0	37–10	19–7

A few firms allow double slings to be used for the normal working load, provided the angle between the legs does not exceed 90°. This is an unsafe practice, as at this angle the tension in each leg is 0·7 times the total load, and unless the normal working load for the sling is taken with a very ample factor of safety the chains may be overstrained.

Shortening Slings.—Chain slings should never be shortened by tying the chain into a knot, a method frequently adopted, but one which causes excessive bending stresses in some of the links, and may result in permanent injury or fracture. If it is necessary to shorten a chain a suitable adjuster should be used, and the links bearing the load will then be subjected only to their normal stresses.

TESTING.

TABLE D.

ORDER BY THE BOARD OF TRADE.

Tests for Short-Link Chain Cables.

Order of the Board of Trade dated June 26, 1900, under s. 8 of the Anchors and Chain Cables Act, 1899.

Short-Link Chain Cables.

Breaking and Tensile Strains.

Diameter of Iron Common Links. (1)	Breaking strain which 3 links in each 15 fathoms must withstand previous to the tensile strain being applied. (2)	Tensile strain to be applied to every 15 fathoms separately. (3)	Diameter of Iron Common Links. (1)	Breaking strain which 3 links in each 15 fathoms must withstand previous to the tensile strain being applied. (2)	Tensile strain to be applied to every 15 fathoms separately. (3)
Ins.	Tons.	Tons.	Ins.	Tons.	Tons.
2	96	48	1 3/32	28 7/10	14 3/10
1 31/32	93	46 1/2	1 1/16	27	13 1/2
1 15/16	90	45	1 1/32	25 1/2	12 3/4
1 29/32	87 2/10	43 6/10	1	24	12
1 7/8	84 3/8	42 1/8	31/32	22 1/2	11 1/4
1 27/32	81 1/2	40 3/4	15/16	21	10 1/2
1 13/16	78 8/10	39 4/10	29/32	19 7/10	9 8/10
1 25/32	76	38	7/8	18 3/8	9 1/8
1 3/4	73 1/2	36 3/4	27/32	17	8 1/2
1 23/32	70 9/10	35 4/10	13/16	15 8/10	7 9/10
1 11/16	68 1/4	34 1/8	25/32	14 6/10	7 3/10
1 21/32	65 8/10	32 9/10	3/4	13 1/2	6 3/4
1 5/8	63 3/8	31 5/8	23/32	12 3/8	6 1/8
1 19/32	60 9/10	30 4/10	11/16	11 1/4	5 5/8
1 9/16	58 1/2	29 1/4	21/32	10 1/4	5 1/8
1 17/32	56 1/4	28 1/8	5/8	9 3/8	4 5/8
1 1/2	54	27	19/32	8 4/10	4 2/10
1 15/32	51 3/4	25 7/8	9/16	7 1/2	3 3/4
1 7/16	49 1/2	24 3/4	17/32	6 3/4	3 3/8
1 13/32	47 4/10	23 7/10	1/2	6	3
1 3/8	45 3/8	22 5/8	15/32	5 1/4	2 5/8
1 11/32	43 1/4	21 5/8	7/16	4 1/2	2 1/4
1 5/16	41 1/4	20 5/8	13/32	3 9/10	1 9/10
1 9/32	39 3/8	19 5/8	3/8	3 3/8	1 5/8
1 1/4	37 1/2	18 3/4	11/32	2 8/10	1 4/10
1 7/32	35 6/10	17 8/10	5/16	2 1/4	1 1/8
1 3/16	33 8/10	16 9/10	9/32	1 8/10	9/10
1 5/32	32	16	1/4	1 1/2	3/4
1 1/8	30 3/8	15 1/8			

NOTES.

Wood or other suitable packing should be provided to prevent the links coming in contact with corners of loads of hard material.

All chains and other lifting appliances should—(*a*) If used for lifting in connection with molten metal or molten slag, be examined at least once in every month and be annealed at least once in every six months; or (*b*) if used in connection with lifting machinery or plant operated mechanically or electrically, or used for lifting in metal foundries other than that mentioned in paragraph (*a*) above, be examined at least once in every three months and be annealed at least once in every twelve months; or (*c*) if used for classes of lifts other than those mentioned in (*a*) and (*b*) above, e.g., in connection with hand cranes, be examined at least once in every six months and be annealed at least once in every two years; or (*d*) if used for holding down any crane or other lifting machinery upon a stage or platform should be examined as far as reasonably practicable, *in situ*, at least once in every three months, and be fully examined and annealed at least once in every twelve months, unless the nature of the operations preclude such examinations and annealing.

A chain or other appliance should not be used for lifting its maximum working load if the wear at any part exceeds 10 per cent. of the nominal dimension of that part, or for lifting any load if it has been subjected to a severe shock or is deformed or otherwise injured.

A chain or other appliance which has been altered or repaired should be re-tested before use.

Sling chains and other lifting appliances, when not in use, should be stored under cover and kept on suitable racks or pegs.

A record with full particulars should be kept of every chain.

ANNEALING AND NORMALISING WROUGHT IRON CHAINS.

The surface metal of chain links becomes brittle as a result of repeated local impacts. When a link section is subjected to bending strains, the hardened skin cracks. Either annealing at a dull red heat (1200° F.) or normalising at 1830° F. will effectively restore the chain to a condition of great ductility.

The best method is 'close annealing.' The chain is placed in a gas- or oil-fired muffle furnace, heated to redness out of contact with the air, and then allowed to cool slowly, either in the furnace or by covering it with dry sand or ashes after removal. Close annealing prevents oxidation and subsequent scaling, and the chain is heated uniformly. Coal or coke may also be used for a close annealing furnace, and a fireclay gas retort makes a good muffle.

Chains are sometimes softened merely by heating them on a smith's fire and allowing them to cool on the hearth for about 36 hours. This method is faulty; chains so treated cannot be considered to have been effectually annealed.

After annealing, the chain should be cleaned by brushing, and each link examined for faulty welds, cracks, etc.

CHAINS AT LOW TEMPERATURES.

Grave danger of brittle failure may be incurred at 'frost' temperatures in chains with links which have received surface injuries. (See below.)

Brittleness of Chains at Low Temperatures.

Gough (*Proc. Inst. Mech. Engineers*, 1930), as a result of investigations on new wrought iron chains at low temperatures, arrived at conclusions, of which the general trend is as follows:

(1) Best quality chain iron does not develop brittleness in absence of notches at temperatures 170° C. to −78° C.

(2) The 'notched-bar' value of best chain iron decreases slightly from 200° C. to air temperature. Notch brittleness becomes increasingly evident as temperature is further reduced.

(3) Brittleness of wrought iron at moderately low temperatures disappears when normal temperatures are again attained.

WIRE ROPE SLINGS AND LIFTING TACKLE.

For general information, see British Standard Specification, No. 1290—1946 and British Standard Handbook, No. 4 (1946).

Strength of Ropes.*

The following rules are useful for very roughly estimating the safe working loads for ropes:

Steel Ropes.

C = Circumference of rope in inches; S = Safe working load in tons.

$$S = C^2 \div 2.$$

The strength naturally depends on the quality of the steel, and the factor of safety upon the service of the rope.

See Section XXI, Part III, for haulage and mining ropes.

Hemp and Manila Ropes.

For hemp and Manila ropes the strength depends greatly on the age and condition of the rope, also on the number of strands and lay of the rope. The ultimate strength, in lb., of new Manila ropes for slings, etc., is approximately $C^2 \times 700$ to $C^2 \times 1000$. (C = circumference in inches.)

See British Standard Specification No. 451—1946. (Amended 1946–1947.)

The factor of safety for lifting ropes should be at least 6, and in adverse conditions 7 to 8.

*** See paper read by Mr. Daniel Adamson, M.I.Mech.E., before the Inst. of Mech. Engrs. 1912.**

C = Circumference of rope in inches: S = Safe working load in cwts. (very roughly).

$$S = C^2.$$

From a series of 368 tests of Manila rope in sizes from ½ in. to 4½ in. diameter carried out at the United States Bureau of Standards it was found that the average breaking strength of a three-strand regular lay rope is very closely represented by the formula

$$L = 5000\, d\, (d + 1).$$

where L is in lb., d in inches.

Testing Manila Rope.

This test has been adopted by the United States Bureau of Standards: Free the rope from oil, soak it for 20 seconds in a solution of bleaching powder acidulated with acetic acid, rinse in water and then in alcohol, and finally expose the fibres of the rope for a minute to the fumes of ammonia. Manila fibre turns russet brown, while all other rope fibres turn a cherry red.

Preservation of Manila and Wire Ropes.

Manila ropes and slings and wire cables through neglect do not last as long as they should because they are allowed to decay and rust, and apart from the waste of not preserving them they are a source of accidents. Ordinary ropes should be painted over with beeswax and resin, about equal parts of each, melted together, and applied with a paint brush. The rusting of wire cables and ropes is arrested by treating them with a mixture of tallow and oil, seven or eight pounds of melted tallow to a gallon of oil, lime being added, when the mixture is cold, until it is thick. The composition is then applied with a brush.

Hemp ropes may be preserved by being dipped when dry into a solution of 1 oz. of sulphate of copper dissolved in 2½ pints of water, and kept in this solution for four days, afterwards being dried; the rope should then be soaked in a solution of 5 ozs. of soap dissolved in 2½ pints of water, and again dried.

Sisal Ropes.

For corresponding details on sisal ropes, see British Standard Specification 208—1946 (amended 1946–17).

Strength of Soldered Joints.

It is generally recognised that the strength of soft-soldered joints is increased if the film of solder between the surfaces of the join is kept as thin as possible. Some experiments carried out by T. B. Crow and described in *The Journal of the Society of Chemical Industry*, No. 13, vol. xliii, showed that by keeping the film as thin as possible a tensile strength 2½ times as strong as the original solder is obtainable in a joint between copper pieces, although in no case was the strength up to the yield point of the copper. It was demonstrated that the tin of the solder diffuses into the copper. With solder films 0·1 to 0·15 mm. in thickness the strongest joints, in tension, were those made at 265° C., while those made at a temperature above 280° C. were noticeably weak and brittle.

Effect of High Temperature on the Strength of Metals.

For data relating to the strength of metals at high temperatures, see Section XXIII, Part I.

Stress due to Temperature.

If change of temperature occurs when change of dimensions is prevented, stress is induced in the material, of amount $f = E\, a\, t$

where E = Young's modulus, a = coefficient of linear expansion, t = change of temperature.

Temperature Change in ° Fahrenheit Corresponding to Stress of 1 Ton per Square Inch.

Metal.	° F.	Metal.	° F.
Wrought iron	12·0	Brass	17·0
Mild steel	12·0	Copper	15·5
Cast iron	21·0	Aluminium	16·5

Corrosion of Metals.*

RELATIVE CORROSION OF SOFT STEEL, WROUGHT IRON, AND NICKEL STEEL, TAKING WROUGHT IRON AS A STANDARD.

Metal.	Sea Water.	Fresh Water.	Weather.	Average.
Wrought iron	100	100	100	100
Soft steel	114	94	103	103
3 per cent. nickel steel .	83	80	67	77
26 per cent. nickel steel .	32	32	30	31

(*H. M. Howe.*)

COMPARATIVE TESTS WITH WROUGHT IRON, STEEL, AND DELTA BRONZE NO. IV., IMMERSED DURING A PERIOD OF $6\frac{1}{2}$ MONTHS IN ACID MINE WATER.

Details.	Wrought Iron.	Steel.	DELTA BRONZE No. IV.
Weight of bar when put in	1·1805	1·2125	1·2787
Weight of bar after $6\frac{1}{2}$ months . . .	0·6393	0·6614	1·2633
Loss in $6\frac{1}{2}$ months	46·3 per cent.	45·45 per cent.	1·2 per cent.

CORROSION OF IRON ROPES.

Experience has shown that wire ropes of compound construction, subjected to corrosion influences, are likely to deceive engineers as to the strength remaining in them. Where reduction of diameter or circumference of the rope has taken place, not accounted for by the evidence of wear, the part of the rope under examination should first be fully loaded and then relieved of the load. Any noticeable difference in circumference in these circumstances, and the slackening of the outside wires when the load is off, will indicate that internal corrosion has taken place. The extent of corrosion inside the strand can only be estimated by the slackness of the outside wires. The corrosion between the strands can be further examined by untwisting the rope or displaying the strands sufficiently with a marlin-spike.

Poisson's Ratio.

Definition.—The ratio of lateral to longitudinal strain under load, usually denoted by $\frac{1}{m}$ (Strains are measured relative to breadth and length respectively.) The value of $\frac{1}{m}$ for a range of engineering materials is given in the following table.

VALUE OF POISSON'S RATIO $\left(\frac{1}{m}\right)$.

Material.	$\frac{1}{m}$	Material.	$\frac{1}{m}$
Aluminium	0·34	Tin	0·33
Copper	0·33	Nickel	0·31
Brass	0·33	Silver	0·38
Bronze (phosphor) . .	0·38	Zinc	0·21
Gold]	0·42	Glass (Flint)	0·20–0·26
Iron (cast)	0·23–0·27	Stone	0·20–0·30
" (wrought) . . .	0·28	Concrete	0·08–0·18
Steel (1% C.) . . .	0·29	Indiarubber	0·46–0·50
Lead	0·44		

* See also METALLURGY, Section XXIII, Part I.

B.—STRENGTH AND ELASTICITY OF TIMBER.

The strength of timber varies greatly according to the condition and specific gravity. The following table gives approximate relations between the specific gravity and some mechanical properties in green and dry conditions. (Specific gravity = d.)

Condition.	Modulus of Rupture.	Compressive Strength across Grain.	Shear Strength parallel to Grain.	Modulus of Elasticity (E).
Green . .	$17 \cdot 6d^{\frac{5}{4}}$	$3d^{\frac{9}{4}}$	$2 \cdot 7d^{\frac{4}{3}}$	$2{,}360d$
Air-dry . .	$25 \cdot 7d^{\frac{5}{4}}$	$4 \cdot 6d^{\frac{9}{4}}$	$4d^{\frac{4}{3}}$	$2{,}800d$

Unit = 1,000 lb. per sq. in.

The table of strength data below gives typical values of approximate ranges. It should be noted that in the case of timber, increased rate of loading usually gives an increase in apparent strength.

STRENGTH DATA FOR TIMBERS.

Variety.	Tension. Elastic Limit.	Tension. Ult. Strength.	Tension. Young's Modulus (E).	Shear. Ult. Strength. Along grain.	Shear. Ult. Strength. Across grain.	Shear. Modulus (C).	Bending Modulus of Rupture.	Compression Ult. Strength.	Torsion. Relative Resistance to Shock.	Torsion. Relative Torque for a given Twist.
Ash . .	7·7	8–16	2·2	1·5	4–6	0·14	10–12	6–9	2·25	1·87
Beech .	7·5	9–18	1·3–1·5	1·7	—	—	8–12	5–8	—	—
Birch (Amer.)	8·4	—	1·9	1·6	—	0·17	13	6·6	—	—
Deal . .	—	12	1·6	—	—	0·13	10	—	—	—
Elm . .	—	6–12	1·3	0·6–0·9	3–5	0·10	12	5–8	—	—
Hickory .	8·9	—	1·9	1·8	—	—	16	7	6·90	4·1
Lignum Vitæ .	—	11	—	—	—	—	—	9·9	—	—
Mahogany	—	8–16	1·4	1·4	—	0·08	12	—	1·65	3·00
Oak . .	6·7	10–15	1·4	1·8	4·5	0·15	12	6–10	6·60	2·53
Pine, pitch .	—	8–12	1·5	0·4	4·9	—	8	4–6	3·87	2·25
,, red .	—	6–9	1·2	0·5–0·8	—	0·15	5	4–6	—	—
,, white .	5·0	7	1·2	0·8	2·5–5·0	0·15	7·5	4·5	1·00	1·00
Spruce .	5·5	9	1·4–1·8	0·5	—	0·08	7·9	5·5	1·50	0·67
Teak . .	—	8–15	2·4	—	—	0·22	14·0	—	—	—
Walnut .	6–8	15	2·0	1·5	—	0·12	11·0	6	3·95	2·63

Stresses in thousands of lbs. per sq. in.
Moduli E and C in millions of lbs. per sq. in.

Resistance of Wood to Reversals of Stress.

The limiting resistance of spruce (used for aeroplane wing spars) to alternate tension and compression was proved by experiment to be above 1,600 lbs. per sq. inch, and below 1,970 lbs. per sq. inch. For spruce of this quality the safe range of stress under reversals of bending would be approximately ± 1,800 lbs. per sq. inch, or one-quarter of the ultimate stress, which was found to be 6,800 lbs. per sq. inch. In this respect spruce appears to be at a slight disadvantage compared with mild steel, in which the safe limit of stress under similar circumstances is generally about one-third of the ultimate stress. (*The National Physical Laboratory.*)

Holding Power of Spikes and Nails in Wood.

The holding power of spikes varies roughly in proportion to the depth to which they are driven. Six-inch S.W.G. No. 1 smooth wire nails were found by Warren to require approximately 1,000 lbs. per inch of depth for withdrawal, in Australian hardwoods.

Screwed spikes exert 30–80 per cent. more resistance than smooth. The resistance is also higher with unseasoned than with seasoned timber.

Holding Power of Glue.

Wood.	Pounds per Square Inch across the Grain End to End.	Pounds per Square Inch with the Grain.
Beech	2,133	1,095
Elm	1,436	1,124
Oak	1,735	568
White wood	3,149	341
Maple	1,422	896

C.—STRENGTH OF BUILDING MATERIALS.

Mechanical Properties of Building Materials.

Crushing Strength and Modulus of Rupture in thousand lbs. per sq. in.
Young's Modulus in million lbs. per sq. in.

Material.	Ultimate Crushing Strength.	Modulus of Rupture (Bending).	Young's Modulus.
Basalt	28–50	—	—
Granite	14–28	1·6–2·9	5·6–8·1
Marble	11–21	1·4–2·7	3·8
Limestone, Argillaceous	20	—	—
,, Portland	4·3	1·1	2·6
,, Bath	1·3	0·3	—
Sandstone, Craigleith	13	0·6	—
,, miscellaneous	5–21	0·6–1·4	2·6–9·0
Slate	10–20	2·3	8·7–13·0
Concrete, age 28 days, 4 : 2 : 1	2·5	0·45	2·8
,, ,, ,, 6 : 3 : 1	1·4	—	1·8
Cement (neat)	7·0	—	—
Brick, Staffordshire Blue	5·0–10·0	1·0	—
,, London Stock	1·6–2·8	0·5	—
,, Fletton	2·6–3·9	—	—
,, Aylesford red pressed	2·2	—	—
,, Sand-lime	1·5–2·0	0·3–0·4	—
Glass (Crown)	100–160	4·5–12	10·1

Relation between Strength of Brickwork and Strength of Brick alone.

(R.I.B.A. Experiments.)

Type of Brickwork.	Crushing Strength of Brick alone.	Crushing Strength of Brickwork.
	Tons/ft.²	Tons/ft.
London Stocks in lime mortar	84·3	12
,, ,, cement ,,		17
Burham Gaults in lime ,,	182·2	21
,, ,, cement ,,		24
Leicester Reds in lime ,,	382·1	32
,, ,, cement ,,		54½
Staffordshire Blues in lime mortar	701	74
,, ,, cement ,,		77½
Flettons in lime mortar	220·8	30½
,, cement ,,		56

The above figures apply for average workmanship, except in the case of Flettons, where the workmanship was specially good. In general, good workmanship may increase the strength of brickwork by up to 50 per cent., as against average values.

Adhesive Strength of Mortars.

Nature of Substance.	Tensile Strength, in Pounds per Square Inch.	After Mixture.
		Months.
Common mortar to compact limestone	15	6
" " brick	33	6
Good hydraulic mortar	140	12
Ordinary " "	85	12
Good common mortar	50	12
Portland cement from compact limestone and clay 30 to 50 days after mixture	1,200 to 1,500	

Crushing Strength of Mortars.

The following figures supplied by the Cement Marketing Co. Ltd., show the compressive strength of 2¾-in. cubes tested at 28 days. The sand was Stonecourt Sand through $\frac{1}{16}$-in. sieve:

4 parts of Sand 1 part Portland Cement	6 parts Sand 1 part Grey 'Hydralime,' 1 part Portland Cement	8 parts Sand 2 parts Grey 'Hydralime,' 1 part Portland Cement
17% Water (by vol.)	7% Water (by vol.)	17·5% Water (by vol.)
1200 lb. per sq. in.	750 lb. per sq. in.	400 lb. per sq. in.
1200 " " "	700 " " "	450 " " "
1100 " " "	700 " " "	350 " " "

MOHS Scale of Hardness.

This is the recognised scale of mineralogical hardness and is also used in metallurgy. It is based on the ability of a material to scratch those below it on the scale.

Material.	MOHS Number.
Liquid	0
Talc	1
Gypsum	2
Calcite	3
Fluorite	4
Apatite	5
Orthoclase	6
Quartz	7
Topaz	8
Corundum	9
Diamond	10

FATIGUE IN METALS.*

(By F. W. Thorne, B.Sc., A.M.I.C.E., A.M.I.Mech.E.)

It is well known that metal parts, that may be tough and ductile and able to withstand severe test loads, commonly break in a brittle manner after smaller loads have been applied or reversed large numbers of times. This mode of fracture, by which a ductile metal breaks in a brittle manner after repeated changes of a moderate load, is known as fatigue.

Fatigue fractures may be recognised by the shell-like markings that radiate from the origin of the crack, and by the absence of ductile yield near the origin. Before actual fracture, the fatigue crack may often be observed as a hair crack which grows slowly. In laboratory tests fatigue cracks often extend so rapidly that they are not observed on the surface before fracture.

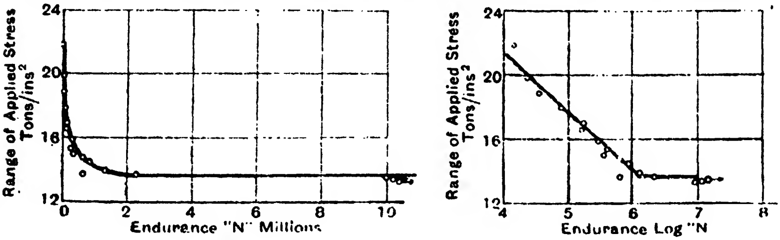
* For further information on this subject the reader is referred to *The Fatigue of Metals*, H. J. Gough, O.B.E. (Scott, Greenwood); *The Fatigue of Metals*, Moore and Kommers (McGraw-Hill Book Co.). Bibliographies in the above text-books contain respectively 183 and 448 individual references to original papers.

Fatigue fracture is liable to occur when the stress acting at any point in the part or test-piece varies, in magnitude or in direction, beyond certain ascertainable limits. The object of fatigue testing is to ascertain these limits of stress, or the corresponding limits of variation of load that can be permitted without liability to fracture.

Fatigue testing practice comprises two distinct fields of experiment with different objects: (1) the testing of different metals, in test-pieces of the simplest possible shape such that the stresses are reliably calculable; and (2) the testing of parts of complex shape, for which the intensities of stress are only nominally calculable. The second field has attained great importance in recent years, and is employed as a method of design in lieu of calculation. Fatigue-testing machines differ widely in size and design according to their applications, for testing small test-pieces or larger models of complex parts.

Methods of Testing.

To ascertain the safe conditions of loading for a given metal or a given design of part, a number of identical samples are tested under different conditions, *e.g.* with different ranges of alternating load combined with one and the same steady load. The 'endurance' is defined as the number of cycles of loading imposed before fracture terminates the test. After carrying out a number of such tests on identical pieces, the ranges of stress or load and the endurance are plotted as in fig. 16, A and B. A logarithmic base for endurance, as first adopted by Sir Thomas Stanton



A FIG. 16. B

and shown in diagram B, is usually preferred for its convenience, although both diagrams give the same data.

The form of the diagram usually reveals the 'fatigue limit,' defined as the semi-range of stress (or load in the case of a complex model) that can be applied continually without resulting in fracture. In different cases tests have to be continued to different numbers of millions of cycles to ascertain any such limit, and in some few cases no such limit has yet been established to exist even at 100 million cycles. In most cases the knee of the fatigue graph is already evident when the endurance reaches about 2 million cycles in a Haigh machine or 10 million cycles in a Wöhler rotating-beam or cantilever.*

Testing Machines.

Fatigue-testing machines may be roughly classified as follows:

(*a*) Wöhler machine for rotating-beam or cantilever test-piece. The rotating test-piece is gripped at one end in a chuck and loaded at the other through a ball bearing. The profile may be

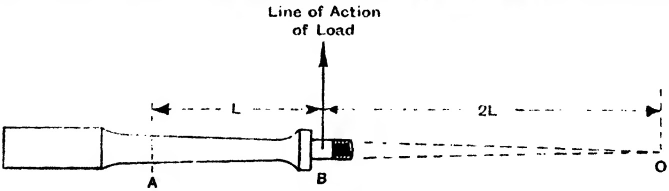


FIG. 17.

varied at will, but is preferably of the form shown in fig. 17, in which case fracture occurs at the section A, remote from sudden changes of section, so that the stress is more reliably calculable.

* R. D. Francis (Bureau of Standards, Washington, A.S.T.M., 1931).

(*b*) Reversed plane bending machines, with adjustable angle of flexure. These have been used in the U.S.A., and also at the National Physical Laboratory, for testing flat springs. The range of stress is deduced by calculation from the observed deflection.

(*c*) Haigh machine for direct push and pull, with adjustable range of load and adjustable mean or 'steady' load. The alternating load is applied and measured electro-magnetically, and the steady load is applied and measured by means of a heavy spring, which is adjusted in stiffness to compensate the force required to accelerate the moving parts.

Although the machine operates on the 'load principle' in respect that it repeats the same measured loads during each successive cycle, it operates on the 'strain principle' as regards the distribution of that load on the test-piece. As the ends of the test-piece are moved in parallel motion, all parts are equally strained when the piece is of uniform section. The machine is of large size, suitable for testing models or even full-sized parts.

(*d*) Stromeyer or other direct torsion machines, with or without bending, are made in a variety of forms for special investigations.

(*e*) Rotating strut machine for wires (Haigh & Robertson Patent No. 33113, Nov. 23, 1932). To eliminate the difficulty that wires, being of uniform section, usually break at the grips in other fatigue tests, the test-piece in this case is flexed as a strut so that maximum bending moment occurs at mid-span remote from the ends. The machine works at a very high speed of rotation, usually about 15 millions per day.

Fatigue Limit.

It has been shown conclusively that the fatigue limit bears no relation to the tensile elastic limit, and in annealed metals may be higher than the yield point.

As a result of great numbers of tests it appears that the ultimate tensile strength affords the most consistent basis of comparison for different metals. The following table gives values for the ratios between the fatigue limit and the ultimate tensile strength for stresses that reverse between equal values of push and pull.

Metal.	Fatigue limit as percentage of Ultimate tensile strength.	Metal.	Fatigue limit as percentage of Ultimate tensile strength.
Softest annealed iron, of high quality	up to 60%	Alloyed steels, 40 to 70 tons/inch² ult. quenched and tempered	35–50%
Low carbon steels, annealed	up to 60%	Alloyed steels, 70 to 100 tons/inch² ult. quenched and tempered	35–45%
Ditto. As rolled, or quenched and drawn	about 50%	Aluminium alloys of good quality	20–45%
Ditto. Poor qualities	down to 35%	Duralumin, heat-treated	25–40%
Moderate carbon steels, annealed	up to 45%	Rolled brasses, 70–30, 65–35	30–50%
Ditto. As rolled	down to 35%	Monel metal, as rolled or annealed	about 40%
Cast steel and iron	45–35%	Nickel, as rolled or annealed	about 40%
Alloyed steels, 30 to 50 tons/inch² ult. normalised	40–55%	Copper, ,, ,,	30–40%

At the frequencies usually adopted in fatigue-testing machines the fatigue limit appears to be sensibly independent of the frequency of test. Professor Jenkin found an increase in the fatigue limit of 15 per cent. when working up to 2,000 cycles per second and a further rise of 60 per cent. was recorded when working up to 20,000 cycles per second.

The form of the specimen and the method of testing have a marked influence on the fatigue limit in some metals. Different investigators have found ratios as follows in steels:

$$\frac{\text{Fatigue limit in rotating beams}}{\text{Fatigue limit in reversed plane bending}} = 1{\cdot}07 \text{ to } 1{\cdot}31$$

$$\frac{\text{Fatigue limit in reversed plane bending}}{\text{Fatigue limit in direct push and pull}} = 1{\cdot}00 \text{ to } 1{\cdot}26$$

$$\frac{\text{Fatigue limit in rotating beam}}{\text{Fatigue limit in direct pull and push}} = 1{\cdot}00 \text{ to } 1{\cdot}35$$

Thus the rotating beam appears consistently higher and the direct pull and push consistently lower when differences are observed.

S—a DIAGRAMS.

When a stress varies in magnitude between unequal extreme values it may be described as a combination of a steady stress (S) equal to the mean between the extremes, and an alternating stress (a) equal to half the difference, see fig. 18.

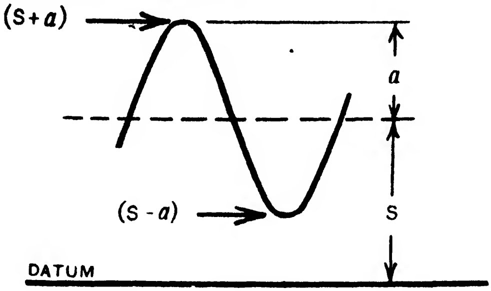


FIG. 18.

The two extreme values are then described as $(S + a)$ and $(S - a)$.

It is found that the limiting safe alternating stress value depends on the value of the steady stress; and that the value can be expressed as follows:

$$a_s = a_0 \left\{1 - K_1\left(\frac{S}{U}\right) - K_2\left(\frac{S}{U}\right)^2\right\}$$

where a_s = fatigue limit when the steady stress is acting, and a_0 = usual fatigue limit found when there is no steady stress.

The coefficients K_1, K_2, etc., can be ascertained only by direct experiment, and their sum is found to be usually greater than unity. Values are given in the following table for a series of metals.

Material.	Tensile ultimate U. Ton per sq. in.	Tensile yield point Y. Ton per sq. in.	Ordinary fatigue limit a_0 Ton per sq. in.	Coefficients. K_1	Coefficients. K_2
Naval brass	28·7	14·5	12·0	1	0
Mild steel	25·2	21·0	13·0	0	1
0·49 per cent. C steel normalised	40·9	21·1	14·7	1	0
" " Sorbitic	43·3	31·2	21·5	0·16	2·5
3½ per cent. Ni steel	55·0	48·5*	26·8	0	1·47
" "	49·8	40·6*	26·8	0	2·26
" "	52·5	42·1*	26·8	—	—
" "	45·3	28·9*	22·0	1·37	0
3·6 per cent. Ni steel	51·3	34·7*	23·0	1	0
0·29 per cent. C steel— annealed	32·3	17·4†	13·0	0·57	0
hard drawn 11 per cent.‡	38·3	32·2†	15·6	0·05	0·75 (tension) 0 (compression)
" " 23 per cent.‡	43·6	38·0†	18·0	0·1	0·75
" " 35 per cent.‡	46·3	37·5†	19·0	0	0·795
Cast iron	14·1	—	4·5	1·6	−0·6
Wrought iron	22·8	14·5	7·1	0	1

* Limit of proportionality or elastic limit.
† Proof stress.
‡ Reduction of area in drawing.

S — *a* AND YIELD.

The relative values of the yield stress and the fatigue strength of a metal have an important bearing on the choice of a material for a given problem. Fig. 19 shows the S — *a* diagram for a high tensile steel, together with lines representing the yield (*yy*) and ultimate tensile strength (*uu*).

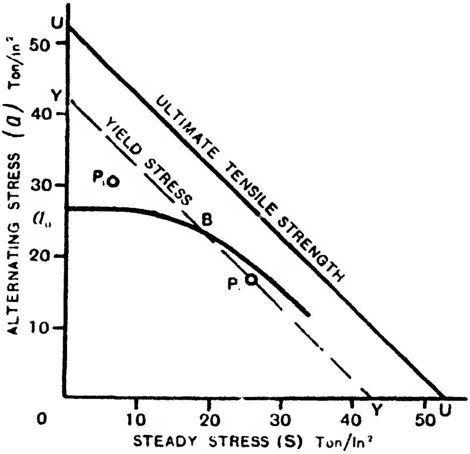


FIG. 19.

Fatigue occurs without yield if or when the state of stress is represented by a point, *e.g.* P_1 in the area a_0YB, and yield occurs immediately if or when the state of stress is changed so that it is represented by a point, *e.g.* P_2 on the line *yy*.

Effect of Temperature.

The fatigue limit does not appear to be greatly affected by rise of temperature, and as the yield point falls rapidly it follows that fatigue failure is unusual at high temperatures in practice. At still higher temperatures creep occurs slowly, and is liable to cause fracture after considerable elongation, or embrittlement without elongation.

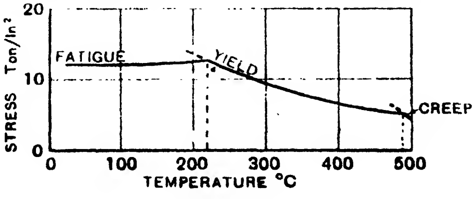


FIG. 20.

Fig. 20 shows how the safe stress is bounded by the fatigue-yield-creep graph for a 0·17 per cent. C steel.

Chemical Action and Fatigue.

Although it was long believed that fatigue was wholly mechanical in nature, experiments published in 1917 showed: (1) that fatigue fracture is accelerated in many metals, although not in all, when water or other reagents act on the surface of the metal during the test; (2) that corrosion prior to the test has much less effect, although the surface may be more severely pitted; (3) that the atmosphere appears in some cases to act as a chemical reagent. Such conjoint chemical and mechanical actions are now known as 'corrosion fatigue.'

Among other features demonstrated in more recent research the following are outstanding: (4) the acceleration of fatigue is reduced when the reagents are heated, (5) the action is markedly influenced by gases in solution, (6) high-tensile carbon steels are affected more than milder in such degree that the corrosion fatigue limit in carbon steels does not vary widely with ultimate tensile strength, (7) in stainless steels the value of the corrosion fatigue limit is higher, particularly in fresh water, (8) in lead fatigue can be delayed by using oil or grease to exclude atmospheric air, (9) in lead fatigue can be eliminated by appropriate chemical action, although this is associated with violent pitting and evolution of gas, (10) in vacuo, the fatigue limit in many instances is higher than in air, (11) in hard-drawn wire, the reduction of the fatigue limit can be almost wholly eliminated by 'galvanising' the wire with zinc.

A series of round figures have been taken from experiments performed by Dr. McAdam and are given in the following table:

Material.	Ultimate tensile strength. Ton/in.^{2}	Corrosion fatigue limit. Ton/in.^{2}	
		Fresh water.	Salt water.
Carbon steels 0·03 per cent.–0·49 per cent. C	19–50	6·7–11·2	—
Nickel-chromium steels	47–66	12·1–12·9	—
3½ per cent. nickel steels	41–56	11·2–12·9	—
5 " "	54–59	8·5–12·0	—
Stainless irons	27–48	13·4–17·8	6·3– 8·0
High chromium steel	49·5	21·4	—
Stainless steels (chromium nickel steels)	38–56	14·3–22·8	6·7–15·2
Nickel	34–59	10·7–11·2	9·8–10·7
Monel metal	37–57	11·2	12·3
Nickel-copper alloys	21–38	9·1– 9·8	8·1–11·2
Electrolytic copper	20·8	7·6	7·6
Aluminium	6–9	2·0	0·9–1·3
Duralumin	15–31	3·5	2·9

Fatigue Strength of Wires and Springs.

The fatigue strength of wires depends to a large extent on the surface finish and strict attention should be paid to this point, especially in such cases as trolley wires, overhead cables and wire ropes which are subjected to alternating stress.

The spiral spring presents a further difficulty in so far that the spring must usually be heat-treated after it is formed.

The wide discrepancy between the fatigue limit of prepared specimens under torsional loads and the fatigue limit of the same steel when coiled into springs was noticed by Lea and Heywood in 1927 (I.Mech.E., April 1927). As a result of their investigations, Lea and Heywood came to the following conclusions amongst others: (*a*) the range of repeated stress (A_s) for a given mean stress (S) is given by $A_s = A_0 - B\,S$. where A_0 is the range of stress at zero mean stress; (*b*) in springs that have been quenched and tempered if the limiting safe range of stress is exceeded, a comparatively small number of repetitions causes failure; (*c*) defects of the wire not brought to light by the tensile test have a considerable effect on the life of the spring. Grinding of the wire preparatory to coiling seems desirable, but does not preclude the possibility of cracks forming due to subsequent heat-treatment.

Experiments on spring plates at the National Physical Laboratory by Batson and Bradley (I.Mech.E., 1931) were performed to determine the effect of grinding away the outer skin of the plate, and also to ascertain if the 'surface effect' of heat-treated spring plates was due to heat-treatment. As a result of these investigations it was found that the condition of the surface layer of the spring steel plates was the principal factor in causing the fatigue limit of the spring to be considerably lower than that deduced from the properties of the metal. The thickness of the layer was small, and when a thin surface layer was removed by grinding, after heat-treatment, the fatigue limiting stress was raised in ratios ranging from 4·75 to 1·75 in different cases. Only slight improvement was observed if the spring plates were heat-treated after grinding.

An interesting paper by Swan, Sutton and Douglas (I.Mech.E., 1931) on valve springs confirms the results of Lea and Heywood in that the removal of the surface layer increases the safe range of stress and reduces the variability. It was found that longitudinal cracks caused premature fatigue failure. The results of torsional fatigue tests on wire at full diameter and on samples which had been reduced in diameter by grinding are given below.

Reduction of diameter. Per cent.	Maximum Values of Limiting Stress Range. Ton/in.^{2}	
	Heat-treated in full diameter.	Heat-treated in final diameter.
0	35	—
10	52 (?)	42
20	58	45
40	64	47

Stress Concentrations.

In practice fatigue cracks almost always start from corners of keyways, sharp fillets, holes, or other discontinuities of section where stresses are concentrated locally. Such stress concentrations being seldom calculable are commonly covered in design by allowing higher 'factors of safety' when dealing with 'live' loads.

The ratios in which different discontinuities and surface finishes reduce the apparent fatigue limit are indicated in the table below, in which the strength of a polished cylindrical test-piece is used as a basis of comparison.

Finish or Discontinuity.	Fatigue limit as percentage of Fatigue limit of Polished specimen.
Emery finish, different grades	90% upwards
Filed finish, different roughnesses	80% „
Scratches produced by use of needle	84%
Sharp changes of section in shafts	50–75%
Changes of section with fillet	70–90%
Small round holes in flanges	40–60%

(Based on results given by Eden Rose and Cunningham (I.Mech.E., 1911), and W. N. Thomas, *Engineering*, 1923).)

Owing to the lower a_e/Y ratio (and the high K_1 value), the weakening effects in high tensile steels of small holes and the like are more evident than in milder qualities. In mild steel in alternating pull and push, the difference in fatigue limit between polished and rough-turned pieces has been found to be less than 5 per cent., but even in mild steel, an unduly sharp radius of curvature on a fillet will reduce the fatigue strength to one-third or less.

Fatigue of Welds.

Structural butt welds, made under practical conditions such as found in the shipyard or in general structural practice, can be relied on to give a fatigue strength of $\pm$ 6 Ton/in.^{2}, provided the usual precautions are taken. Welds in boiler plates, machine-made under ideal shop conditions, will give a consistent strength of $\pm$ 10½ Ton/in.^{2}.

The shape of lap welds so common in ship construction has considerable influence on the strength of the weld due to the bending stresses introduced by eccentricity of loading. If the joint is of the shallow double joggle type the fatigue strength is only $\pm$ 4 Ton/in.^{2}. The single joggled joint often used in the outer bottom plating of ships has a fatigue strength of only $\pm$ 3 Ton/in.^{2}. In this case the fatigue cracks start from the edges of the deposited metal where the bending stresses are greatest and spread through the parent metal.

These figures may be exceeded in special cases, but they afford a sound basis for design in structural work.

Fatigue of Riveted Joints.

When riveted joints crack and fail by fatigue in laboratory conditions, the forms of fracture approximate closely to those of riveted joints cracked in service. Riveted joints rarely if ever fail in the manner observed in tensile tests.

If the plates do not slip fracture occurs in the plate at 0 $\pm$ 3 Ton/in.^{2} reckoned on the gross cross-sectional area of the plate, or 0 $\pm$ 4 Ton/in.^{2} reckoned on the nett cross section between rivets. When slipping occurs, even slightly, under reversing loads failure is due to fatigue in the rivet caused by bending. In this case the rivets are bound to crack at mid-length and also under the heads and points, these often falling off in service. A half-inch diameter rivet under these conditions is found to fail at a load of $\pm$ 5,000 lb.

Hysteresis in Fatigue.

Mechanical hysteresis is readily observed in many metals in a tensile test by means of a sensitive extensometer in conjunction with a suitable testing machine. When the load is applied and removed a hysteresis loop is formed, the area of which is a measure of the quantity of work absorbed. In some metals in the annealed condition this 'primary' hysteresis is considerable. In most cases the area of successive loops frequently becomes smaller with successive stress cycles and after about 250,000 cycles settles down to a much smaller value. Conclusions drawn from 'static' loops or from tests in which the endurance is below 250,000 cycles might be, and often are, misleading when applied to engine vibration. Hysteresis plays an important part in damping forced oscillations such as are met with in crankshafts of engines and propeller shafts. It has been found that hysteresis can be represented very closely by the empirical formula

$$H = Cf^n$$

where H is the hysteresis heat and f the semi-range of applied stress. In steel n lies between 3 and 4. In brasses the values of n vary widely with composition and previous history, and may vary between 6 and 12.

SECTION VI

METERING APPLIANCES

(pp. 191–207)

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

SECTION VI

METERING APPLIANCES.

(Revised by S. A. Wood, M.Sc., M.I.Mech.E.)

METERING OF WATER, AIR, GAS, STEAM, AND COAL.

The metering of fluids is undertaken to enable running costs to be apportioned, sales to be effected on an equitable basis, the mixing of reagents in various chemical processes to be controlled, wasteful or inefficient use to be detected so that working costs may be reduced and unnecessary additions to plant obviated, etc.

Various efficient and accurate types of water and gas meters have been in use for many years. It is only recently, however, that accurate and reliable air, steam and oil meters have been developed. The first large installation of air meters was made on the Rand in 1911, where now more than 400 million h.p. hours per annum are metered, at an ascertained annual aggregate money saving of over £130,000.*

The following is a brief summary of the principal classes of fluid and gaseous meters, with notes as to their application.

1. Meters Based on Volume or Weight Measurement.

(*a*) Tanks or weighing. [For liquids, condensed steam.]
(*b*) Receivers of known capacity discharging between known pressures and temperatures. [For air and gas.]
(*c*) Piston displacement meters. [For water and oil.]
(*d*) Water-sealed drum meters. [For gas.]
(*e*) Bellows meters. [For gas.]
(*f*) Overbalancing bucket meters. [For water or oil.]
(*g*) Conveyor weight meters. [Measure granular material in terms of the weight carried per unit length of the conveyor.]
(*h*) Conveyor volume meters. [Measure granular material in terms of the height of material on the conveyor.]

2. Meters which Measure the Velocity at a Point or Points in the Cross-Section of the Stream.

(*a*) Pitot tube meters. [For gases and non-viscous liquids.]
(*b*) Hot wire, or hot wire plus thermocouple, velocity indicator. [For air and gas.]
(*c*) Impact meters. [For measuring air velocity in mine tunnels.]
(*d*) Anemometer meters. [For water in open channels, or air in conduits and passages.]
(*e*) Floats. [For measuring the velocity of water in open channels. Balloons, smoke, heated air particles can be similarly used for measuring air velocities.]
(*f*) Chain meters. [For coal, etc.]

3. Meters which Depend on the Drop of Pressure due to an Obstruction Placed in the Stream.

(*a*) Weirs. [For liquids in open channels where sufficient 'head' is available.]
(*b*) Venturi flumes. [For liquids in open channels where the available 'head' is small.]
(*c*) Venturi tubes. [For water, air, steam and gas in pipes.]
(*d*) Nozzles. [For gas, water, air, or steam.]
(*e*) Circular, chord, or tongue orifices. [For water, air, gas, or steam.]
(*f*) Variable orifices. [For water, air, gas, or steam.]

* See Hodgson, *Proc. Inst. C.E.*, Vol. CCIV, Part 2, and Mayor, *Trans. Inst. Mining Engineers*, Vol. L, Part 4.

4. Rotary or Turbine Meters.

(a) Which measure the whole of the flow passing. [For water, air, steam or gas.]

(b) Which are placed in a shunt circuit. [For water, air, steam, and gas.]

5. Meters Based on the Movement of a Spring or Weight Controlled Obstruction Placed in the Pipe Line.

(a) In which a disc moves axially in a circular chamber with conical or perforated walls. [For water, gas, steam or air.]

(b) In which a shaped plug opens up an orifice. [For steam, water, gas or air.]

(c) In which a weighted hinged gate is opened by the flow. [For water, gas, steam or air.]

6. Meters Based on the Drop of Pressure caused by Frictional Losses.

(a) In which two pressure connections are arranged at the ends of a length of channel or pipeline. [For measuring irrigation water.]

(b) In which the fluid is caused to pass through a large number of narrow passages arranged in parallel, so that the flow is viscous.

7. Meters Based on the Introduction of a Measured Amount of (a) Heat, or (b) Chemical Reagent, and the Consequent Rise of Temperature of the Fluid or Dilution of the Reagent at some point further downstream.

(a) [For air and gas.]

(b) A method used for measuring the discharge of rivers and canals, recently further developed for the measurement of air and steam flows. Salt solution is usually introduced when measuring water-discharges, and ammonia or carbon dioxide gas when measuring steam or air flows.

8. Meters Based on Increase of Weight due to Chemical Absorption.

Notes.

Meters of classes 1 (c), (d) and (e) can be made accurate to within plus or minus 1 per cent. from zero flow up to the maximum flow which they are designed to measure. The chief sources of error of meters of class 1 (d) are variation in the water level and wear of bearings, and of meters of classes 1 (b) and 1 (e) wear of the valves.

Meters of classes 2 (a), (b) and (c) are usually only accurate down to one-third or one-sixth of the maximum flow for which they are designed, as their indication depends upon the square (or the fourth power) of the velocity of the fluid. Few anemometers are accurate when working at less than one-tenth of their maximum rated capacity.

Meters of classes 3 (a), (b), (c), (d) and (e) are usually accurate down to one-fifth of the maximum flow. If specially sensitive differential pressure-measuring devices are provided, such meters may be made accurate down to one-twentieth or one-thirtieth of the maximum flow. By using a variable orifice, the range over which accurate measurement is possible with any given indicator or recorder is greatly extended. These meters have the very real advantage of having in the pipe line no moving parts which can be deranged by dirt or erosion. The Venturi tube and flume, with their smooth contours which eliminate eddying, are valuable devices when measuring dirty gas or sewage. Meters of class 3 show many ingenious arrangements for obtaining an accurate measurement of the differential pressure over a wide range and for compensating for variations in the specific gravities of the fluids or gases to be measured.

Meters of class 4 are not usually accurate in practice below one-tenth of the maximum flow, though many makers claim much higher ranges of accuracy. Endeavours have been made to increase the range of accurate measurement by opening up additional jets at low flows, or by introducing additional resistance at the high flows. While these devices frequently give excellent results under test conditions, they rapidly get out of adjustment under working conditions. Water meters of this class can often carry considerable overloads, thus greatly increasing the range over which accurate measurements may be obtained. Gas meters of this class can be fitted with devices which automatically correct for variations in gas pressure, or in gas pressure and gas temperature.

Meters of classes 5 (a), (b) and (c) are capable of high accuracy over a large range of flow when measuring clean fluids. They rapidly lose their accuracy, however, when measuring dirty fluids, owing to the small clearances becoming choked.

The accuracy of meters of class 6 (a) is dependent upon an accurate knowledge of the area of the channel and its resistance coefficient. These are apt to change owing to silting up and the growth of weeds. Since with meters of class 6 (b) the drop of pressure is directly proportional to the velocity, the accuracy of measurement is maintained down to very low rates of flow.

Meters of class 7 (a) register in weight units irrespective of the pressure and temperature of the gas.

In the following pages the main features of some of the best known, or the most noteworthy, meters are described and illustrated.

Water Meters.

The Reciprocating Piston Water Meter.

This meter is shown diagrammatically in fig. 1. Its sequence of operations is as follows :—

1. The piston A descends, and the disc P throws over the valve C.
2. The piston B descends, and the disc R throws over the valve D.
3. The piston A descends, and the disc Q throws over the valve C.
4. The piston B descends, and the disc S throws over the valve E ; and so on.

Stops M limit the motion so that an exact quantity of water is measured off each stroke.

Projections K enter the outlet passage at the end of the stroke and prevent jar by slightly throttling the flow.

The cylinders are lined with brass, and the valves and valve faces are made of gun-metal. The valves are caused to oscillate slightly each time they are thrown over in order to keep the rubbing surfaces polished. For the same reason the pistons are caused to rotate a slight amount at each stroke.

The valves are held up against the port block by rollers L carried by springs which press inwards towards the valve faces.

The meter body is jointed along the line XY, which enables all parts to be readily accessible (actually the inlet and outlet passages are at right angles to the positions shown in the diagram).

A recent modification of this type of meter, stated to have a high accuracy, uses four cylinders at right angles, their pistons driving a central crank pin by means of two Scotch-yoke connecting rods.

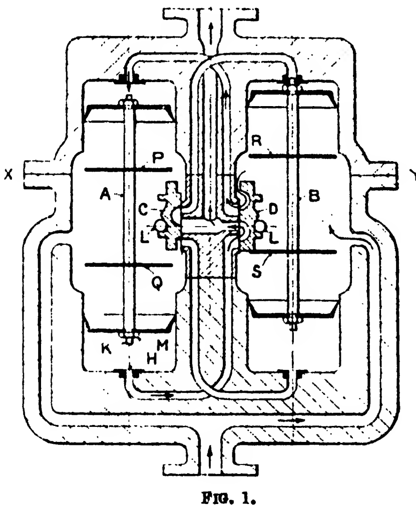


Fig. 1.

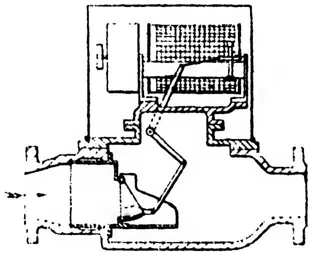


Fig. 2.

The Moving Flap Water Meter.

This meter (fig. 2) consists of a weighed gate pivoted on a horizontal axis and surrounded by a machined shield. The gate opens with increase in the flow, and takes up a definite position at each flow. Its motion is transmitted to the diagram through a stuffing box by means of link mechanism.

It is possible with this meter to measure with accuracy down to very small fractions of the full flow.

Owing to the absence of all sliding parts in this meter, the friction of the mechanism is extremely small.

The bulk of the meter, and the loss of pressure across it, are also reduced to a minimum by its arrangement, which allows of a direct flow between inlet and outlet.

For dirty water, the gate has been replaced in some cases by an aerofoil-shaped vane, the lift on which is balanced by a coil spring.

The Cylindrical Piston Meter.

In this meter (figs. 3 and 4) the motion of a cylindrical piston having a slot down the side is constrained by external and internal cylindrical walls and by a radial partition. The meter is suitable for measuring flows of water, petrol or oil.

For greater uniformity of recording, especially in the larger sizes, the piston is made an elliptical instead of a circular cylinder.

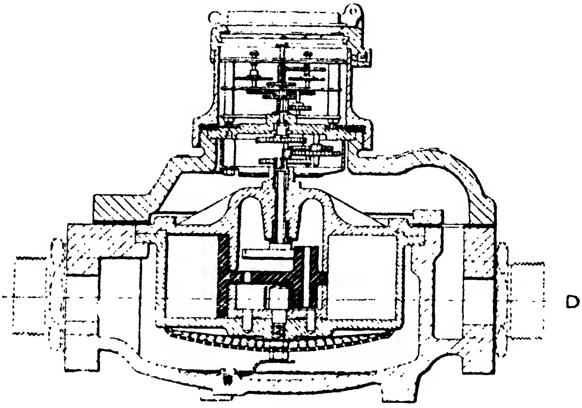


Fig. 3.

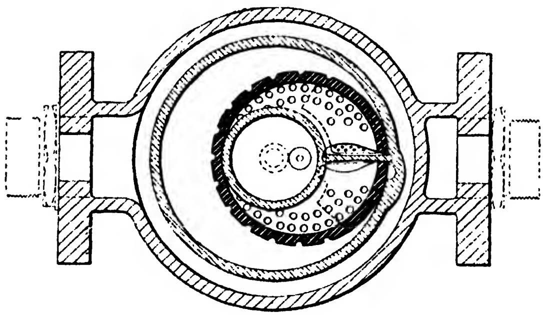


Fig. 4.

The Nutating Piston or Swash-Plate Water Meter.

In this meter (figs. 5 and 6) the spherical valve cap V rocks on its seating S, thus opening the lower parts of each of the three cylinders C successively to the inlet and discharge sides of the meter.

The valve cap V is prevented from turning round on its own axis by the projections Q, which engage in recesses left in the valve seat S. The ball-ended pin at the top of the valve cap thus sweeps out a cone and drives the counter through the arm A and the spindle D. The length of the stroke of the pistons is limited by the flange P on the valve cap V, which comes into rolling contact with a machined surface on the valve seat S.

It will be seen that the valve cap V moves with a lens-grinding action. This ensures that it and the seating will remain polished and unscratched even when gritty water is being measured.

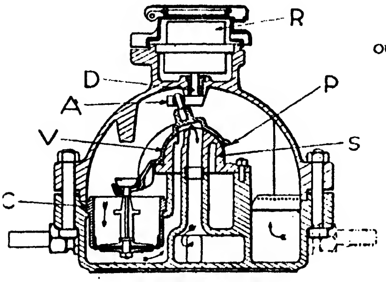


FIG. 5.

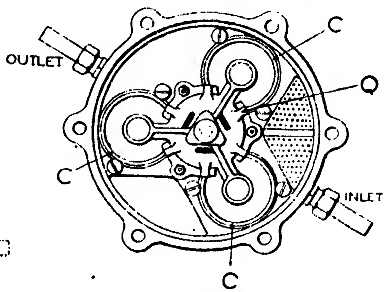


FIG. 6.

The valve cap and the pistons are made of hard gun-metal, the valve seat and bushes of vulcanite, and the cylinders of either vulcanite or gun-metal, vulcanite being used for water of exceptional hardness.

MOVING DISC WATER METER.

In this meter (fig. 7) a weighted disc or 'float' moves vertically in a conical shield. Its motion is transmitted to the diagram by means of thin wire passing through a gland. Guide plates are provided to steady the flow at the entrance to the conical shield. This principle is also used in several air and gas meters.

THE VENTURI WATER METER.

This meter (fig. 8) consists of (*a*) a Venturi tube VT through which the water passes; (*b*) a large mercury U-tube provided with floats FU and FT (which are displaced by the differential pressure which exists when water is passing between the upstream and throat pressure holes of the Venturi tube); and (*c*) diagram counter and recording mechanisms, which are actuated by the movements of the floats.

The motion of the floats is transmitted to the outside of the U-tube by means of gland spindles GU and GT, which actuate toothed wheels that engage with the racks RU and RT.

The law of flow through a Venturi tube is given by the equation :—

$$Q = \frac{\Omega a_1}{7\cdot 88}\left[\frac{hW}{n^2 - 1}\right]^{\frac{1}{2}}$$

where,

Q = flow in lbs. per sec.; Ω = coefficient of discharge for the tube (usually about 0·98); a_1 = area of tube at upstream pressure holes in sq. ins.; $\frac{a_1}{n}$ = area of tube at throat pressure holes in sq. ins.; h = difference of pressure between upstream and throat pressure holes in ins. of water; W = weight in lbs. of 1 cub. ft. of the fluid passing.

One form of counter recording mechanism for this meter operates as follows :—A cylindrical cam LC, which has a slightly raised portion, is rotated once every ten minutes by clockwork. A pivoted arm A, which carries the roller L, which is kept in contact with the surface of the cam by a spring (not shown), is lowered by the rack to which it is attached when the flow increases. The greater the distance the arm is lowered, the larger is the fraction of each revolution of the cam during which the roller is on the unraised portion of the cam. By means of mechanism (not shown) it is arranged that a toothed wheel which drives the counter is put into gear with the clock train during the period that the roller L is on the unraised portion of the cam. The boundary line between the raised and the unraised portions of the cam is so shaped that when the pivoted arm A carrying the roller L moves down a distance H from its zero position an amount proportional to $\sqrt{H}$ is added on to the counter reading. In this way, with suitable gear wheels in the counter train, a counter record of the amount of fluid that has passed is obtained.

Valves U and T are provided so that the upstream and throat pressures may be shut off, and the meter checked by opening the equalising valve E.

Any air that accumulates in the pressure pipes may be blown off through the air vents provided.

To reduce the chance of air getting into the pressure pipes, these should be taken off horizontally, and not at the top, as shown for clearness in the diagram.

These meters will measure accurately down to one-fourteenth of the maximum flow (and special meters of this type down to one-thirtieth of the maximum flow).

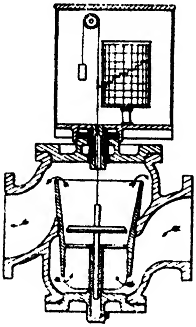


FIG. 7.

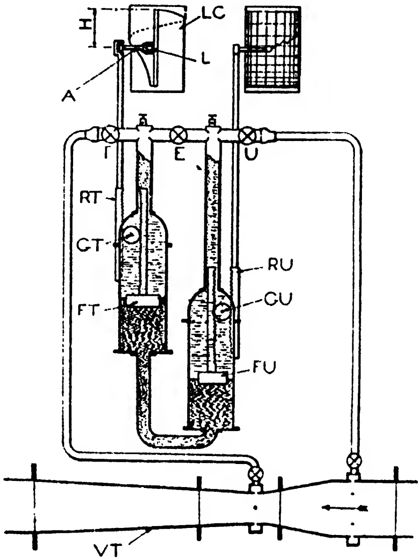


FIG. 8.

The functions of the downstream cone of the Venturi tube is to recover the kinetic energy which exists at the throat, thus enabling the major portion of the pressure difference which has been created between the upstream and throat pressure holes to be recovered.

The friction loss due to the insertion of the water Venturi tube is about 2 lbs. per sq. in. at the maximum flow.

Venturi meters have been used to measure water in all sizes of pipes from about ½ in. to 20 ft. in diameter. They provide the most convenient and accurate method known of measuring the flow of water in large mains. There is a minimum of obstruction to the flow, and there are no moving parts in contact with the flowing water.

Automatic Water Weighers.

In the automatic water weigher illustrated in fig. 9 the weight box B is suspended at one end of an equal-armed beam A, and the weigh hopper C at the other.

A quantity of water, determined by the weights in the weight box B, is allowed to enter the weigh hopper C through the feed valve D.

When the correct amount of water is obtained, the supply from the feed valve or valves D is, by means of a combination of levers, automatically cut off.

Immediately after the supply is cut off, the weigh hopper C automatically tips and discharges its load.

As soon as the contents of the weigh hopper C have been discharged, the weigh hopper automatically returns and assumes the weighing position, at the same time opening the feed valve or valves D, and the cycle of operations is repeated.

On the large-capacity machines there are two valves at D, the larger one feeding the main flow into the weigh hopper C, then closing, with the result that the final dribble of liquid to complete the weighing and ensure accuracy is fed through the supplementary small valve D.

A mechanical counter keeps an exact record of the number of times the machine operates.

Overbalancing bucket meters, which consist of two triangular buckets fixed together and pivoted below their common centre of gravity, are largely used in oil measurement. They operate

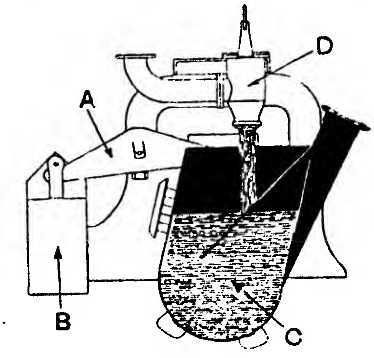


Fig. 9.

without interrupting the flow, and also usually take a small sample of the fluid passing each time they tip over. Where the oil is viscous and sticks to the buckets a correction is applied.

Weir Recorders and Venturi Flumes.

'V' Notch Weir Recorders.

These recorders are specially suitable for the measurement of boiler feed water and condensates, also rivers, streams, reservoir supplies, sewage effluents, etc., or in such cases where it is necessary to measure the water in some form of open channel.

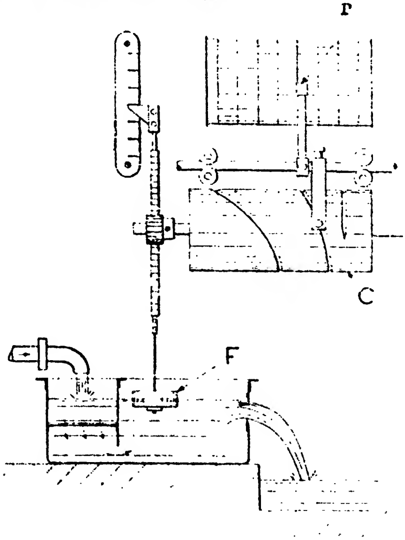


Fig. 10.

The discharge for a 90° V notch with sharp edges bevelled at 45° on the downstream side is given by the equation.

$$Q = 2{\cdot}48H^{2{\cdot}48}$$

where,

Q = discharge in cub. ft. per sec.; H = head, in ft. above the bottom of the notch.

In the Recorder shown in fig. 10 the movement of the float F in the weir chamber, which is proportional to H, is transmitted to a cylindrical cam C. By means of this cam a movement of the pen arm P. which is proportional to Q, is obtained.

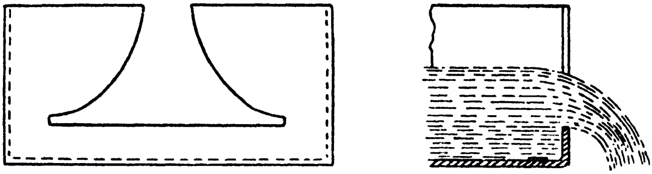


FIG. 11.

In another type of weir recorder **mechanism a flat cam is used, and the straight line movements are replaced by movements about fixed pivots, while in a third type the notch is made of such a shape that the motion of the float is proportional to the quantity passing. In this last recorder there is, therefore, no cam between the float and the recorder pen. The approximate shape of the weir notch is shown in fig. 11.**

Weirs, when used for stream measurement, are liable to considerable errors due to wind effects and other causes. A Venturi flume, or a nest of Venturi flumes (such as may be formed by the specially constructed piers of a bridge) enable much more reliable results to be obtained.

The discharge of a Venturi flume is given by the formula,

$$Q = C\sqrt{2g}\, bH^{\frac{3}{2}}x\sqrt{\frac{1-x^2}{1-S^2x^2}}.$$

where Q = discharge in c.f.s., C = coefficient of discharge (to be determined experimentally), b = width at throat (ft.), H = depth upstream of contraction, x and S = ratios of depth and width at throat to depth and width upstream, respectively.

Gas Meters.

THE 'DRUM' TYPE STATION GAS METER.

In this meter (fig. 12) gas enters through the hollow axis in the four compartments of the drum in a pipe which is put just above the water level, and fills each compartment successively, and by

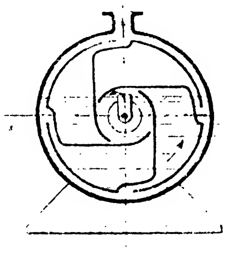


FIG. 12.

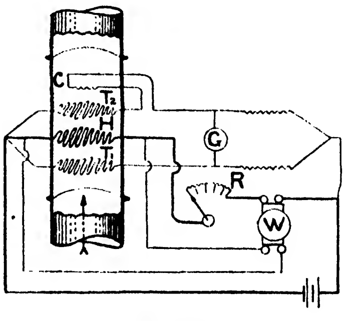


FIG. 13.

its pressure causes the drum to revolve in the direction of the arrow and to register the volume passed on a set of dials.

Meters of this type are used to measure gas in mains of all sizes, from ½ in. upwards.

Owing to their costliness, the larger meters of this description are now being superseded by meters of classes 3, 4 and 7.

The main sources of error of this meter are deformation or corrosion of the drum, deposition of naphthalene in the drum, and variations in the water level. There are various ingenious devices for preventing the meters from being over-filled, and for maintaining the correct water level. If the meter is overloaded, the water level is depressed, and the meter reads slow.

The Heat Injection Gas Meter.

In this meter (fig. 13) the gas is passed through a short length of pipe which contains two electrical resistance thermometers T_1 and T_2, which form part of a Wheatstone Bridge. An electrical heater H is placed between the two thermometers. Any increase in the flow of gas causes a lowering of the temperature of the downstream thermometer and a deflection of the needle of the galvanometer G. This is made automatically, by means of a suitable mechanism, to reduce the resistance in the heater circuit, so that the temperature of the gas is always raised the same amount (about 2° F.) by the heater.

The readings on a Wattmeter W, placed in the heater circuit, give the number of cubic feet of gas reduced to standard temperature and pressure.

The Venturi Gas Meter.

In this meter (fig. 14) the 'Venturi Head' $(p_1 - p_2)$ is measured by means of the bell B and an amount proportional to $(p_1 - p_2)$ is added on to the counter reading by means of the cam, feeler, ratchet, and pawl device C.F.R. every time the cam G, which is driven by the wet gas meter W, makes one revolution

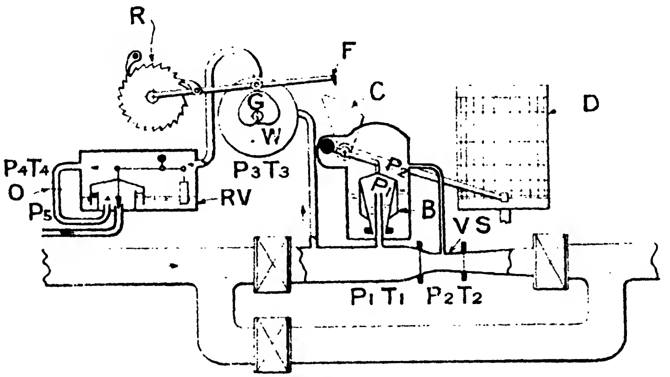


FIG. 14.

The speed of rotation of this meter is determined by the amount of gas that escapes to atmosphere through a small orifice at O, across which the pressure difference is maintained constant by means of the regulating valve RV.

In this way the counter is made to register the volume of gas passing through the Venturi tube, irrespective of variations in the specific gravity of the gas, the registration of the meter being identical with that of the ordinary Drum type Station Gas Meter.

A diagram, D, is provided which shows the rate of flow of gas at any instant. The readings of this diagram are, however, not corrected for variations in the specific gravity of the gas.

If tar, naphthalene, or other depositable matter settles in the Venturi throat, the meter tends to read fast. It is found that when measuring coal gas, such deposits can be entirely prevented by keeping the walls of the throat at a few degrees higher temperature than that of the gas passing through.

The Venturi tube is installed on a by-pass, and the throat section VS is made so that it can easily be swung out of the pipe line for inspection and cleaning.

The loss of pressure at the maximum flow due to the insertion of the gas Venturi tube is arranged not to exceed 1 in. of water gauge.

The range of accurate registration is down to one-tenth or one-fifteenth of the maximum flow.

If it is desired to reduce the meter readings to standard cubic feet of dry gas, corrections must be made for variations in the barometric pressure, atmospheric temperature, and the tension of aqueous vapour by means of tabular numbers, as is the case with the Drum type Station Meter.

The Venturi Gas Meter is also made in a cheaper form in which the integrating mechanism is driven by clockwork, and any variation in the specific gravity, temperature, and moisture content of the gas is corrected for by means of tabular numbers, *i.e.* the meter readings are multiplied by

$$\frac{(p_1 - p'_1)}{p_1}\sqrt{\frac{W_D}{W_S}} \cdot \frac{p_1}{T_1} \cdot \frac{T_S}{P_S}$$

where,

W_D = weight of a cub. ft. of dry gas at p_1 and T_1; W_S = weight of a cub. ft. of gas containing water vapour at p_1 and T_1; p'_1 = pressure of water vapour at temperature T_1, P_S and T_S the standard temperature and pressure.

See Hodgson. *Proc. Inst. C.E.*, vol. cciv., pp. 164–165.

THE ORIFICE GAS METER.

This uses a plate orifice with upstream and downstream pressure tappings either to the two sides of a U-tube with dial indication or (for low pressures) to a ring balance, *i.e.* an annular chamber partly full of liquid, to each of the free surfaces of which the pressure from one side of the orifice is applied. The chamber swings to regain its balance and in doing so moves an indicating pointer.

THE ROTARY GAS METER.

In this meter (fig. 15), a counter C is actuated by an anemometer fan F, which rotates about a vertical axis.

The gas, after being carefully guided by a series of nozzles N into a vertical direction, impinges on the underside of the blades, thus helping to take the weight of the fan off the bottom pivot. The rate of revolution of the fan is proportional to the actual volume of gas passing, except at low flows, when the friction of the mechanism is appreciable. At these flows a weighted valve V (which lifts at the higher flows) causes the whole of the gas to pass through a few jets J, and so to impinge on the fan at an increased velocity. In this way the friction of the mechanism is compensated for, and accurate readings obtained down to one-tenth of the maximum flow.

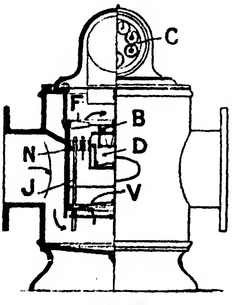


FIG. 15.

The dashpot D is provided to steady the motion of the weighted valve. A brake B is attached to this valve, which stops the fan immediately the flow is turned off, as the valve then falls.

This meter registers the actual volume of gas passed, and is subject to the same corrections for variations in the temperature pressure and moisture content of the gas as is the ordinary Drum type Station meter, or the Kent-Hodgson Venturi Gas meter.

Clogging of the nozzles through which the gas discharges on to the fan causes the meter to read fast. A special form of meter in which the nozzles can be cleaned from the outside, and in which any deposit thrown off from the fan does not accumulate r. . .d it, has been devised for measuring dirty gas.

The Shunt Gas Meter.

This meter (fig. 16) consists of a body casting inserted in the main, or by preference in a by-pass in the main, a turbine T placed in such a position that gas passing through the two nozzles J on the upstream side of the orifice K impinges on its blades, and a counter mechanism actuated by the turbine. The turbine, orifices, and nozzles can be inspected and cleaned through the hand-holes provided.

The meter carries the measurement of gas on the rotary principle to the utmost refinement. The speed of rotation of the turbine is kept low by means of the damping fan F, so that wear on the bearings is reduced to a minimum. The weight of the turbine is carried on a mercury float, so that the all-important pivot bearing never wears appreciably, and is also protected by being covered with mercury. The upper thrust bearing and the high-speed wheels of the counter train run in an oil bath. The motion of the turbine is transmitted to the counter dial by means of a magnetic drive similar to that shown in fig. 16, thus eliminating all gland friction and the possibility of moisture depositing on the counter glass.

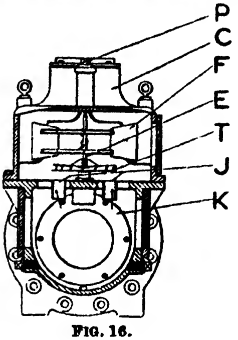


Fig. 16.

The turbine chambers and the mechanisms they contain are identical, whatever be the size of the main. The only parts of the meter which vary in size are the body casting which contains the orifice, the orifice itself, and the change wheels which are fitted into the counter train. The length of the body casting in practically all sizes is 24 ins., so that the meter is very light to handle and requires only a short length of pipe line for its installation. It is accurate down to one-tenth of full flow, and will stand considerable temporary overloads without damage.

The Hot Wire Anemometer.

In this instrument a bare wire with a large temperature coefficient is heated electrically and exposed to the gas flow whose velocity it is required to measure. The velocity V of the gas can then be obtained from the relation

$$I^2 = A\theta + B\theta\left[V.W\,K_p\right]^{\frac{1}{2}}$$

where,

θ = temperature elevation of the wire;
I = current in amps.;
V = velocity of the gas;
W = specific weight of the gas;
K_p = specific heat of the gas at standard pressure
A & B = constants.

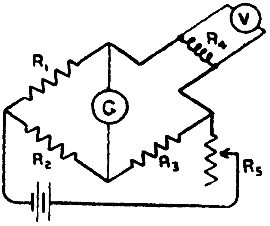


Fig. 17.

Referring to the diagram (fig. 17), R_1, R_2, and R_3 are resistances with no temperature coefficient, R_4 is the anemometer wire which is maintained at constant temperature when the gas velocity varies by adjusting the resistance R_5 so that the needle of the galvanometer G remains at zero. The current through R_4 is thus proportional to the reading of the high resistance voltmeter V.

The formula given above holds for ordinary gas speeds (9·5 to 30 ft. per second). At speeds below these, free convection currents carry away a considerable proportion of the heat, and the formula requires modification.

The accuracy of the instrument is affected by changes in the temperature of the gas itself, and for this reason it is desirable, when high accuracy is required, to work at a temperature elevation of about 1,000° C. In this case the anemometer wire should be of platinum. For ordinary work a temperature elevation of 200° C. or 300° C. is sufficient, and the anemometer wire may be made of electrolytic nickel.

It will be noticed that the velocity of the gas is approximately proportional to the fourth power of the current. This makes the accurate range of the instrument small, and it also makes it unsuitable for obtaining the mean velocity of fluctuating flows.

In a modification of the instrument the four arms of the Wheatstone bridge are made of electrolytic nickel. The wires which compose the two opposite arms of the bridge are shielded from the air current, and the two remaining wires are exposed to it. The instrument is thus automatically compensated for changes in the air temperatures, so that it is not necessary for the temperature of the wires to be more than 100° F. above that of the air current to be measured. A constant potential difference is maintained across the bridge, and the air velocity is measured in terms of the out-of-balance current of the galvanometer.

In a further modification, an electrically heated platinum wire and a thermocouple are enclosed in a double bore quartz sheath, and exposed to the wind velocity to be measured. If a constant current is sent through the platinum wire, the thermocouple reading will be a measure of the wind velocity.

The instrument is calibrated against known velocities in a wind tunnel.

An instrument of this type is adaptable for recording the proportion of any gas (*e.g.* CO_2) in the air travelling through a duct, since the heat lost from the wire will vary according to the composition (and therefore heat conductivity) of the ambient gas.

The Viscous Flow Air Meter.

This meter is designed to eliminate the serious errors which arise with pulsating flow in air meters of the orifice, Venturi, or other 'kinetic' types unless a very large smoothing capacity is provided. Those errors are due partly to 'root-mean-square' velocity effects and partly to the effect of the flow in and out of the connections to the indicating manometer as the pressure drop across the meter varies.

In the viscous flow air meter, the 'meter element' is a honeycomb of long narrow triangular passages each about 0·02 in. in width, formed by winding upon a core alternate layers of flat and corrugated strip metal. The manometer connections are tubes, spanning the upstream and downstream faces of the element, each with a row of holes along one side, those on the upstream connection facing downstream and vice versa.

Within the working range of the meter, the velocity through the passages of the element is below the critical, so that the flow is viscous and the resistance of the element is directly proportional to the velocity; this automatically eliminates the 'root-mean-square' error. The construction of the manometer connections provides a reverse kinetic head which automatically corrects the small pressure drop in the element due to entry effects, and the error due to flow in the manometer connections is corrected by providing these with felt pads which render the flow in the connections viscous.

Since the pressure difference across the meter is proportional to the flow rate, the range of flow measurable with reasonable accuracy is greater than with 'kinetic' meters where the pressure varies with the square of the flow rate. Viscous meters are, of course, not absolute standards and must be calibrated against a standard measuring nozzle or orifice. After proper calibration, a viscous meter can be relied on to repeat its readings to within 1 per cent. In order to prevent clogging of the meter element, an air filter is usually fitted upstream of the element.

Steam Meters.

The Orifice Steam Flow Meter.

A typical pressure-corrected steam-meter (fig. 18) consists of an orifice O placed in the steam main, two cooling chambers OU and OD (whose function it is to condense any steam that passes into them and to serve as reservoirs of water), and pressure pipes PU and PD which transmit the difference of pressure between the upstream and downstream sides of the orifice, due to the steam flow, to the two sides of the differential pressure diaphragms D and to the inside of the pressure diaphragms P.

Any increase of the flow causes an increase of the difference of pressure between the upstream and downstream sides of the orifice, and consequently causes the diaphragms D to contract and the pointer B to move about its pivot C (the motion of the diaphragms D is transmitted to the outside of the diaphragm case through a stuffing box G); while any increase in the pressure of the steam causes the diaphragms P to expland and the arm H to descend.

Air vents V and an equalising valve E are fitted.

The product of the angular movement of the arm B (which is proportional to the differential pressure $(p_1 - p_2)$) and the travel of the roller R in its slot (which is proportional to the absolute pressure of the steam p_1) is transmitted to the pen arm A, which is thus given a motion proportional to $(p_1 - p_2)p_1$.

Since the rate of discharge of steam through an orifice is represented approximately by the equation,

$$Q = K \sqrt{(p_1 - p_2)p_1}$$

where,

p_1 = absolute pressure of the steam on the upstream side of the orifice; p_2 = absolute pressure of the stream on the downstream side of the orifice; Q = lbs. of steam per hour; K is a constant depending upon the size and proportions of the orifice used.

THE ORIFICE STEAM FLOW METER.

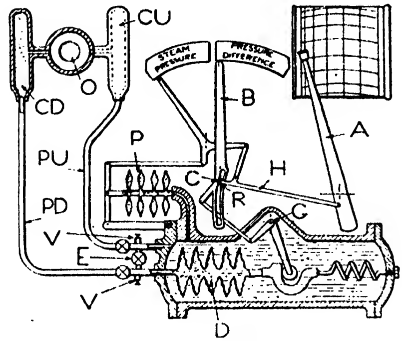


FIG. 18.

THE MERCURY U-TUBE STEAM METER.

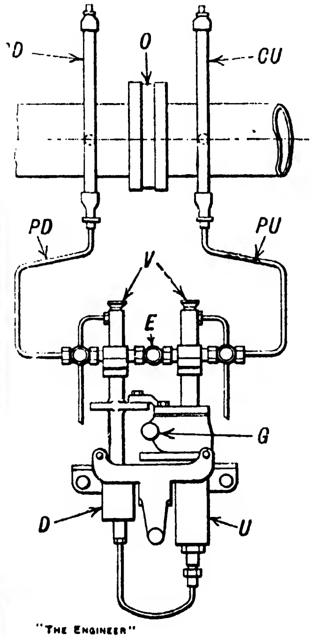


FIG. 19.

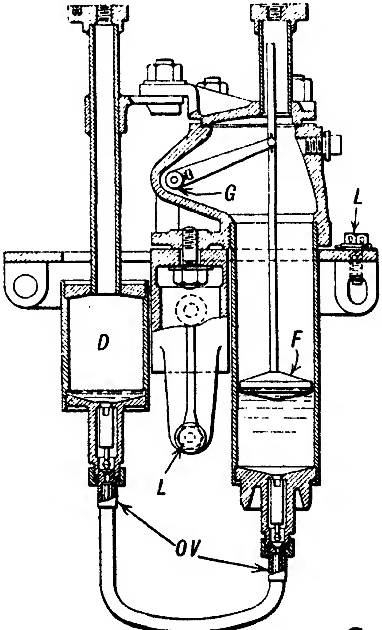


FIG. 20.

the diagram can be divided to show the rate of steam flow at any instant, automatically corrected for variations in the pressure. (The corrections for variations in the superheat or wetness of the steam* are made by means of tabular numbers.)

These meters can be arranged to record by counter also if desired.

With any given orifice plate the meter is accurate down to one-fifth of the maximum flow for which the plate is designed to deal. If it is desired to measure over a greater range of flow than this, a new orifice plate must be inserted. Each meter is designed to work in connection with any orifice plate. Each orifice plate is calibrated with actual fluid before leaving the works, and the accuracy obtained in practice with this meter is within plus or minus 1 per cent.

With slight modifications, such a meter can be used for the measurement of compressed air.

This meter, which is shown installed (fig. 19), and with the mercury pots in section (fig. 20), consists of an orifice, O, and cooling chambers, OU and OD, as in the case of the meter shown in fig. 18. The differential pressure across the orifice is measured by the motion of a cast-iron float, F, which transmits its motion to the gland spindle, G. The motion of the gland spindle is made to indicate or record the flow as desired. Overload valves, OV, are provided to prevent the mercury being blown out. Levelling screws, L, are also provided. The downstream pot, D, is made easily interchangeable so that a higher or lower maximum 'head' may be worked at.

This type of mercury pot unit is suitable for measuring air, gas, steam, water and oil flows.

The Rotary Steam Flow Meter.

This meter is shown in section by fig. 21. A portion of the steam passing through the instrument is diverted by means of an orifice plate through nozzles A, by means of which it is caused to impinge on a turbine. The speed of this turbine is kept low by means of a damping fan, O, which rotates in water, resulting in a minimum of wear on the turbine bearings. The retarding torque at any given speed due to this fan is made so great that any variation in the corresponding torque due to the turbine, caused by variations in the density of the steam in which it rotates, can be neglected. The turbine rotates at normal flows with a speed proportional to $Q\sqrt{W}$, where Q is the weight of steam passing per second, and W is the density of the steam in pounds per cubic foot. From this it will be seen that a variation of 2 per cent. in the density of the steam will involve a correction of only 1 per cent. in the flow reading. Besides reducing wear on the pivots, the use of the damping fan increases the range of the meter, since it increases the relative velocity between the impinging steam and the turbine. The counter E is mounted in a compartment below the main, a magnetic drive D from the turbine being employed so that leakage of steam or water into this compartment is impossible. Radiating fins assist in keeping the counter cool. The maximum flow which the meter will measure can be altered by changing the orifice plate B; and by fitting a blank plate in place of this orifice plate, and putting the meter in a by-pass in a large main into which a suitable orifice plate is inserted directly, the largest mains can be metered at a small cost.

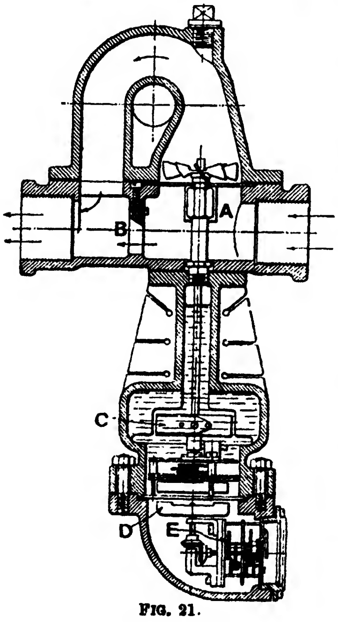


Fig. 21.

The ratio of the revolutions per minute to the rate of flow is constant.

The meters are accurate down to $\frac{1}{10}$ of the maximum flow, and will stand temporary overloads of 100 per cent. without damage.

The same type of meter is often used for compressed air in pipes up to 2 ins. diameter.

* About 1 per cent. for every 2 lbs. change in the pressure, or 20° F. change in the superheat, or 2 per cent. change in the wetness.

Conveyor Volume Meters.

The object of this meter, illustrated in fig. 22, is to measure the amount of coal consumed by a chain grate mechanical stoker.

The coal, as it passes from a hopper underneath a movable fire door, assumes the form of a continuous stream of rectangular cross-section. The volume of coal, therefore, having passed during any given time is the product of the width of the chain grate, the height of coal as it passes under the fire door, and the linear distance travelled by the grate.

Of these three factors two are variable : firstly, the depth of coal passing over the grate (due to the variation in lift of the fire door) ; secondly, the speed of the grate (or the distance travelled by the grate over a given time). The meter is so constructed as to take both these variables into account in the following manner :—

A specially cut toothed drum, having at one end teeth cut round the whole of its periphery, and a uniformly decreasing number of teeth cut along the whole of its length, is driven by a small driving chain from the grate itself ; thus the speed of the drum is proportional to that of the grate, and accordingly proportional to the linear distance travelled by the grate over any given time.

Meshing with this toothed drum is a sliding pinion, which is directly connected to the fire door by means of a steel cable ; its position on the toothed drum, and consequently the number of teeth with which it engages, is, therefore, dependent on the amount to which the fire door is raised or lowered. Thus, the rotation of this pinion is dependent on both the grate speed and the fire door lift, and, consequently, this rotation is in a direct proportion to the volume of coal having passed.

The principle of operation of this Meter is as follows :

If W = width of grate in ft. ; T = thickness of fire in ft. ; V = velocity of grate in ft. per hour ; then, cubic ft. per hour = W × T × V = cross-sectional area × V.

The rotation of the pinion is transmitted directly to a row of seven counting dials, the movement of which gives a visible indication of the volume of coal having been fed on to the grate.

Such Coal Meters can be fitted with a ' Rate of Flow ' Indicator, which, when worked in conjunction with the various speeds of the Stoker drive, gives, on specially graduated scales, the actual amount of coal being consumed at any moment in lbs. per hour.

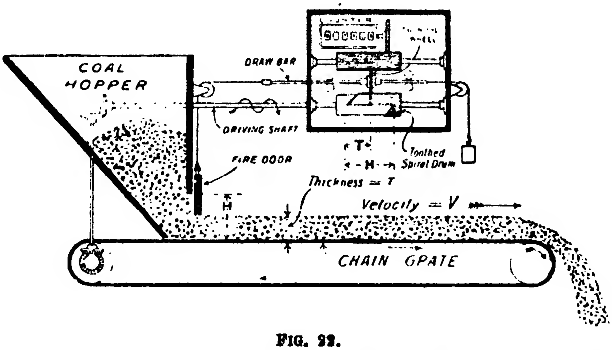


FIG. 22.

A modification of the above meter enables it to measure bulk supplies of coal, ores, grain, or any other similar granular material, in a continuous manner, in conjunction with elevators or conveyors, on which the material is being handled.

The principle of measurement is volumetric, and is as follows :—

The material to be measured is passed on to an endless belt, in a similar manner to the coal passing on to the grate of a chain grate stoker. The meter itself is similarly driven by a small chain from this belt.

As the material travels over the belt it passes underneath a swinging door, the lift of which varies with the thickness passing under it. Directly connected to this swinging door is a link

motion which operates the sliding pinion in the meter. Thus the rotation of the meter dials is proportional to the speed of the belt, and also to the height of the material passing along it, the two factors being combined in the meter giving a true indication of the volume of material having passed through the machine.

These machines may be accurate within 2 per cent. by volume.

In a meter of the volumetric type for measuring coal the counter is operated directly by the rams, or pusher plates, of stokers of the Proctor, Bennis, and other similar types. A granular substance such as coal may also be measured volumetrically by means of a counter, provided that this counter is so arranged as to take into account variations of both the speed at which the rams are operated and the differences of travel of the rams, so that a true record of the amount of coal fed into the furnaces is given.

Such a meter may be made to be correct within 2·5 per cent. by volume.

Conveyor Weight Meters.

These are usually applied to belt conveyors.

They record the product of the weight of stuff carried per unit length of conveyor passing a given point, and the total length passed.

In one such conveyor weigher the pull on a steelyard arm is balanced by a cylindrical float immersed in mercury, the movement of the arm being thus made proportional to the weight on the weigh-bridge. Integration is effected by means of a feeler and free-wheel device, the steelyard arm being clamped while the feeler is in contact with it.

Automatic Coal Weighers.

A coal weigher similar to the water-weigher shown in fig. 9 is shown in fig. 23.

In this instrument a weight box B is suspended at one end of an equal-armed beam A and the weigh hopper C at the other.

A quantity of coal, determined by the weights in the weight box B, is allowed to enter the weigh hopper C through the feed shoot controlled by the valve D.

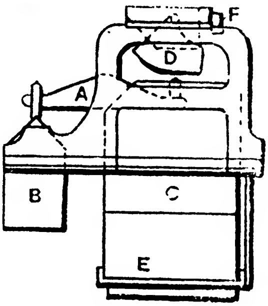


FIG. 23.

When the correct amount of coal is obtained, the supply from the feed shoot is, through a combination of levers, automatically cut off by the valve D.

Immediately after the supply is cut off, the door E of the weigh hopper C automatically opens and the load is discharged.

When the load is clear of the weigh hopper door E it closes again and automatically opens the feed valve D, and the cycle of operations is repeated.

A mechanical counter F keeps an exact record of the number of times the machine operates.

Chain Type Chute Coal Meter.

In this type of meter an endless chain passes down the chute, embedded in the coal, and then runs over a sprocket geared to an electric generator connected with a remote indicator or recorder.

Notes on the Use of Orifices, Nozzles, and Venturi Tubes for Test Purposes.

Q = discharge in lbs./sec.;
E = coeff. of discharge;
a_1 = area of main in sq. ins.;
a_2 = area of orifice or throat of nozzle in sq. ins.;
n = a_1/a_2;
N = $a_1 \div \sqrt{n^2 - 1}$;
d_1 = dia. of main in ins.;
d_2 = dia. of orifice on throat or nozzle in sq. ins.;
h = differential pressure across orifice or nozzle in ins. of water;
p_1 = abs. pressure in lbs./sq. in. at upstream pressure hole;
p_2 = abs. press. in lbs./sq. in. at downstream or throat pressure hole;
T_1 = abs. temp. in F.° at upstream pressure hole;
W_1 = specific weight of fluid at upstream pressure hole in lbs./cu. ft. = $2 \cdot 7p_1/T_1$ for air (values for water, steam, and other fluids can be obtained from various published tables):
μ = viscosity of the fluid in foot, pound, sec. units;
γ = ratio of the specific heats if the fluid is gaseous.

The discharge through an orifice, shaped nozzle, or Venturi tube for gaseous or liquid flows can be calculated from the formula:

$$Q = 0 \cdot 127\ EN\sqrt{hW_1}\ \text{lbs./sec.}$$

The value of the coefficient of discharge, E, is the same for the same values of the viscous flow criterion $Q/d_2\mu$, whatever be the density and viscosity of the fluid used, provided that when the fluid which is being metered is a gas, it does not appreciably change in density as it passes through the orifice or nozzle (*i.e.*, p_2/p_1 should be between $1 \cdot 0$ and $0 \cdot 98$).

For shaped nozzles and Venturi tubes, if the numerical value of the viscous flow criterion $Q/d_2\mu$ is greater than 3,000,

$$E = 0 \cdot 95–1 \cdot 00.$$

For orifices:

$$E = 0 \cdot 605–0 \cdot 615,$$

provided that:

(*a*) The numerical value of viscous flow criterion $Q/d_2\mu$ is greater than 10,000;
(*b*) d_2/d_1 is not more than $0 \cdot 6$, and that orifices are used which have;
(*c*) square edges if the orifice plate is thin (less than $\frac{1}{30}$th the diameter of the orifice);
(*d*) square edges bevelled off on the downstream side if the orifice plate is thick;
(*e*) pressure holes not more than one pipe diameter upstream, and not more than half a pipe diameter downstream of the orifice.
(*f*) the orifice or nozzle is installed with at least 10 diameters upstream and 5 diameters downstream of straight parallel pipe free from bends, tees, branch pipes and control cocks.

If the flow is gaseous and pulsating, sufficient capacity or throttling must be introduced between the source of pulsation and the metering point to smooth out the pulsation of flow at the metering point, otherwise the meter will read fast. In the case of liquid flows, an air vessel or an open tank may be used.

If gaseous discharge takes place above the critical pressure ratio (*i.e.* if p_2/p_1 is less than $\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$ when $n = \infty$), the following formulæ hold for shaped nozzles:

$$Q = 0 \cdot 325\ EN\sqrt{p_1W_1}\ \text{for air;}$$

$$= 0 \cdot 316\ EN\sqrt{p_1W_1}\ \text{for steam.}$$

For fuller treatment see Hodgson on 'The Orifice as a Basis of Flow Measurement,' *Proc. I.C.E.*, 1925; 'The Metering of Industrial Fluids,' *Proc. Inst. Mining Eng.*, Nov. 1927.

'The Measurement of the Flow of Gases and Liquids by Means of Orifices, Nozzles and Venturi Tubes,' World Engineering Congress, Tokyo, 1929, also Finnicome, *Mechanical World*, Vol. 120, 1946, p. 371, No. 3118. 'Flow through standard nozzles, orifice plates and venturis.'

For further information on water and gas flow measurement, see British Standard Specifications 599—1939 (amended 1945) and 1042—1943 (amended 1946–48).

SECTION VII

BEAMS — GIRDERS — STRESSES — BENDING MOMENTS — INFLUENCE VALUES — MOMENT OF INERTIA — DEFLECTION — COLUMNS — STANCHIONS — RIVETING — STEEL ROOF FRAMING AND COVERINGS — BRITISH STANDARD SECTIONS

(pp. 211-245)

(Revised by J. D. W. Ball, A.M.I.C.E.)

SECTION VII

BEAMS — GIRDERS — STRESSES — BENDING MOMENTS — INFLUENCE VALUES—MOMENT OF INERTIA—DEFLECTION — COLUMNS — STANCHIONS — RIVETING — STEEL ROOF FRAMING AND COVERINGS—BRITISH STANDARD SECTIONS.

(Revised by J. D. W. Ball, A.M.I.C.E.)

STRESSES ON BEAMS AND GIRDERS.

When designing framed structures, the external forces (or those acting *on* the structure, are always known or assumed, and from these, and the configuration or design of the structure, the internal stresses set up in the several portions of the structure can be determined.

The external forces may be assumed as acting transversely, i.e., at right angles to the beam, or girder, the supporting forces or reactions necessarily acting parallel to and in the opposite direction to the applied forces. Where any force acts other than transversely, it should be resolved into two component forces, one acting in the direction of the length of the beam (and causing compression or tension throughout the complete cross-section), and the other acting transversely: to obtain the total stress at any section of the beam, the tension or compression due to the former component must be added algebraically to the stress from the transverse component.

The beam or girder, whatever its form may be, will tend to shear transversely, and to bend under any system of loading, and the values of these tendencies at any section are termed the 'shearing stress' and the 'bending moment' values at that particular section.

At any vertical section throughout a beam the shearing stress to be resisted is the algebraic sum of all the vertical forces acting to the right or the left of the section chosen; the forces being reckoned, say, as positive or negative as they act in an upward or downward direction.

For any series of transverse forces W_1, W_2, W_3, etc., acting to one side of the chosen vertical section, the general formula for the shearing stress is:—

$$S = \Sigma W = \pm W_1 \pm W_2 \pm W_3, \text{ etc.} \qquad (1)$$

At any vertical section throughout a beam, the bending moment to be resisted is the algebraic sum of the moments of the several vertical forces acting to the right or left of the chosen section, these moments being reckoned, say, as positive or negative as they tend to turn the beam about the section line in a clock-way or anti-clock way.

Thus, for any series of transverse forces W_1, W_2, W_3, etc., acting at the several horizontal distances l_1, l_2, l_3, etc., from the chosen vertical section, the general formula for the bending moment is:—

$$M = \Sigma(Wl) = \pm W_1 l_1 \pm W_2 l_2 \pm W_3 l_3, \text{ etc.} \qquad (2)$$

(In the case of uniformly distributed loads the value W is that for the total load under consideration, the horizontal distance being then measured from the chosen section to the centre of the load.)

Where the supports to a beam are not more than two in number, and the ends of the beam are free, the reactions or supporting loads can readily be found by the law of the lever. These values are then added to the list of external forces, and the shearing forces and bending moment values calculated from formulæ (1) and (2).

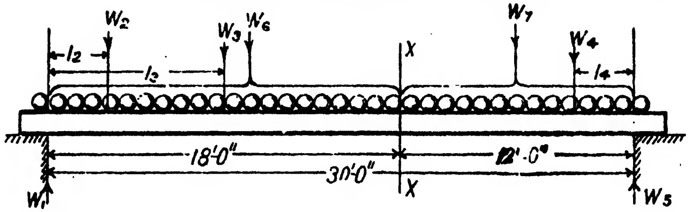


FIG. 1.

EXAMPLE :—A beam 30 ft. clear span between reactions carries three concentrated loads of 150 lbs. each, spaced as shown (fig. 1); weight of beam 60 lbs. per linear foot. l_2 $l_4 = 3'\text{-}0''$, $l_3 = 9'\text{-}0''$.

	Reaction, left $= W_1$.	Reaction, right $= W_5$.
Beam	$= 30 \times 60 \times \frac{1}{2} = 900$ lbs.	$= 30 \times 60 \times \frac{1}{2} = 900$ lbs.
Load W_2	$= 150 \times \frac{30-3}{30} = 135$ „	$= 150 \times \frac{3}{30} = 15$ „
„ W_3	$= 150 \times \frac{30-9}{30} = 105$ „	$= 150 \times \frac{9}{30} = 45$ „
„ W_4	$= 150 \times \frac{30-27}{30} = 15$ „	$= 150 \times \frac{30-3}{30} = 135$ „
	Total $= 1,155$ lbs.	Total $= 1,095$ lbs.

Shearing stress at section XX, fig. 1, calculated from left :—

Formula (1) $\quad S = W_1 - (W_2 + W_3 + W_6)$
$= 1,155 - [150 + 150 + (60 \times 18)] = -225$ lbs.

Calculated from right $\quad S = W_5 - (W_4 + W_7)$
$= 1,095 - [150 + (60 \times 12)] = 225$ lbs.

Bending moment at section XX, fig. 1, calculated from left :—

Formula (2) $\quad M = W_1 l_1 - (W_2 l_2 + W_3 l_3 + W_6 l_6)$
$= 1,155 \times 18 - [(150 \times 15) + (150 \times 9) + (18 \times 60 \times 9)] = +7,470$ ft. lbs.

and from right, $\quad M = -W_5 l_5 + (W_4 l_4 + W_7 l_7)$
$= -1,095 \times 12 + [(150 \times 9) + (60 \times 12 \times 6)] = -7,470$ ft. lbs.

Graphic diagrams for shear and bending moment values can be drawn so as to represent these values at all points throughout the length of a beam or girder, and such diagrams can always be used when setting out some preliminary or trial design, or checking a design; when carefully and accurately drawn they can be utilised directly in designing all ordinary structures, thus saving a large amount of time and labour which would be necessary if all required values were calculated arithmetically. Since these diagrams show the variations in stress throughout the beam, they can be further utilised to set out the sections or scantlings of the web and flange members, such as the thickness of the web or shear resisting members, and the cut-off in the flange plates or bending resisting members.

The tables of bending moments and shearing stresses given on pages 216 and 217 embrace all simple conditions for concentrated or uniformly distributed loads. Any complex loading occurring in practice can always be resolved into a series of simple loadings; so that diagrams for each simple loading can be drawn on one base line, and these diagrams combined to produce the required diagram for the complex loading, by adding together the several ordinates at any desired section. An example of this is given in fig. 2.

It should be noted that for any series of concentrated loads the bending moment line will form a polygon, the line changing its direction over each of the loaded points; at each of these points the shearing stress is suddenly changed in value, and the shearing stress diagram takes a stepped outline. For a uniformly distributed load, the bending moment line is always a parabolic curve, while the shearing stress changes by a regular gradation.

When beams or girders are of varying depth, the diagrams on pages 216 and 217, which are based on the condition of uniform depth, must be adjusted, for the following reason :—By formula (3) it will be obvious that the flange stresses are dependent on the depth, and, in fact, *vary inversely as this depth*; it is, therefore, necessary to alter the bending moment heights in the *inverse ratio to the alteration in depth of the beam.* Reducing the depth of a beam increases the extreme fibre stress, and necessitates the increasing of the height of the bending moment diagram. Fig. 3 and the brief explanation accompanying it will indicate how this adjustment can be readily carried out.

Where the shearing stress diagram is intended to be used in setting out the varying thickness of the web plate of a beam, the shear stress diagram must similarly be adjusted by altering its heights in *inverse ratio to the alteration in depth of the beam*; a shallower girder necessitates a thicker web, when the shear value remains unaltered (see fig. 3).

When these graphic diagrams on pages 216 and 217 are used in connection with panel or lattice girders, a modification is necessary, since the flange stress remains constant between the points of connection of flange to lattice bars; the diagram becomes stepped in outline and may not be the same in both flanges (see fig. 4).

For the majority of the simple panel or lattice girders met with in practice, the various stresses can be most readily obtained by the reciprocal diagram method or by the method of moments.

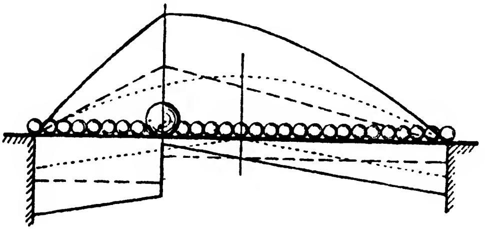


Fig. 2.

This example shows a free-ended beam supporting a uniformly distributed load (see Table 1, type (d)), the bending moment and shearing stress diagrams being indicated by dotted lines; and a single concentrated load (see Table 1, type (e)), the two diagrams being here indicated by broken lines. The combined diagrams for these two loads are shown by full lines, and are got by adding together the two heights at any section, to obtain the new height there.

In some cases, as for girders with fixed ends, negative moments may occur, and these must obviously be added algebraically to the positive moments to obtain the required new combined height.

This figure shows the adjustment of the bending moment and shearing stress diagrams due to a change in the depth of the girder, and a simple graphic method of arriving at this adjustment. The left half of fig. 3 shows diagrams for a free-ended girder of uniform depth carrying a uniformly distributed load (Table 1, type (d)); in the right half of the figure the depth of the girder is reduced as indicated. From the centre to the first section line the depth is uniform, so that the bending moment line remains unaltered, but beyond this section the *height of diagram must be increased in proportion to the reduction in depth of the girder*. This increase can be obtained graphically thus: — Take the vertical section line at point *a*, and draw any line *ab*; on this line mark off *ab* equal to the centre depth of the girder, and *ac* equal to the reduced depth at this section: join *cd* (*d* being at the point of intersection of original diagram line and section line), and draw *be* parallel to *cd* and cutting the section line at *e*. It is obvious that the new vertical heights *ae* and *ad* are proportionate to the original depth and the reduced depth of the girder, or to the lengths *ab* and *ac*, and consequently the adjusted bending moment diagram must pass through the new point *e*. Repeating this operation at a number of other sections will give a series of new points through which the complete adjusted diagram can be drawn.

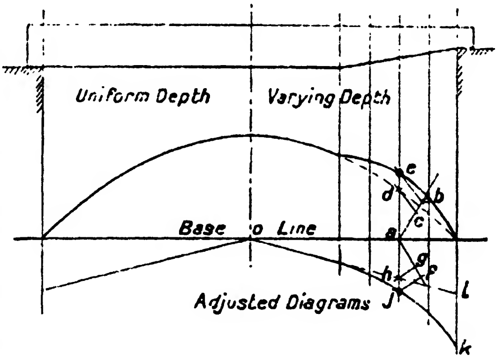


Fig. 3.

Similarly if the shear diagram is to be used for determining the thickness of web plate required, *its depth must first be increased in proportion to the reduction in depth of the girder*, since the sectional area of the web plate is directly proportionate to its depth. Graphically the same operation as above explained can be followed, marking off on the line *af* the lengths *af* and *ag* equal to the centre and reduced depths of the girder, drawing the lines *gh* and *fj* parallel to one another, and deepening the diagram at this section line to the point *j*. The complete adjusted diagram would take the form of the curved line *ojk*.

In determining the stresses on the several members of a lattice girder such as that outlined

in fig. 4, it is assumed that at the junctions between the several members the connections will allow of free movement, as if pin jointed: thus the flanges are assumed as hinged at each junction with the lattices, and no allowance is made for the stiffness due to their being actually formed in one piece.

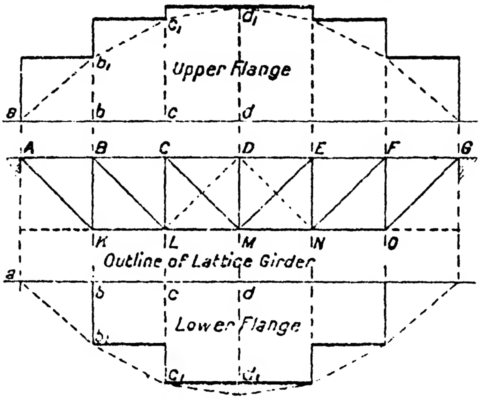


FIG. 4.

This girder rests on abutments at its ends AG, and is loaded at the points BCDEF, say by means of cross girders resting on and fixed to the top flange.

Considering first the load at centre point D, the upper flange does not support it, but the load passes downwards through the vertical strut DM, and is carried by the two inclined ties MC, ME. At the upper ends of these ties the tension due to load at D becomes resolved into compression stress between CE in top flange and compression stress in CL and EN, to which latter stresses must be added a further compression due to loads at points C and E. This transfer of stress is continued to the ends of the girder A and G, the final compression stresses at each end being equal to half the total load, and acting on the bearings at the abutments.

The dotted lines on upper and lower diagrams indicate the bending moment diagrams, and the full lines touching the same show the adjustments necessary owing to the breakage up of flanges and web into a series of members jointed together. The above referred to adjustment is made as follows:—The bending moment diagrams are divided up into sections to correspond with the number of bays in the girder, and as the stress in any section of top or bottom flange must be uniform from joint to joint, the several lines in the diagrams representing the stresses must be parallel to the base line of the diagram throughout the lengths between the joints or junctions between flanges and lattices, all variations in stress taking place suddenly, and being represented by vertical lines in diagram. The stress in the portion AB of upper flange is determined from the moment at point B, and is equal to *bb*; at BC it is equal to *cc*; and at centre portion CDE it is equal to *dd*. Similarly in the lower flange, since the girder is complete without any members extending from K or O to the abutments (as dotted lines), the first stressed bar in lower flange is KL, the amount of stress being determined from height *bb*; the centre portion LMN having its stress determined by the height *cc*; the stress in the two inclined members CM and EM being equal—on the assumption of equal loads at all upper flange joints—must balance one another, and consequently do not affect the lower flange stress at centre.

Influence Value Tables.

It has already been pointed out on page 212 that the several bending moment diagrams given on pages 216 and 217 may be combined to produce one diagram for a series of different loadings.

The same result may be readily obtained, particularly where the value at one particular section is desired, by calculating the bending moment value at that section for each loading and adding the several values together. This process can be much simplified by the use of the tables on pages 218–219, where the value is given of the effect or influence at a series of other points or sections throughout the length of the beam, of a concentrated load applied at certain points. These values are for a load of unit value for a beam of unit length, and for a beam with free ends, with one end fixed, and with both ends fixed; and reactions and inflection points are also noted. The brief notes accompanying the tables will explain their application.

INTERNAL STRESSES.

The tendency of a beam to shear, or to deflect, under any system of loading, is opposed by the resistance of the material of which the beam is composed to shearing, or to tension or compression.

It is the general practice to consider the flanges as resisting the bending moment stresses, the web providing the required shear resistance only.

Considering in the first instance the flange stresses, it is evident, if failure is not to occur, that the moment of resistance at all points in every beam or girder must oppose and balance the bending moment there; this resistance moment is the sum of the moments of all the fibre resistances reckoned about the neutral axis. In a beam of ordinary section the resisting fibres lie at various distances from the neutral axis (owing to the thickness of the flanges); but the simplest case is that of a girder with flanges whose thickness is small in comparison to the depth of the girder, so that without material error all the fibres of one flange may be considered to be practically at one distance from the neutral axis. Thus if d = vertical depth of girder between centres of gravity of the two flanges, M = bending moment due to the external loads, we have the general formula,

$$\pm F = \frac{M}{d} \qquad (3)$$

F being the horizontal stress in either of the two flanges, being compressive stress in the one and tensile strength in the other.

Where the flanges of a girder are of considerable thickness, in comparison with the depth of the girder, or the beam is of a non-symmetrical or irregular cross-section, the fibres of either flange cannot be considered as practically at one distance from the neutral axis, and it is necessary to take into account the varying distances of the fibres from the neutral axis, in the following manner:—

Under the action of any transverse bending, the intensity of the tensile and compressive stresses varies in different fibres of the beam, being greatest in the extreme fibres at the top and the bottom of the cross-section, while the fibre-stress will be zero at the 'neutral axis.'

Whatever may be the figure of the beam's cross-section, the neutral axis will be a horizontal line drawn through its centre of gravity; and the intensity of the stress in any fibre will be proportional to the vertical distance y of the fibre above or below the neutral axis.

If a denotes the sectional area of any thin layer of fibres lying at the height y above the axis, the total stress in that layer will be proportional to ay, and its moment about the neutral axis will be proportional to $ay \times y = ay^2$. The summation of the quantities ay^2 for the different layers is the so-called 'moment of inertia' of the beam, or

$$I = \Sigma\,(ay^2) \qquad (4)$$

When the bending moment, M, due to the external forces has been found in the manner already described, the problem will often be to find the maximum stress per square inch, f, in the extreme layer of fibres; and if y denotes the vertical distance of that layer from the neutral axis, we have

$$f = \frac{My}{I} \qquad (5)$$

Where the section of the beam is symmetrical about the neutral axis, as in rolled joists y = half the vertical depth of the beam between centres of gravity of the two flanges, or $\frac{1}{2}d$, and for such sections the above formula may be written.

$$f = \frac{Md}{2I} \qquad (6)$$

In the case of cast iron, whose strength in compression is much greater than in tension, the tension flange should be made considerably larger than the compression flange. In rolled steel joists the two flanges are made of equal area, and an approximate value for the moment of resistance may be readily found by adding to the actual area of each flange $\frac{1}{6}$th of the area of the web, and then reckoning the depth d as the depth between the centres of gravity of the two flanges; then with A = area of one flange and $\frac{1}{6}$th web, and F = max. safe stress on steel,

$$\text{Approx. resistance moment} = A \times F \times d \qquad (7)$$

When the thickness of the web is small in comparison with the width of the flanges, the extreme tensile strength, f, under the breaking load, as determined by formula (5), corresponds nearly with the ultimate tensile strength of the material; but should the girder have a very substantial web, or should the section be that of a solid rectangle, the calculated breaking load will generally be found to be lower than the actual breaking load. For such cases see formula on page 222.

TABLE I.—BENDING MOMENTS, SHEARING STRESSES, AND DEFLECTION OF STEEL GIRDERS. Graphic Diagrams and Formulæ.

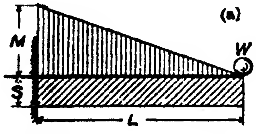
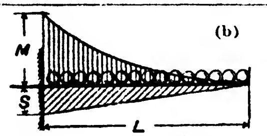
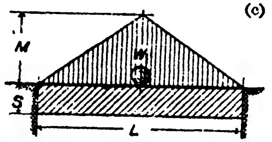
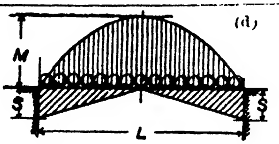
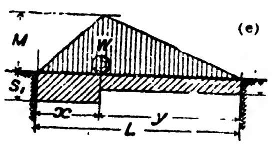
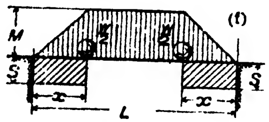
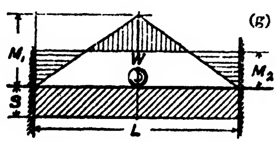
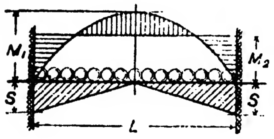
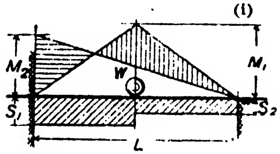
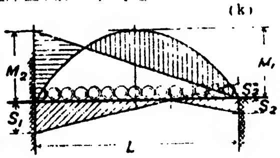
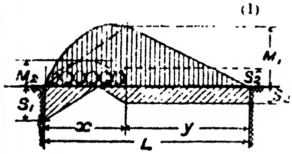
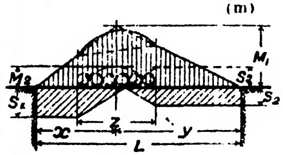
Type.	Formulæ. Bending Moment.	Shearing Stress.	Maximum Deflection.
(a) 	$M = WL$	$S = W$	$\frac{WL^3}{3EI}$
(b) 	$M = \frac{WL}{2}$	$S = W$	$\frac{WL^3}{8EI}$
(c) 	$M = \frac{WL}{4}$	$S = \frac{W}{2}$	$\frac{WL^3}{48EI}$
(d) 	$M = \frac{WL}{8}$	$S = \frac{W}{2}$	$\frac{WL^3}{76{\cdot}8EI}$
(e) 	$M = \frac{Wxy}{L}$	$S_1 = \frac{Wy}{L}$ $S_2 = \frac{Wx}{L}$	$\frac{Wxy(2L-x)}{27EIL} \times \sqrt{3x(2L-x)}$
(f) 	$M = \frac{Wx}{2}$	$S = \frac{W}{2}$	$\frac{Wx(3L^2-4x^2)}{48EI}$
(g) 	$M_1 = \frac{WL}{4}$ $M_2 = \frac{WL}{8}$	$S = \frac{W}{2}$	$\frac{WL^3}{192EI}$

TABLE II.—BENDING MOMENTS, SHEARING STRESSES, AND DEFLECTIONS—*contd.*

Type.	Formulæ.		
	Bending Moment.	Shearing Stress.	Maximum Deflection.
(h) 	$M_1 = \frac{WL}{8}$ $M_2 = \frac{WL}{12}$	$S = \frac{W}{2}$	$\frac{WL^3}{384\ EI}$
(i) 	$M_1 = \frac{WL}{4}$ $M_2 = \frac{3}{16} WL$	$S_1 = \frac{11}{16} W$ $S_2 = \frac{5}{16} W$	$\frac{WL^3}{107{\cdot}3\ EI}$ at a point ·5528 L from Fixed End
(k) 	$M_1 = \frac{WL}{8}$ $M. = \frac{WL}{8}$	$S_1 = \frac{5}{8} W$ $S_2 = \frac{3}{8} W$	$\frac{WL^3}{184{\cdot}6\ EI}$ at a point 5785 L from Fixed End
(l) 	$M_1 = \frac{Wxy}{2L}$ $M_2 = \frac{Wx}{8}$	$S_1 = \frac{W\left(y + \frac{x}{2}\right)}{L}$ $S_2 = \frac{Wx}{2L}$	—
(m) 	$M_1 = \frac{Wxy}{L}$ $M_2 = \frac{Wz}{8}$	$S_1 = \frac{Wy}{L}$ $S_2 = \frac{Wx}{L}$	—

NOTE.—*All curves in these tables are parabolic curves.*

For an example of the combination of bending moments and shearing stress diagrams, and adjustment of same, see page 218.

TABLE III.—INFLUENCE VALUES.

BEAM FREE AT BOTH ENDS.

Load at Section.	Bending Moment Coefficient at Section noted.																			Left end reaction.	Right end reaction.
	·05	·10	·15	·20	·25	·30	·35	·40	·45	·50	·55	·60	·65	·70	·75	·80	·85	·90	·95		
·05	·047	·045	·042	·040	·037	·035	·032	·030	·027	·025	·022	·020	·017	·015	·013	·010	·007	·005	·002	·95	·05
·10	·045	·090	·085	·080	·075	·070	·065	·060	·055	·050	·045	·040	·035	·030	·025	·020	·015	·010	·005	·90	·10
·15	·042	·085	·127	·120	·112	·105	·097	·090	·082	·075	·067	·060	·052	·045	·037	·030	·022	·015	·007	·85	·15
·20	·040	·080	·120	·160	·150	·140	·130	·120	·110	·100	·090	·080	·070	·060	·050	·040	·030	·020	·010	·80	·20
·25	·037	·075	·112	·150	·187	·175	·162	·150	·137	·125	·112	·100	·087	·075	·062	·050	·037	·025	·012	·75	·25
·30	·035	·070	·105	·140	·175	·210	·195	·180	·165	·150	·135	·120	·105	·090	·075	·060	·045	·030	·015	·70	·30
·35	·032	·065	·097	·130	·162	·195	·227	·210	·192	·175	·157	·140	·122	·105	·087	·070	·052	·035	·017	·65	·35
·40	·030	·060	·090	·120	·150	·180	·210	·240	·220	·200	·180	·160	·140	·120	·100	·080	·060	·040	·020	·60	·40
·45	·027	·055	·082	·110	·137	·165	·192	·220	·247	·225	·202	·180	·157	·135	·112	·090	·067	·045	·022	·55	·45
·50	·025	·050	·075	·100	·125	·150	·175	·200	·225	·250	·225	·200	·175	·150	·125	·100	·075	·050	·025	·50	·50
·55	·022	·045	·067	·090	·112	·135	·157	·180	·202	·225	·247	·220	·192	·165	·137	·110	·082	·055	·027	·45	·55
·60	·020	·040	·060	·080	·100	·120	·140	·160	·180	·200	·220	·240	·210	·180	·150	·120	·090	·060	·030	·40	·60
·65	·017	·035	·052	·070	·087	·105	·122	·140	·157	·175	·192	·210	·227	·195	·162	·130	·097	·065	·032	·35	·65
·70	·015	·030	·045	·060	·075	·090	·105	·120	135	·150	·165	·180	·195	·210	·175	·140	·105	·070	·035	·30	·70
·75	·012	·025	·037	·050	·062	·075	·087	·100	·112	·125	·137	·150	·162	·175	·187	·150	·112	·075	·037	·25	·75
·80	·010	·020	·030	·040	·050	·060	·070	·080	·090	·100	·110	·120	·130	·140	·150	·160	·120	·080	·040	·20	·80
·85	·007	·015	·022	·030	·037	·045	·052	·060	·067	·075	·082	·090	·097	·105	·112	·120	·127	·085	·042	·15	·85
·90	·005	·010	·015	020	·025	·030	·035	·040	·045	·050	·055	·060	·065	·070	·075	·080	·085	·090	·045	·10	·90
·95	·002	·005	·007	·010	·013	·015	·017	·020	·022	·025	·027	·030	·032	·035	·037	·040	·042	·045	·047	·05	·95

These tables are calculated for a unit load, and for a girder of unit length; the length of the girder is divided into 20 equal sections and the bending moment values are given at each section point for unit load placed on each section line in turn. For any weight and any length of span it is necessary only to multiply the proper tabular coefficient by the number of units in the load and in the span to obtain the actual

(*Continued on p.* 219.)

TABLE IV.—INFLUENCE VALUE

BEAM FIXED AT RIGHT END.

Load at Section.	Bending Moment Coefficient at Sections noted—(To left of heavy broken line—positive moments; to right—negative). ·05	·10	·15	·20	·25	·30	·35	·40	·45	·50	·55	·60	·65	·70	·75	·80	·85	·90	·95	1·00	Left end reaction.	Right end reaction.	Inflection point.
·05	·046	·042	·039	·035	·031	·027	·024	·020	·016	·012	·009	·005	·001	·002	·006	·010	·014	·017	·021	·025	·925	·075	·333
·10	·042	·085	·078	·070	·063	·055	·048	·040	·033	·025	·018	·010	·003	·005	·012	·020	·027	·034	·042	·049	·851	·149	·331
·15	·039	·078	·116	·105	·094	·083	·072	·061	·050	·038	·027	·016	·005	·007	·018	·030	·041	·054	·066	·073	·777	·223	·328
·20	·035	·070	·106	·141	·126	·111	·096	·082	·067	·052	·037	·023	·008	·007	·022	·037	·051	·066	·081	·096	·704	·296	·324
·25	·032	·063	·095	·127	·158	·140	·122	·103	·085	·066	·048	·030	·011	·007	·025	·044	·062	·081	·099	·117	·633	·367	·319
·30	·028	·056	·085	·113	·141	·169	·147	·125	·103	·082	·060	·038	·016	·006	·027	·049	·071	·093	·114	·136	·564	·436	·313
·35	·025	·050	·075	·099	·124	·149	·174	·148	·123	·098	·073	·048	·025	·002	·028	·053	·078	·103	·128	·153	·497	·503	·305
·40	·022	·043	·065	·086	·108	·129	·151	·173	·144	·116	·087	·059	·031	·002	·026	·054	·083	·111	·140	·168	·432	·568	·296
·45	·018	·037	·055	·074	·092	·111	·129	·148	·167	·135	·104	·072	·041	·009	·020	·053	·085	·116	·148	·179	·371	·629	·285
·50	·016	·031	·047	·063	·078	·094	·109	·125	·141	·156	·122	·087	·053	·019	·016	·050	·084	·119	·153	·187	·313	·687	·273
·55	·013	·026	·039	·052	·065	·077	·090	·103	·116	·129	·142	·105	·068	·031	·006	·043	·080	·118	·155	·192	·258	·742	·258
·60	·010	·021	·031	·042	·052	·062	·073	·083	·094	·104	·114	·125	·085	·045	·006	·034	·074	·113	·152	·192	·208	·792	·243
·65	·008	·016	·024	·033	·041	·049	·057	·065	·073	·082	·090	·098	·106	·064	·022	·020	·062	·104	·146	·188	·162	·838	·224
·70	·006	·012	·018	·024	·030	·036	·042	·049	·055	·061	·067	·073	·079	·085	·041	·003	·047	·090	·135	·178	·122	·878	·203
·75	·004	·009	·013	·017	·021	·026	·030	·034	·039	·043	·047	·052	·056	·060	·064	·019	·027	·072	·118	·164	·086	·914	·179
·80	·003	·006	·008	·011	·014	·017	·020	·022	·025	·028	·031	·034	·036	·039	·042	·045	·003	·050	·097	·144	·056	·944	·153
·85	·002	·003	·005	·006	·008	·010	·011	·013	·014	·016	·018	·019	·021	·022	·024	·026	·027	·021	·070	·118	·032	·968	·122
·90	·001	·001	·002	·003	·004	·004	·005	·006	·006	·007	·008	·009	·009	·010	·011	·012	·012	·013	·036	·085	·015	·985	·087
·95	·000	·000	·001	·001	·001	·001	·001	·001	·002	·002	·002	·002	·002	·003	·003	·003	·003	·003	·003	·046	·004	·996	·046

bending moment value desired. Thus for a load of 2 tons on a girder of 20 feet span the tabular values multiplied by 2 × 20 will give the actual values in foot tons.

Taking the above load and span, and a free-ended girders, with load at ¼th span (or line ·25) the bending moment under the load and at centre of span will be respectively (·187 × 2 × 20) 7·48 and (·125 × 2 × 20) 5 foot-tons respectively. For a girder fixed at one end the corresponding values will be (·158 × 2 × 20) 6·32 and (·066 × 2 × 20) 2·64 foot-tons respectively.

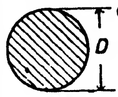
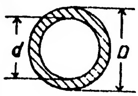
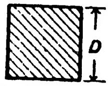
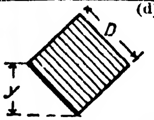
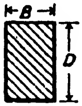
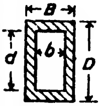
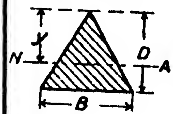
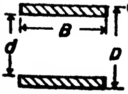
TABLE V.—INFLUENCE VALUES.

BEAM FIXED AT BOTH ENDS.

Load at Section.	Bending Moment Coefficient at Sections noted (outside heavy broken lines—negative moment; inside—positive).																					Left end reaction.	Right end reaction.	Left Inf. pt.	Right Inf. pt.
	0	·05	·10	·15	·20	·25	·30	·35	·40	·45	·50	·55	·60	·65	·70	·75	·80	·85	·90	·95	1·00				
·05	·045	·004	·004	·004	·003	·003	·003	·002	·002	·002	·001	·001	·000	·000	·000	·001	·001	·001	·002	·002	·002	·993	·007	·045	·333
·10	·081	·032	·016	·015	·013	·012	·011	·009	·008	·006	·005	·004	·002	·001	·001	·002	·003	·005	·006	·007	·009	·972	·028	·083	·321
·15	·108	·061	·014	·032	·029	·026	·023	·020	·017	·014	·011	·008	·005	·002	·001	·004	·007	·010	·013	·016	·019	·940	·060	·115	·315
·20	·128	·083	·038	·006	·051	·046	·041	·036	·030	·025	·020	·015	·010	·004	·001	·006	·011	·016	·022	·027	·032	·896	·104	·143	·308
·25	·141	·098	·056	·013	·028	·070	·063	·055	·047	·039	·031	·023	·016	·008	·007	·008	·016	·023	·031	·039	·047	·845	·155	·166	·300
·30	·147	·108	·069	·030	·009	·049	·088	·077	·066	·056	·045	·034	·023	·013	·002	·009	·020	·031	·041	·052	·063	·784	·216	·186	·292
·35	·148	·112	·076	·040	·004	·032	·068	·103	·090	·075	·061	·047	·033	·019	·005	·009	·023	·037	·051	·066	·080	·718	·282	·206	·282
·40	·144	·111	·079	·047	·014	·018	·050	·083	·115	·098	·080	·063	·045	·027	·010	·008	·025	·043	·061	·078	·096	·648	·352	·222	·272
·45	·136	·107	·079	·050	·021	·007	·036	·065	·094	·122	·101	·080	·059	·037	·016	·005	·026	·048	·069	·090	·111	·575	·425	·237	·262
·50	·125	·100	·075	·050	·025	·000	·025	·050	·075	·100	·125	·100	·075	·050	·025	·000	·025	·050	·075	·100	·125	·500	·500	·250	·250
·55	·111	·090	·069	·048	·026	·005	·016	·037	·059	·080	·101	·122	·094	·065	·036	·007	·021	·050	·079	·107	·136	·425	·575	·262	·237
·60	·096	·078	·061	·043	·025	·008	·010	·027	·045	·063	·080	·098	·115	·083	·050	·018	·014	·047	·079	·111	·144	·352	·648	·272	·222
·65	·080	·066	·051	·037	·023	·009	·005	·019	·033	·047	·061	·075	·090	·103	·068	·032	·004	·040	·076	·112	·148	·282	·718	·282	·206
·70	·063	·052	·041	·031	·020	·009	·002	·013	·023	·034	·045	·056	·066	·077	·088	·049	·009	·030	·069	·108	·147	·216	·784	·292	·188
·75	·047	·039	·031	·023	·016	·008	·000	·008	·016	·023	·031	·039	·047	·055	·063	·070	·028	·013	·056	·098	·141	·155	·845	·300	·166
·80	·032	·027	·022	·016	·011	·006	·001	·004	·010	·015	·020	·025	·030	·035	·041	·046	·051	·006	·038	·083	·128	·104	·896	·308	·143
·85	·019	·016	·013	·010	·007	·004	·001	·002	·005	·008	·011	·014	·017	·020	·023	·026	·029	·032	·014	·061	·108	·060	·940	·315	·115
·90	·009	·007	·006	·005	·003	·002	·001	·001	·002	·004	·005	·006	·008	·009	·011	·012	·013	·015	·016	·032	·081	·028	·972	·321	·083
·95	·002	·002	·002	·001	·001	·001	·000	·000	·000	·001	·001	·002	·002	·002	·003	·003	·003	·004	·004	·004	·045	·007	·993	·333	·045

In some cases the values are only approximate, since the tabular values give 3 decimal figures only: for example, in the first calculation the full tabular value would be ·1875, and the accurate result 7·5 foot-tons. The values for a load distributed over a part or the whole of the span may be obtained approximately by assuming the total load as formed of a number of equal loads placed at adjoining section points; the result here would generally be near enough to the true value for all practical purposes.

TABLE VI.—MOMENTS OF INERTIA, SECTIONAL AREA, ETC., FOR VARIOUS SIMPLE SECTIONS.

Section.	Area.	Neutral Axis.	Moment of Inertia.
(a) 	$\frac{\pi D^2}{4}$ $\pi = 3{\cdot}1416$	Through centre	$\frac{\pi D^4}{64}$
(b) 	$\frac{\pi(D^2 - d^2)}{4}$	Through centre	$\frac{\pi(D^4 - d^4)}{64}$
(c) 	D^2	Through centre	$\frac{D^4}{12}$
(d) 	D^2	Through centre $y = \frac{D}{1{\cdot}4142}$	$\frac{D^4}{12}$
(e) 	BD	Through centre	$\frac{BD^3}{12}$
(f) 	$BD - bd$	Through centre, parallel to B Through centre, parallel to D	$\frac{BD^3 - bd^3}{12}$ $\frac{DB^3 - db^3}{12}$
	$\frac{BD}{2}$	$y = \frac{2}{3}D$ Through Base Line, through Apex, and parallel to Base	$\frac{BD^3}{36}$ $\frac{BD^3}{12}$ $\frac{BD^3}{4}$
(h) 	$B(D - d)$	Through centre and parallel to B	$\frac{B(D^3 - d^3)}{12}$

The value for the moment of inertia for the standard steel sections of rolled joists, angles, tees, etc., can be got from British Standards Nos. 4 and 4A.

The value for the moment of inertia about any other axis parallel to and at a distance y. from the neutral axis = I for neutral axis + (area × y^2) . . . . . . (8)

The value for the radius of gyration is $R = \sqrt{\frac{I}{\text{Area}}}$ . . . . (9)

Solid Rectangular Beams.

Where the strength can be shown to vary directly as the breadth of the beam and as the square of the depth, it is better, simpler, and more direct to calculate the strength by means of coefficients derived from direct experiments in cross-breaking; thus

$$\text{Breaking load in tons} = \frac{\text{Breadth in inches} \times \text{depth in inches}^2}{\text{Span in inches}} \times c \times k \qquad (10)$$

TABLE VII.

Material.	Coeff. c.	Ends of Beams.	Coeff. k.
Mild Steel	3·50	One fixed, other end loaded	1
Cast Iron	2·30	,, ,, load distributed	1
Fir (Spruce)	0·60	Both supported, load in centre	4
English Oak	0·75	,, ,, ,, distributed	8
Red Pine	0·65	Both fixed, ,, in centre	6
Yellow Pine	0·50	,, ,, ,, distributed	12
Pitch Pine	0·75		

The factor of safety for steel may be ¼th to ⅙th breaking load; for the other materials it may vary between ⅛th and 1/10th breaking load. If the beam carries a heavy live load the safe load should be further reduced.

TABLE VIII.—BENDING STRESSES.

In tons per sq. in.

British Standards 449—1948, for the Use of Structural Steel in Building.*

Mild Steel B.S. 15.	High Tensile Steel. B.S. 548 Up to 1½ in. thick.	B.S. 548 Over 1½ in. thick.	B.S. 968. Up to 1 in. thick.	B.S. 968. Over 1 in. thick.	Other Steels.
In tension: Beams					
10	15	13	13·5	12·25	$0{\cdot}65\,f_y$†
	Plate girders				
10	14·25	12·25	12·75	11·5	$0{\cdot}65\,f_y$†

In compression: the above values or $1{,}000K_1 \div l/r$ tons per sq. in., whichever is the lesser, where l is the length between effective lateral restraints, r is the radius of gyration of the section perpendicular to the plane of bending, and K_1 is a factor taken as unity except for rolled steel joists, compounds and plate girders, symmetrical about both principal axes and subject to bending about xx axis. For these the value of K_1 shall be :—

r_{xx}/r_{yy}	5·0	4·5	4·0	3·5	3·0 or less
K_1	1	1·125	1·25	1·375	1·5

There are special clauses dealing with cased beams and encased filler beams.

SHEAR STRESS IN WEBS.

The average sheer stress on the gross cross section of the web shall not exceed the lesser of (1) and (2) below for unstiffened webs, and of (1) and (3) below for stiffened webs.

(1) 6·5 tons per sq. in. for steel to British Standard 15.
9 tons per sq. in. for steel to British Standard 548 up to and including 1½ in. thick.
8 tons per sq. in. for steel to British Standard 548 over 1½ in. thick.
9 tons per sq. in. for steel to British Standard 968 up to and including 1 in. thick.
8 tons per sq. in. for steel to British Standard 968 over 1 in. thick.
$0{\cdot}42\,f_y$ for other steels, where fy is the guaranteed yield stress in tons per sq. in.

* By permission of the British Standards Institution.

† f_y is the guaranteed yield stress in tons per sq. in.

(2) $\left(\frac{225}{d/t}\right)^2$ tons per sq. in. for unstiffened webs, where d is the clear distance between flange angles, or where there are no flange angles, the clear distance between flanges, and t is the thickness of the web.

(3) $\left(\frac{225}{b/t}\right)^2\left[1+\frac{3}{4}\left(\frac{b}{a}\right)^2\right]$ tons per sq. in. for effectively stiffened webs, where a is the greater unsupported dimension of web in a panel, b is the least unsupported dimension of web in a panel and t is the thickness of the web.

PERMISSIBLE STRESSES IN RIVETS AND BOLTS.

In tons per sq. in.

	Steel to B.S. 15.	H.T. Steel to B.S. 548.
In tension—		
Axial stress on gross area of rivets and net area of bolts:		
Shop-driven rivets	5	7·5
Field-driven rivets	4	6
Bolts $\frac{3}{4}$ in. and over in diameter	6	9
Bolts less than $\frac{3}{4}$ in. diameter	5	7·5
In shear—		
Shear-stress on grass area of rivets and bolts:		
Shop-driven rivets	6	9
Field-driven rivets	5	7·5
Turned and fitted bolts	6	9
Black bolts	4	6
In bearing—		
Bearing stress on gross diameter of rivets and bolts:		
Shop-driven rivets	12	18
Field-driven rivets	10	15
Turned and fitted bolts	12	18
Black bolts	8	12

Permissible bearing stress for high tensile steel to B.S. 968: 18 tons per sq. in.

DEFLECTION OF BEAMS.

Deflection formulæ for various arrangements of loading are given in Tables I. and II., pp. 216 and 217. For structural steel the value of E (the modulus of elasticity) may be taken at 30,000,000 lbs. and the value of W must then be expressed in lbs. also.

CONTINUOUS BEAMS OR GIRDERS.

When a beam is carried upon more than two supports the reactions of the several supports cannot be directly determined, and, in consequence, the bending stresses can only be found by reference to the changes of form which must always accompany those stresses, and which are fixed or limited at certain points by the conditions of the structure—conditions which vary in each case with the width and the number of spans, as well as with the disposition of the load.

When the beams are of uniform section, such as rolled steel joists or rectangular timber beams, the strength or load which may be safely carried will depend on the greatest bending moment of *either kind* which occurs throughout the whole length of the bridge, and the advantage of a continuous beam to cover a number of spans will depend on the number and spacing of the supports. In cases where all the spans are of uniform length, and the load is uniformly distributed throughout, the maximum bending moment values for varying numbers of spans can be found in many text-books.

For the General Problem, however, where the spans are of different lengths and the loads vary, the determination of the stresses in continuous beams is a very complex one. The easiest solution is to be found in the Graphic Method, generally known as the 'Method of Characteristic Points,' described by Professor Fidler in a paper read before the Inst. C.E. (*Proc.*, vol. lxxiv.), and treated in detail in Fidler's 'Bridge Construction.'

Before proceeding to determine graphically the stresses in a continuous beam, it is necessary to find the Characteristic Points, and the following examples will illustrate how these points can be fixed for any bending moment diagram for a free-ended single span beam.

In fig. 5, AB represents the span, and the parabola ACB the bending moment diagram due to a uniformly distributed load. Divide the span into three equal parts, AE, EF, and FB, and at points E and F erect perpendiculars EJ and FK. The height of the points J and K (which lie on the line GH) above the base line is fixed by the condition that *the moment of the area ACB about one end of the span shall equal the moment of the rectangle AGHB about the same point*. This

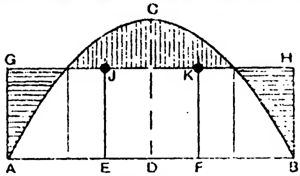


FIG. 5.

diagram being symmetrical about the vertical centre line, the centre of gravity of these two areas will fall on this centre line, and the moment arm for each area will be equal to ½ AB, or one-half span.

In this case of the *parabolic B.M. diagram*.

$$\text{Area } ACB = \tfrac{2}{3}\, CD \times AB, \text{ and area } AGHB = AG \times AB$$
$$\therefore EJ = FK = \tfrac{2}{3}\, CD.$$

The Characteristic Points for a uniformly distributed load, therefore, occur at a height equal to ⅔ the central height of the parabola.

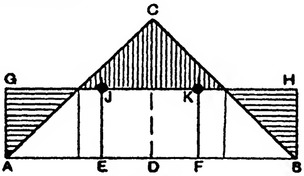


FIG. 6.

Fig. 6 shows the diagram for a *concentrated central load*, represented by the B.M. diagram ACB, and by the same reasoning as for a distributed load, the height EJ and FK will now be equal to ½ the central height CD.

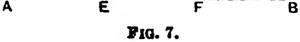


FIG. 7.

Fig. 7 shows another symmetrical case where two equal concentrated loads are placed at the same distance from the centre line; here again the same reasoning as before is adopted. Assuming a span AB = 20 ft., with the two equal loads of 6 tons at 4 ft. from ends A and B. BM = 6 × 4 = 24 ft.-tons = scale height for CL and DM. Then moment of area ACDB about A = AM × MD = 16 × 24 = 384, and height of an equal rectangle AGBH = $\frac{384}{20}$ = 19·2 = EJ and FK.

Where the *B.M. diagram is not symmetrical about the centre vertical line*, the two Characteristic Points, although still situated on the two perpendicular lines at ⅓ AB from each end, *are not now at the same height above the base line*, since the moment of a non-symmetrical B.M. diagram will not be the same about the ends A and B.

For example, fig. 8 shows a beam, say, 30 ft. long with a load at D of, say, 10 tons placed at a distance of 8 ft. from B. The upward reaction at end A = $\frac{8}{30}$ of 8 tons = 2·13 tons, and the height DC of B.M. diagram ACB = 2·13 × 22 = 46·9 ft.-tons. Draw a line from the centre point

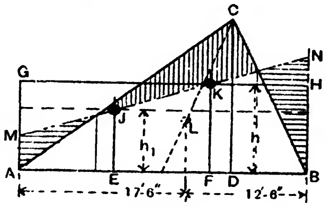


Fig. 8.

of AB to O, and set off OL = ⅓ of the total length, then L is the centre of gravity of the B.M. diagram AOB. The moment of the area AOB will be greater about end A, thus h (= height of rectangle to balance the greater moment) will be found thus:—

$$\text{Area AOB} \times 17\cdot5 = h \times 30 \times 15.$$

or, $$\frac{30 \times 46\cdot9}{2} \times 17\cdot5 = h \times 450$$

$$\therefore\ h = \frac{704 \times 17\cdot5}{450} = 27\cdot4,$$

and similarly about B,

$$\text{Area AOB} \times 12\cdot5 = h \times 30 \times 15;$$

$$\therefore\ h = 19\cdot6.$$

These values, therefore, give the relative heights of the Characteristic Points K and J.

This method of locating the Characteristic Point is applicable for any condition of loading, as it is only necessary to set up the combined B.M. diagram for any desired combination of loads, calculate the area, and the positive of the centre of gravity of this combined diagram, and proceed as indicated to calculate the height of the Characteristic Points, noting that two calculations will be required if the B.M. diagram is not symmetrical about the vertical centre line.

In the case of any Bending Moment Diagram for a simple free-ended beam, the bending stress at any point in its length is proportionate to the vertical height at the required point, measured from the original base or datum line AB, *and Characteristic Points are unnecessary*. Should the beam be rigidly fixed at the ends, use must be made of the new base or datum line found by drawing a line through the Characteristic Points and scaling the required bending moment values from this new line (marked GH in figs. 5–7, and MN in fig. 8). The positive B.M. stresses (in the middle portions of the beam) are measured *upwards* from the new base line to that of the B.M. diagram, while the *negative* stresses are measured downwards. The B.M. is zero at the contraflexure points, where the bending stresses change from positive to negative. Thus theoretically each span consists of two end cantilevers supporting a hinged or free-ended beam, it being obvious that at the points where the bending *directions* change, the bending stress must be zero. (*Note.*—The bending stress in the end portions are generally termed negative stresses and represented thus −, while the centre portions are positive stresses and represented thus +. In the diagram, negative stressed portions are indicated by horizontal line shading, and positive stressed portions by vertical shading.)

Bending Moment for Continuous Girders.

If the rigid fixing at either end of a beam is altered or interfered with, the alteration in bending stresses which then follows throughout the length of the beam can be represented by an alteration in the position of the straight base line, so that it would pass above or below the Characteristic Points. When a continuous length of beam is carried over two or more spans, this interference occurs at the points of support between the several spans, and Fidler's method enables the effect of this interference to be graphically represented in the following manner: see fig. 9, which represents a four-span continuous bridge carrying a distributed load; the B.M. diagrams in this case take the form of a parabola. For any other form of loading the suitable diagram must be used.

Calculate the bending moment for each span separately as for a free-ended beam, draw the corresponding diagram for each span on the base line AB, and set off the Characteristic Points, as previously explained, marking these points 1 to 8 as shown. If the outer ends of end spans,

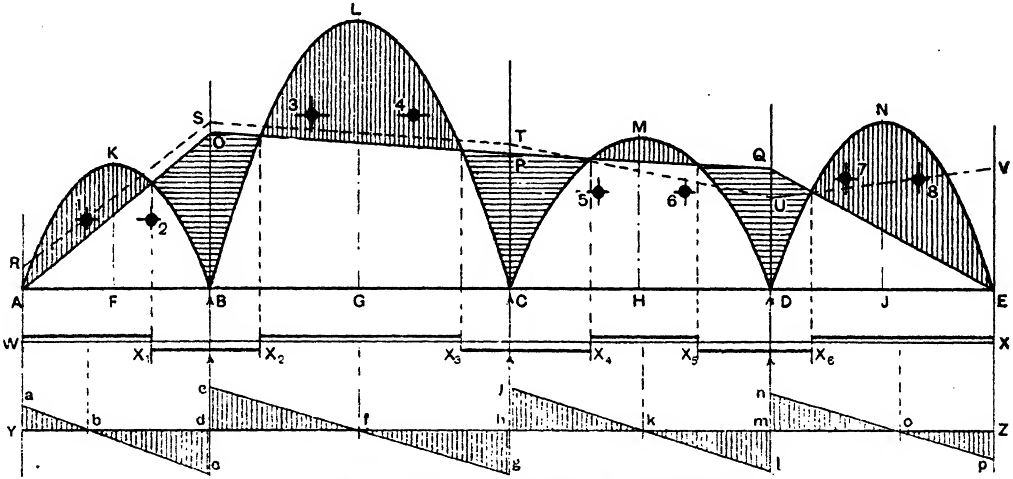


FIG. 9.—Four-Span Continuous Bridge carrying a Distributed Load.

If this bridge is designed so that the outer ends of the end spans are rigidly fixed, the new base line must pass through the two extreme Characteristic Points 1 and 8, and the new line will take the form indicated by the line RSTUV, the heights AR and EV indicating the maximum amount of negative bending moment at the extreme end points.

These diagrams should be drawn out to as large a scale as possible, and it will facilitate the fixing of the irregular base or datum line if tracing paper is first placed over the bending moment layout, and the necessary trial lines pencilled on same, the correct setting out, when found, being pricked through to the diagram below.

THERE IS ONLY ONE SERIES OF BROKEN LINES WHICH WILL FULFIL THE CONDITIONS REQUIRED IN ANY ONE CASE, and some difficulty may at first be experienced in drawing this line.

at A and E, rest freely on the supports, the two end Characteristic Points 1 and 8 are disregarded. The problem now is to draw a series of straight lines, AO, OP, PQ, and QE, but *these lines must be so placed that they pass above and below, or below and above the Characteristic Points on either side of the vertical lines over the supports B, C, and D, at distances inversely proportionate to the adjoining spans*; for instance, in fig. 9 the distance of the line AO above point 2 should be greater than that of the line OP under point 3 in the proportion of the lengths BO to AB. These conditions are carried to the end, points 4 and 5 being at proportionate distances above and below, and points 6 and 7 below and above the broken line, which finally must end at E. This broken line, AOPQE, will now represent the new base line from which the positive or negative bending moment values for any point in the bridge can be scaled off. As in the simpler diagrams previously given, the portions subjected to positive bending are shown by vertical shading, and negative bending by horizontal shading.

The sketch diagram WX (p. 226) indicates graphically the system or combination of simple beams and cantilevers into which a continuous beam over several supports may be divided.

The sketch diagram YZ gives a graphic representation of the shears and reactions: the reaction is always equal to the shear. The shear at the points of maximum bending becomes zero, or passes from positive (+) above the base line, to negative (—) below the base line at these points.

The shear and reaction can be readily calculated from a consideration of diagram WX.

Shear at A (+) = reaction as from a free-ended beam WX; this value is set up on line YZ, at Y*a*, shear left of B (—) = total load on span less left reaction (set off *dc* on line YZ). Shear on right of B (+) = load on cantilever X_1, X_2, plus half-load on suspended span X2, X3 (set up *de* on line YZ). Reaction on support B = sum of above (—) and (+) shears. By the same reasoning, the remaining shears and reactions can be determined. Draw lines connecting the points *ac*, *eg*, *jl*, and *np*, to complete the shear diagram; and note that the shear zero points *b*, *f*, *k*, and *o*, should occur at the centres of the imaginary beams in diagram WX.

COLUMNS AND STRUTS.

BRITISH STANDARD 449—1948, FOR THE USE OF STRUCTURAL STEEL IN BUILDING.*

Effective Length of Struts.

Effective Length.—The effective length (*l*) of a compression member for the purpose of determining allowable axial stress shall be completed as follows:—

Where both ends are held in position and restrained in direction, 0·7 of the actual length.

Where both ends are held in position and one end restrained in direction, 0·85 of the actual length.

Where both ends are held in position but unrestrained in direction, the actual length.

Where one end is held in position and restrained in direction and the other end is partially restrained in direction but not held in position, 1·5 times the actual length.

Where one end is effectively held in position and restrained in direction and the other end is not held in position or restrained in direction, twice the actual length.

The effective lengths of stanchions relative to end and beam connections are illustrated in fifteen pages of drawings, the effective length and slenderness ratio for design being indicated in each case.

Maximum Slenderness Ratio of Struts.

The ratio of effective length to the appropriate radius of gyration shall not exceed the following values:

(*a*) For any member carrying loads resulting from dead weights and superimposed loads: 180.

(*b*) For any member carrying loads resulting from wind forces only, provided the deformation of such member does not adversely affect the stress in any part of the structure: 250.

(*c*) For any member, normally acting as a tie in a roof truss but subject to possible reversal of stress resulting from the action of wind: 350.

There are separate tables for the permissible working stresses for discontinuous angle struts, both for single bolted or single riveted end connections and for double bolted, double riveted or welded end connections.

There are special clauses, also, relating to cased struts.

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* By permission of the British Standards Institution.

TABLE IX.

PERMISSIBLE WORKING STRESSES FOR AXIAL LOADS ON STRUTS, IN TONS PER SQ. IN. OF GROSS SECTION.

$\frac{l}{r}$	Mild Steel B.S. 15.	H.T. Steel to B.S. 548. Up to 1½ in.	H.T. Steel to B.S. 548. Over 1½ in.	H.T. Steel to B.S. 968. Up to 1 in.	H.T. Steel to B.S. 968. Over 1 in.
0	9·00	13·5	11·50	12·50	11·06
10	8·51	12·65	10·83	11·73	10·37
20	8·03	11·80	10·17	10·69	9·74
30	7·54	10·94	9·50	10·19	9·11
40	7·06	10·09	8·83	9·42	8·48
50	6·57	9·24	8·16	8·65	7·85
60	6·09	8·39	7·50	7·89	7·22
70	5·60	7·54	6·83	7·12	6·59
80	5·12	6·68	6·16	6·35	5·96
90	4·62	5·77	5·10	5·54	5·26
100	4·13	4·96	4·71	4·80	4·60
110	3·67	4·27	4·09	4·16	4·02
120	3·26	3·70	3·57	3·61	3·51
130	2·89	3·22	3·12	3·16	3·08
140	2·57	2·83	2·75	2·78	2·72
150	2·30	2·50	2·44	2·46	2·41
160	2·06	2·22	2·17	2·19	2·15
170	1·86	1·98	1·95	1·96	1·93
180	1·68	1·78	1·75	1·76	1·74
190	1·52	1·61	1·59	1·60	1·58
200	1·30	1·46	1·44	1·45	1·43
210	1·27	1·33	1·32	1·32	1·31
220	1·17	1·22	1·21	1·21	1·20
230	1·08	1·12	1·11	1·11	1·10
240	0·99	1·03	1·02	1·03	1·02
250	0·92	0·96	0·95	0·95	0·94
300	0·65	0·67	0·67	0·67	0·66
350	0·49	0·50	0·50	0·50	0·49

ECCENTRICITY FOR STANCHIONS AND SOLID COLUMNS.

The assumed eccentricity of loads on stanchions and solid columns shall be:

(i) At the mid point of stiffened seating of a stiffened bracket.
(ii) At the outer face of vertical leg of an unstiffened bracket.
(iii) At face of strut in the case of cleat connections to web of beam.
(iv) *a.* At the mid point of cap bearing continuous beams of approximately equal span and load.
b. At the edge of stanchion towards span of beam in the other cases, excepting roof truss bearings.
c. No eccentricity for simple roof truss bearings without connections capable of developing an appreciable moment.

Axial Stresses in Tension.

The direct stress in pure tension on the net area of the section in tons per sq. in. shall not exceed the following values:

Mild Steel to B.S. 15.	H.T. Steel to B.S. 548. Up to 1½ in. in Thickness or Diam.	H.T. Steel to B.S. 548. Over 1½ in.	H.T. Steel to B.S. 968. Up to 1 in. in Thickness or Diam.	H.T. Steel to B.S. 968. Over 1 in.	Other Steels.
9	13·5	11·5	12·5	11	$0{\cdot}59 f_y$

where f_y is the guaranteed yield stress in tons per sq. in.

Combined Stresses.

Bending and axial compression.—Members subject to both axial compression and bending stresses shall be so proportioned that the quantity

$$\frac{f_a}{F_a} + \frac{f_{bc}}{F_{bc}}$$

does not exceed unity,

where f_a = the axial compressive stress.
F_a = the permissible compressive stress in axially loaded struts.
f_{bc} = the sum of the compressive stresses due to bending about both rectangular axes.
F_{bc} = the minimum permissible compressive stress for members subject to bending.

Bending and axial tension.—Members subject to both axial tension and bending stress shall be so proportioned that the quantity

$$\frac{f_t}{F_t} + \frac{f_{bt}}{F_{bt}}$$

does not exceed unity,

where f_t = the axial tensile stress.
F_t = the permissible axial tensile stress.
f_{bt} = the maximum tensile stress due to bending about both principal axes.
F_{bt} = the permissible tensile stress in bending.

Permissible Stresses in Welds.

There are special clauses relating to the permissible stresses in welds for mild steel and for high tensile steel.

Stresses due to Wind Forces.

The working loads and stresses per sq. in. specified for beams, pillars, and all their connections in B.S. 449, as computed for all loads and forces other than wind pressure may be increased by 25 per cent. in cases where such increase is solely due to stresses induced by wind pressure.

TABLE X.—CAST-IRON COLUMNS.

Maximum Safe Stresses per sq. in. of Section. (London Building Act Amendment, 1909.)

$\frac{L}{R}$	Hinged ends.	Hinged and fixed.	Fixed ends.	$\frac{L}{R}$	Hinged ends.	Hinged and fixed.	Fixed ends.	$\frac{L}{R}$	Hinged ends.	Hinged and fixed.	Fixed ends
2	4·4	4·9	5·4	32	2·9	3·4	3·9	62	1·4	1·9	2·4
4	4·3	4·8	5·3	34	2·8	3·3	3·8	64	1·3	1·8	2·3
6	4·2	4·7	5·2	36	2·7	3·2	3·7	66	1·2	1·7	2·2
8	4·1	4·6	5·1	38	2·6	3·1	3·6	68	1·1	1·6	2·1
10	4·0	4·5	5·0	40	2·5	3·0	3·5	70	1·0	1·5	2·0
12	3·9	4·4	4·9	42	2·4	2·9	3·4	72	0·9	1·4	1·9
14	3·8	4·3	4·8	44	2·3	2·8	3·3	74	0·8	1·3	1·8
16	3·7	4·2	4·7	46	2·2	2·7	3·2	76	0·7	1·2	1·7
18	3·6	4·1	4·6	48	2·1	2·6	3·1	78	0·6	1·1	1·6
20	3·5	4·0	4·5	50	2·0	2·5	3·0	80	0·5	1·0	1·5
22	3·4	3·9	4·4	52	1·9	2·4	2·9	82	0·4	0·9	1·4
24	3·3	3·8	4·3	54	1·8	2·3	2·8	84	0·3	0·8	1·3
26	3·2	3·7	4·2	56	1·7	2·2	2·7	86	0·2	0·7	1·2
28	3·1	3·6	4·1	58	1·6	2·1	2·6	88	0·1	0·6	1·1
30	3·0	3·5	4·0	60	1·5	2·0	2·5	90	0·0	0·5	1·0

Riveted Joints in Girderwork.

The riveted joint of a tension-member may fail either by the tearing of the plate, or the shearing of the rivets, or in some special cases by the crippling of the bearing area between plate and shank of rivet.

The tearing of the plates is resisted by the tensile strength of the plate per square inch of area taken over the *net effective section* between rivet holes. When the assumed line of fracture is at right angles to the direction of the pull, the sectional area measured on this line between the rivet-holes is the net effective section. But when the line of fracture may follow a zigzag or inclined direction, the effective sectional area should be taken at about three-fourths of the area measured on such a line. The fraction is not universally applicable, but will generally meet the case of ordinary zigzag riveting.

The shearing of the rivets is resisted by the shearing strength of the rivet-material per square inch of area taken over the effective section of the rivets engaged. Assuming all the rivets to take an equal share in the work, the number of rivets engaged in resisting the total pull upon the tie will be the whole number of rivets in the case of a lap-joint such as that shown in figs. 13 and 14; while in the case of a butt joint, such as No. 2 of fig. 15, it will consist only of the rivets employed on either side of the joint. But in any large assemblage of rivets allowance should be made for unequal distribution of the work when calculating the number of rivets required for a joint.

For rivets in 'single shear,' as in the case of lap joints, and in butt joints covered by a single cover-plate, the effective section of each rivet is simply the area of its cross-section. But wherever practicable double cover-plates should be employed, placing the rivets in 'double shear,' and in this case the effective section of each rivet should be reckoned at $1\frac{3}{4}$ times the sectional area of the rivet.

In wrought-iron work the shearing strength of good rivet iron per square inch is fully equal to the tensile strength of the plates; and the effective rivet section is often made equal to the effective plate section, an allowance of 10 per cent. being sometimes added in the case of large joints.

The shearing strength of steel rivets may be taken at 24 tons per square inch, while the plates have often a tensile strength of 26 to 32 tons; and the rivet section will generally require to be somewhat greater than the plate section.

The 'efficiency' of a riveted joint is the ratio between its strength at the weakest part and the strength of the unperforated plate.

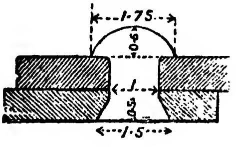
RIVETS AND RIVETING.

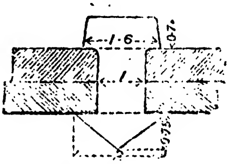
Rivets.

The heads (fig. 10) are usually either snap or pan. Snap heads are used generally in machine riveting. The points are countersunk, conical, or snap; the former are largely employed in iron shipbuilding, where a flush surface is required, the latter in ordinary boiler and bridge work. Conical, also called *staff* or *hammered*, points are chiefly used where the restricted space only allows of hammering.

The following figures give ordinary proportions of rivet heads in terms of diameter of shank.

SNAP HEAD. PAN HEAD.





Countersunk point. FIG. 10. Conical point.

If d = diameter of shank of rivet,

Length allowed for forming pan head				= 2 to $2\frac{1}{4}d$;
"	"	"	snap head	= $1\frac{1}{2}d$.
"	"	"	countersunk head	= $1d$.
"	"	"	conical head	= $1\frac{1}{4}d$.

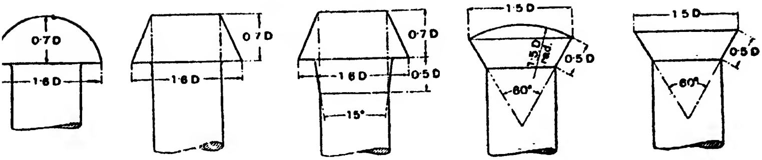
When a conical form is given under the rivet head an additional length of about $\frac{1}{8}''$ is usually allowed in making the rivet.

BRITISH STANDARD SPECIFICATION FOR DIMENSIONS OF RIVETS.*

(½ Inch to 1¾ Inch Diameter.)

(No. 275—1927.) (*Abstract.*)

(This Specification does not apply to Boiler Rivets.)



Snap Head. Pan Head. Pan Head with Tapered Neck. Rounded Countersunk Head. Flat Countersunk Head.

FIG. 11.

D = Nominal diameter of rivet.

Diameter of Rivets or Rivet Holes.

d = diameter of rivet hole ; t = thickness of thickest plate.

Fairbairn's Rule, commonly used { When t is less than ½", $d = 2t$. When t is equal to, or greater than, ½", $d = 1\frac{1}{2}t$

With rivets so proportioned the total number in single shear, such as lap, single cover and grouped, joints will, when the resistance to shearing is taken at 4 tons and bearing at 5 tons, depend on the number required for shearing, in ½" plates and over ½", and for bearing with plates under ½" thick.

When more than 2 plates, besides the covers, are riveted together, as in flanges of girders, some engineers add $\frac{1}{16}$ to d for each extra plate.

Unwin's Rule for Rivets : $d = 1 \cdot 2\sqrt{t}$.

Rivets of more than 1½" diameter are rarely used.

The rivet holes are made about $\frac{1}{16}$" larger than the rivets, to allow for expansion when heated. For ordinary work the proportions of diameter of punch to that of bolster hole in the die varies from 1 : 1·15 to 1 : 1·2 ; the greater the difference, the less cleanly cut will the burr be, but the metal round the hole will be stronger, being less reduced and more compressed. The hole when punched is slightly conical.

It is hardly practicable to temper a punch so as to punch a great number of holes if it is of less diameter than the thickness of the plate, or even with a diameter of equal thickness. Punched holes should never be less than 1½ diameter apart or from edge to plate. For drilled holes the limit may be 1 diameter.

Pitch of rivets = distance from centre to centre of rivet. *Longitudinal pitch* = pitch in direction of stress. *Transverse pitch* = pitch at right angles to stress. *Zigzag pitch* = minimum pitch when placed zig-zag ; this should slightly exceed the transverse pitch. *A pitch line* is a straight line running through the centres of a longitudinal or transverse row.

Minimum pitch = 2½ diameters for punched or 2 for drilled holes.

Maximum longitudinal pitch in compression plates = 12 times thickness of weakest outside plate.

* By permission of the British Standards Institution.

Strength of Rivets and Plates.

Mean tensile breaking strength of best iron, such as BBB Staffordshire, best Yorkshire or Lowmoor (and the best only should be used), is from 26·33 tons per square inch (Fairbairn) to 25·98 tons per square inch (Kirkaldy); while the contraction of fractured area should not be less than 40 per cent.

Forge Tests for Rivets.—When cold they should stand being hammered double, or being upset endways, a ¾" rivet 2¼" long being hammered down to a cylinder 1" long, without any sign of cracking. A rivet ready heated for use should now and then be taken, allowed to cool, and treated as above, to see if they are being overheated. Much overheating is detected by the burnt, scaly appearance of the surface of shank; ¾" or even 1" rivets should not be over 20 minutes in the fire. When *hot* the heads should stand hammering down to less than ¼" thick without cracking at edges; and the stalks should stand having a punch, even more than their own diameter, driven through them without cracking round the hole.

The foregoing tests are equally applicable to steel rivets.

Careful punching does not injure soft plates, but weakens hard or steely iron plates from 15 to 30 per cent.

Sharp edges of drilled holes reduce the shearing resistance of rivets, but rounding the edges of the holes increases resistance of rivet.

Drilled rivet holes reduce the strength of a plate by 1¼ per cent., and punched holes about 6 per cent. Punching and counter-sinking reduce the strength 4 per cent., but if the holes are punched, reamered, and countersunk, there is little reduction in the strength of the plate. Comparative tests of high tensile plates, connected by iron and by high tensile rivets, show that ½-in. iron rivets worked cold in single shear have a shearing strength of 3·6 tons compared with 6·6 in the case of high-tensile steel worked cold, and 9·6 in case of the same steel worked hot. For double shear the same figures are 6·6, 13·1, and 18·4. Experiments show that the bearing pressure with high tensile steel rivets does not exceed 60 lbs. per square inch.

(*Sir P. Watts.*)

Riveting.

Riveting gangs consist of a holder-up, two riveters, and one or two boys for heating and supplying rivets. Rivets are heated in a portable hearth or a reverberatory furnace. Before inserting the rivet the plates round the hole should be hammered into positive contact, and the holes ascertained to be concentric.

Riveting hammers vary from 2 to 7 lbs., the holding-up hammer or 'dolly' from 10 to 40 lbs. 120 to 150 rivets is a day's work of 13 hours of a riveting gang. 360 to 720 can be put in by a riveting machine;

Riveted Joints.

The loss of strength is ordinarily, for chain-riveted joints (fig. 12), 15 per cent.; for double-riveted, 30 per cent.; for single-riveted joints, 44 per cent., as compared with the strength of the plate unpunched.

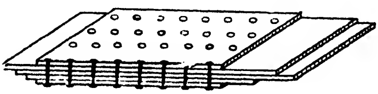


FIG. 12. Grouped Butt Joint. Chain Riveting.

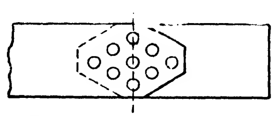


FIG. 13. Lap Joint. Lozenge Riveting.

Zigzag Riveting.—In this (fig. 14) the distance between the transverse pitch lines should be at least two-thirds the transverse pitch of the rivets.

Lozenge Riveting.—Arranging the rivets in this way, as illustrated by the lap-joint of fig. 13, and calculating the plate strength and rivet strength at each of the possible lines of fracture, it is possible to obtain a strength of joint equal to the strength of the whole plate less one rivet hole. With the same efficiency a still better joint for a plate-tie may be made with two cover-plates of lozenge form, the maximum row of rivets being repeated on each side of the butt joint.

Cover-Plates.—In the special case last mentioned the united thickness of the cover-plates will have to be considerably greater than that of the main tie, and will be determined by the net section of cover taken through the maximum row of rivets. In chain riveting generally it would be theoretically sufficient to make the united cover thickness equal to the plate thickness, but in practice it is often made somewhat greater, both in single and double covers. If there is

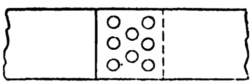


Fig. 14. Lap Joint. Zigzag Riveting.

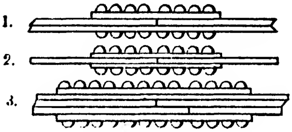


Fig. 15.

a whole plate on one side, as in No. 1, fig. 15, a single cover-plate may be used on the other side equal in thickness to the jointed plate. When the member consists of several thicknesses of plate, as in No. 3, they may with advantage be all jointed between one pair of cover-plates, arranging the several layers to break joint one with another, and leaving between every two joints a space containing a sufficient number of rivets to take up the pull of one layer of plates.

STEEL ROOF FRAMING AND COVERING.

Some points to be considered in designing steel roofing :—

(a) *Nature of Covering.*—If of corrugated sheeting, either ridged or curved type may be adopted ; in the latter the joints near crown of roof lie nearly horizontal, and leakage through rain being blown in must not be overlooked. A continuous covering, such as tarred felt or asphalt laid on timber boarding, is applicable to ridged or curved types.
With slates or tiles, the ridged type (except in very large spans) is necessary.

(b) *Rise of Roof.*—Where this is not controlled by architectural requirements, a rise of about $\frac{1}{4}$ span will generally prove most economical. Where water-tightness is essential. $\frac{1}{4}$ rise may be adopted ; a rise of even $\frac{1}{3}$ may be used with slates or tiles. In very exposed positions a low rise of $\frac{1}{6}$ has been provided, but here leakage is likely to prove troublesome.

(c) *Lighting and Ventilation in Roof.*—Skylights can be arranged in any position on the slope of a ridged roof, or set low down on curved roofs. Lights are often combined with ventilators at the centre or ridge of the roof. Ventilation at the ridge may consist merely in slightly raising the ridge cap above the sheets ; by the use of special ridge sheeting (often curved) supported on raised purlins ; or where greater ventilation is desired, the cover may be raised to a considerable height and louvres provided for weather-tightness. See sketch examples.

(d) *Temporary Structures.*—Here the framing and covering can be designed as light as possible, and factor of safety reduced to about 3, since allowance for rusting and general deterioration need not be considered. The self-supporting type of curved roof should be considered in this case.

(e) *Special Headroom* generally requires a special type, as the main tie cannot economically be raised much over $\frac{1}{10}$ rise for ridged or curved trusses of ordinary design. Two suggestions are given in sketch examples to meet this requirement.

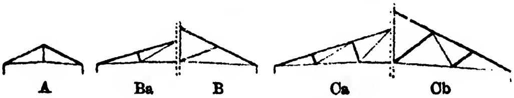
Struts, or members in compression, should be kept as short in length as possible. The strength of *ties* to resist tension is independent of their length, and no consideration on this point is necessary, except that relating to the carrying of their own weight.

The Purlins (or horizontal bearers which carry the weight of the roof covering, and the wind and snow load on the same, and transmit these loads to the roof trussing) should rest, wherever possible, on the rafters close to the points where the struts and ties meet, and stiffen the same: loads applied between these junction points (usually termed 'joints') may induce considerable *bending stresses* which must be carefully allowed for, generally at considerable expense, owing to the large increase in section which may be necessary.

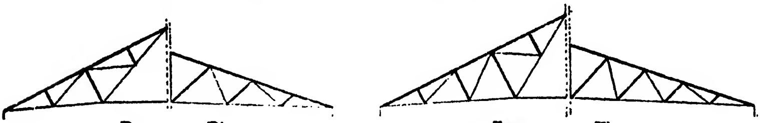
The unsupported length of the Rafter, that is to say the length between the joints, should not exceed 7 to 8 feet where the purlins are fixed close to the joints; where purlins occur between the joints, this length should be considerably reduced, or the section of the rafter increased to resist the bending stresses.

In the brief notes accompanying the outline types of roofs shown here, the application of some of the preceding hints will be evident.

For ordinary conditions the types shown below are suitable up to the following spans:—A, 14 to 16 feet; B, 25 to 30 feet; C, 45 to 50 feet; D, 55 to 60 feet; and E, 70 to 80 feet.

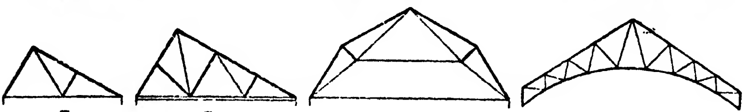


A Ba B Ca Cb



Da Db Ea Eb

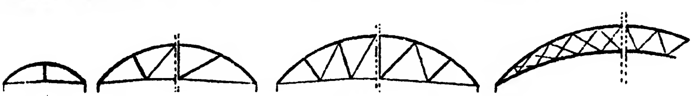
The two rises indicated in B to E are ¼ span and ⅙ span. Where special conditions do not arise, the form of bracing should be decided by considerations of length and inclination of the several struts, and convenience for designing efficient junctions between the struts and ties.



F G H J

Types of ridged roofs designed to provide special lighting, such as is required in spinning and weaving factories, are shown in F and G; the long slope is covered with sheets or slates, and the short, upright slope glazed and turned towards the north, thereby ensuring efficient light free from direct sunshine. These types are frequently called 'Saw Tooth' or 'Northern Light' roofs, and are suitable for spans of 18 to 20 ft., and 25 to 30 ft. respectively.

Types H and J show a high rise, with main tie raised to give increased headroom; the former may frequently be adopted for factories, the lower slopes being glazed and the upper slopes sheeted or slated.

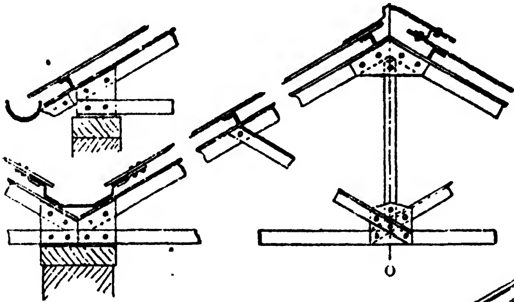


K La Lb Ma Mb Na Nb

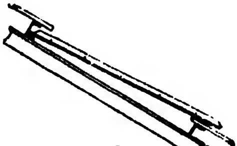
The curved types K, L, and M, are similar to A, B, and C, but arranged for a curved sheet covering, and may be adopted for similar spans.

Types N are in reality curved lattice girders, and may be designed for large spans such as are required for markets, railway stations, public halls, or the like.

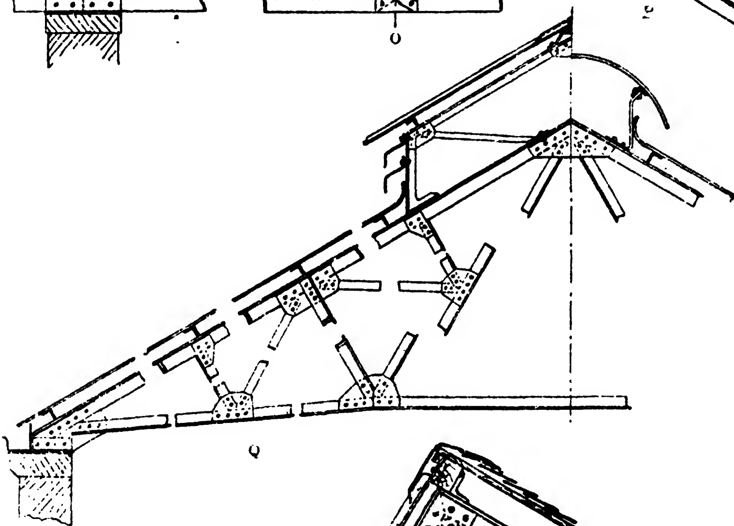
Sketch Details.—Sketch O suggests details for a ridged roof, type Bb ; on left two spans meet and a valley gutter is provided, and above this a sketch of an eaves gutter. At the apex, or ridge an ordinary ridge cap is shown on the left, and on the right a plain ridge cap is slightly raised on light brackets to provide a small amount of ventilation.



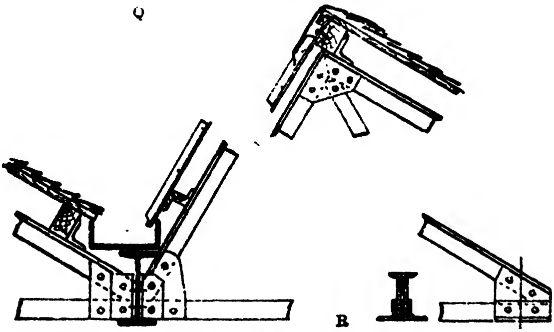
O



P



Q



R

Sketch P indicates a skylight in the slope of a roof. Sketch Q gives suggestions for type Da. At apex a simple ventilator is shown on the right, and a larger louvred ventilator on the left. In the latter arrangement the straight cover may be glazed, thus forming a combined skylight and ventilator.

Sketch R is that for a 'Northern Light' roof, types F or G. A valley detail (showing this part supported on a rolled steel beam) with valley gutter is given, also an eaves shoe detail. The long slope is covered with timber and slates, and the short slope is glazed.

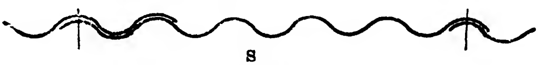
Corrugated Sheets and Fixings.

Sheets can be procured in lengths from 5 to 10 feet, and in thicknesses from No. 16 to No. 26 B.G.; for special or very temporary purposes, No. 28 B.G. might be adopted.

Corrugations from 2 inches width in light sheets to 6 inches in heavy sheets may be got, but for usual conditions the following sizes will meet all requirements :—

5–5 inch corrugated sheets (depth of corrugations 1½ in.)
6–5 " " " (" " ") See Sketch S, Fig. 16.
8–3 " " " (" " ¾ ")
10–3 " " " (" " ")

For ordinary conditions, a side lap of one corrugation (see right-hand of Sketch S) would be adopted, but for special water-tightness or stiffness, two corrugations lap could be given (see left hand of Sketch S). The effective width of the above sheets would be 25 inches or 30 inches, for single lap, and 20 inches or 25 inches for double lap.



S

FIG. 16.

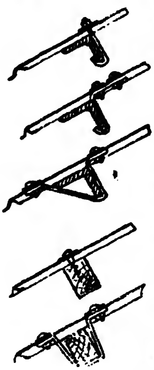
In the 3-inch corrugated sheets, the corresponding widths would be 24 and 30 inches for single lap, and 21 and 27 inches for double lap.

The end lap is generally given at 6 inches.

Sheets should be connected together by means of galvanised rivets or bolts and washers: these would be placed about 9 inches apart along the side laps, and at every second corrugation in the end laps, being in all cases placed on *top* of the corrugations to check leakage. For *double* corrugation side lap, place rivets or bolts in edge of outer sheets 6 inches apart, and 12 inches apart in inner corrugation.

Five-inch corrugations should always be adopted for No. 16 B.G. sheets, and may be used for No. 18 or 20 B.G.: 3-inch corrugations may be used for 18 or 20 B.,G., and always for No. 22 B.G. and upwards

In very exposed positions, and particularly with light gauge sheets, Wind Ties of flat or light angle section are sometimes carried along the length of the roof, and held in position by means of the purlin bolts. A line of these wind ties near each eaves level will considerably strengthen the roofing; and it is sometimes also desirable to add ties over some of the intermediate horizontal joints in the roof sheets, to further resist excessive winds.



T

U

V

W

X

In addition to connecting the roof sheets together, they must be securely fixed to the roof purlins, and suggestions for doing this are here indicated :—

Sketch T, simple hook bolt, with nut and washer outside, connecting sheets to steel angle purlin.

Sketch U, similar to above, but formed of flat clip bolted to sheets.

Sketch V, a more secure form of flat clip carried round purlin and connected to sheets by bolts and washers at ends.

Sketch W, galvanised nail or wood screw and washer, for simple connection to timber purlin.

Sketch X, flat stirrup carried round timber purlin; generally made with slight clearance in width, to admit of adjustment due to change of temperature in the sheet covering.

The washers used with the bolts are sometimes formed of lead, or of some special form, to check leakage at the bolt holes. Red lead putty, or rope yarn soaked in red lead paint, is sometimes specified to be placed under the nut or washer, but such special sreatment is seldom justifiable, or really necessary.

Corrugated Sheets are generally galvanised, but in localities near the sea, or in manufacturing districts near chemical works where acid fumes are likely to be given off, the galvanising quickly disappears, and the sheets become eaten through and spoiled. In such conditions the galvanised sheets should receive two good coats of metallic oxide paint shortly after erection, or plain corrugated sheets used, these being well tarred and sanded after erection, and at such intervals as may be necessary to efficiently protect the sheets from corrosion.

Self-Supporting Roofing.

This term is generally applied to a type of curved roofing in which the curved rafters of the trussing are dispensed with, their place being supplied by the corrugated sheets themselves.

In the smaller spans, say up to 15 to 20 feet, the roof sheets are connected to timber or steel eaves beams, and the bracing (generally as types K or L) bolted direct to the curved roof sheets.

In the larger spans angle purlins are securely clipped to the curved sheets and extend for the whole length of the roof, and the bracing is bolted to these purlins.

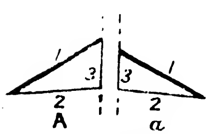
By this method of construction, and the use of corrugated roof sheets of sufficient strength to serve the double purpose of acting as a curved rafter, and as a weather protection, spans over 55 feet have been erected. It will not, however, as a rule be found economical to exceed about 35 feet span with this type of roofing; and as the life of this structure depends on the strength of the covering, it is seldom justifiable to provide self-supporting roofs for permanent use, as this would entail great watchfulness and cost in protecting the sheeting against corrosion or decay.

The curved form of covering must be used for this type of roof, and for the larger spans the sheets may be arranged to *break joint* in the longitudinal joints; greater stiffness may be obtained in the complete sheeting by this arrangement of joints. The lower sheets of the roof, which have to resist a much greater strain than the upper ones, are sometimes made of a heavier gauge. Special attention is also necessary to ensure sufficient strength when bolting or riveting the several sheets together.

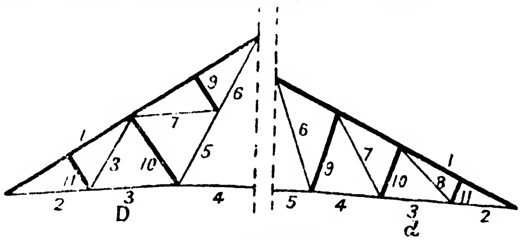
Ridged Roof Truss Multipliers.

The determination of the stresses in the several members of any roof truss is usually a comparatively simple matter, although a considerable amount of time may sometimes be required

Rise of rafters, $\frac{1}{3}$rd and $\frac{1}{4}$th span. Rise of main ties, $\frac{1}{18}$th and $\frac{1}{12}$th span.

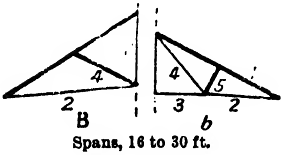


Spans to 15 ft.

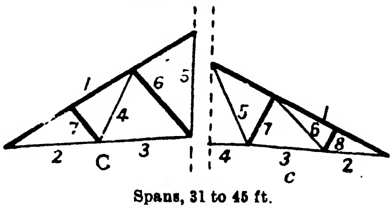


Spans, 46 to 60 ft.

Compression shown by heavy lines. Tension shown by light lines.



Spans, 16 to 30 ft.



Spans, 31 to 45 ft.

to carry out this operation. Where any of the following types of trusses can be adopted, the above calculations can be much simplified and shortened.

For special water-tightness with slate or tile covering, one-third rise (shown by capital letters) should be adopted; one-fourth rise is usual for corrugated sheet covering.

In spans up to 60 feet the rafter is usually kept the same section throughout the full length; the table gives only the max. stress multiplier for rafters.

The max. practical span for any type of truss is indicated by multiplying the numbers of sections formed in the rafters by the joints in the same by 7½ feet. Thus Type B gives 4 × 7½ = 30 feet max. span, and Type D = 8 × 7½ = 60 feet max. span.

Purlins are assumed to be set at the joints; if not, allowance must be made for the extra stress due to the bending tendency produced.

It is also assumed that the trusses are fixed at both ends; if this is not so, the stress in some members may be increased by about 10 per cent.

The multipliers in the following table are calculated as for *unit load and unit span*. Thus for dead-load stress the actual dead load of truss covering, etc., is multiplied by the particular dead-load multiplier for the required member; for live-load stress, the live load from the actual normal wind pressure on one side of the truss is used along with the particular live-load multiplier. Both stresses added together give the max. stress on the member.

Member.	Type A.		Type *a*.		Type B.		Type *b*.		Type O.		Type *c*.		Type D.		Type *d*.	
	D.L.	L.L.	D.L.	L.L.	D.L.	L.L.	D.L.	L.L.	D.L.	L.L.	D.L.	L.L.	D.L.	L.L.	D.L.	L.L.
1	·51	·58	·63	·69	·76	·70	·97	1·04	·86	·85	1·08	1·25	·90	·93	1·13	1·37
2	·42	·29	·57	·48	·63	·81	·87	1·12	·72	·98	·96	1·35	·76	1·06	1·12	1·46
3	·07	·05	·07	·06	·31	·34	·55	·45	·58	·63	·77	·92	·65	·81	·87	1·12
4	—	—	—	—	·24	·58	·33	·69	·12	·30	·56	·46	·42	·27	·72	·81
5	—	—	—	—	—	—	·22	·50	·41	·45	·28	·56	·26	·54	·57	·47
6	—	—	—	—	—	—	—	—	·21	·50	·19	·43	·36	·81	·27	·56
7	—	—	—	—	—	—	—	—	·14	·34	·22	·50	·10	·26	·19	42
8	—	—	—	—	—	—	—	—	—	—	·15	·31	·10	·26	·16	·36
9	—	—	—	—	—	—	—	—	—	—	—	—	·10	·25	·22	·50
10	—	—	—	—	—	—	—	—	—	—	—	—	·21	·50	·17	·37
11	—	—	—	—	—	—	—	—	—	—	—	—	·10	·25	·11	·25

D. L. = Dead Load. L.L. = Live Load.

Galvanised Corrugated Steel Sheets. Approximate Weight.

(Sheets 26 inches in width; 8–3 inch corrugations.)

British Standard 798, Galvanised Corrugated Steel Sheets, specifies that the weight of the galvanised coating shall be not less than 1½ oz. per square foot of sheet (including both sides).

In the case of 8–3 inch corrugations the width of the sheet before corrugation shall be not less than 29½ inches but may be ½ inch over.

The tolerance on weight of black sheets shall be plus or minus 7½ per cent., so that the following figures are only approximate.

Approximate Number of Sheets per Ton.

Length of Sheet in Feet.						
5	6	7	8	9	10	B.G.*
70	58	50	44	39	35	16
89	74	64	56	49	44	18
114	95	81	71	63	57	20
139	116	99	87	77	69	22
168	140	120	105	93	84	24
223	186	159	139	124	111	26
240	200	172	150	133	120	28

* Thickness of black sheet.

Approximate weight per 100 superficial feet of Galvanised Corrugated Straight Iron Sheets when fixed on roof.

18	20	22	24	26 B.G.
cwts. qr. lbs.	cwt. qrs. lbs.	cwt. qrs. lbs.	cwt. qr. lbs.	cwt. qr. lbs.
2 1 21	1 3 21	1 2 7	1 1 7	1 0 7

One ton of Galvanised Corrugated Iron Sheets will cover when fixed, and after **allowing for** laps, the undermentioned **areas**:—

16	18	20	22	24	26	28	B.G
600	800	1020	1280	1530	2100	2250	Square Feet.

It is a bad practice to fasten zinc-iron sheets together with iron or steel rivets. These soon rust if not painted, and the rust penetrates through the edges of the iron encircling the rivets, and thus proves disastrous. Only zinc, or zinc-covered rivets, should be so used.

The great objection to iron as a roofing material is the drip from the moisture which **condenses** on the **underneath** side of it.

Painting for Galvanised Iron Roofs.

Paint adheres badly to new galvanised iron, but after a few years it will do so readily and will add greatly to its life. In order to make paint adhere to new galvanised **iron**, brush it over with the following mixture: Chloride of Copper, 1 part; Nitrate of Copper, 1 part, Salammoniac, 1 part; Water, 64 parts, to which is added 1 part of Hydrochloric Acid. This mixture turns the galvanised iron black; it should be left for at least 12 hours before being painted.

It is stated that paint will **adhere** to galvanised iron if the **surface** is washed with vinegar before painting.

(See also 'Painting,' Section XLII, Vol. II.)

Varnish for Galvanised Iron Roofs.

Heat 100 gallons of tar to a low boiling point, and add 100 lbs. of fresh slaked lime, sifted over the top and then worked down. Boil this mixture until it becomes pasty. Let it settle for a few minutes and then add 20 lbs. of tallow and 5 lbs. of powdered resin. Stir until thoroughly mixed and all ingredients dissolved; then allow to cool. The mixture should not be raised to a higher temperature than 100 degs. F. If too thick, it can be thinned down with paraffin or naphtha.

Transverse Strength of Corrugated Iron.

If W = breaking weight in tons (distributed); L = unsupported length of sheet in **inches**; t = **thickness** of sheet in inches; b = breadth of sheet in inches; d = depth of **corrugations in inches**:

$$W = \frac{44{\cdot}6\,t\,b\,d}{L}.$$

LIMITING DIMENSIONS OF ROLLED STEEL SECTIONS.

Dimensions. O, Ordinary. M, Maximum.	Flat.		Round.		Square.		Angle.		Tee.		Joists.		Channel.	
	O	M	O	M	O	M	O	M	O	M	O	M	O	M
Length in feet	40	60	24	60	24	60	50	60	50	60	36	50	36	60
Width in inches	12	12	4	4	4	4	6×6	8×8	5×3	6×3	24×7½	24×7½	17×4	17×4
Thickness in ins.	1	1	—	—	—	—	⅞	1	⅜	½	½	⅝	⅝	⅝

Universal Mill Plates. Length 95 ft. Width 45 in.
Rolled Steel Plates. Length 65 ft Width 108 in.

BRITISH STANDARD SECTIONS.*

(B.S. 4, 4A & 6.) ***(Abstract.)***

The British Standards Institution list of Rolled Steel Sections for structural purposes comprises seven different sections, viz. (1) Equal Angles, (2) Unequal Angles, (3) Bulb Angles, (4) Bulb Plates, (5) Channels, (6) Beams, and (7) T Bars.

The following are abbreviated tables showing some of the standard dimensions recommended.

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* **By permission of the British Standards Institution.**

BRITISH STANDARD SECTIONS.

EQUAL ANGLES.

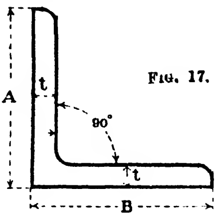


FIG. 17.

BULB PLATES.

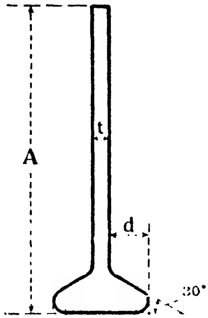


FIG. 21.

BULB ANGLES.

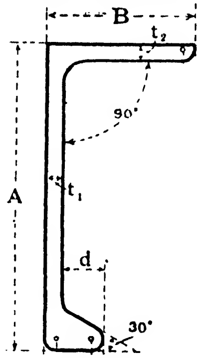


FIG. 20.

UNEQUAL ANGLES

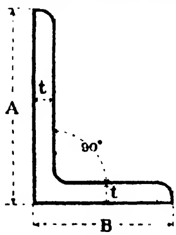


FIG. 18.

T BARS.

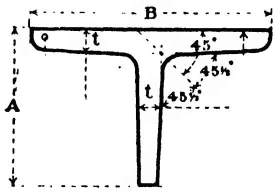


FIG. 19.

BEAMS.

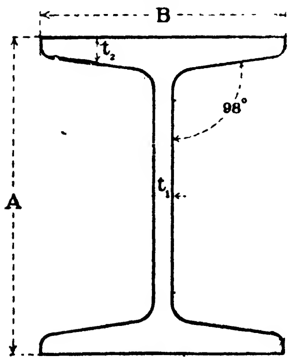


FIG. 23.

CHANNELS.

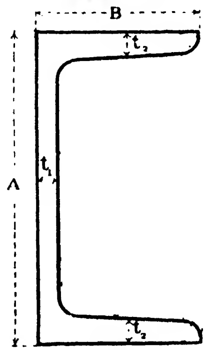
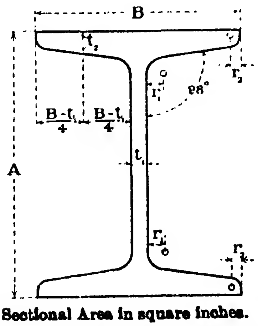


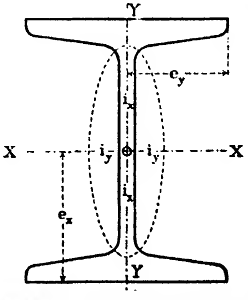
FIG. 22.

BRITISH STANDARD SECTIONS.—BEAMS.

Size. A × B.	Approx. Weight. per ft.	Standard Thickness. Web	Standard Thickness. Flange	Sectional Area.	Moments of Inertia.		Radii of Gyration.		Moduli of Section.	
ins.	lbs.	ins.		ins.²	ins.⁴		ins.		ins.³	
				Girder Sections.						
3 × 1½	4	·16	·249	1·177	1·660	·125	1·188	·326	1·107	·167
4 × 1¾	5	·17	·239	1·470	3·664	·186	1·579	·356	1·832	·213
4¾ × 1¾	6·5	·18	·325	1·910	6·730	·260	1·88	·370	2·830	·300
5 × 3	11	·22	·376	3·260	13·680	1·450	2·050	·670	5·470	·970
6 × 3	12	·23	·377	3·533	20·989	1·461	2·437	·643	6·996	·974
7 × 4	16	·25	·387	4·750	39·510	3·370	2·890	·840	11·290	1·690
8 × 4	18	·28	·398	5·296	55·629	3·506	3·241	·814	13·907	1·753
9 × 4	21	·30	·457	6·177	81·127	4·148	3·624	·820	18·028	2·074
10 × 4½	25	·30	·505	7·354	122·338	6·486	4·079	·939	24·468	2·883
12 × 5	32	·35	·550	9·450	221·070	9·690	4·840	1·010	36·840	3·880
13 × 5	35	·35	·604	10·298	283·507	10·815	5·247	1·025	43·616	4·326
14 × 6	46	·40	·698	13·590	442·570	21·450	5·710	1·260	63·220	7·150
15 × 6	45	·38	·655	13·236	491·912	19 871	6·096	1·225	65·588	6·624
16 × 6	50	·40	·726	14·705	618·092	22·468	6·483	1·236	77·261	7·489
18 × 6	55	·42	·757	16·182	841·759	23·635	7·212	1·209	93·529	7·878
20 × 6½	65	·45	·820	19·119	1226·172	32·559	8·008	1·305	122·617	10·018
22 × 7	75	·50	·834	22·064	1676·796	41·065	8·718	1·364	152·436	11·733
24 × 7½	95	·57	1·011	27·940	2533·040	62·540	9·520	1·500	211·090	16·680
				Heavy Beams and Pillars.						
4 × 3	10	·24	·347	2·940	7·786	1·326	1·627	·672	3·893	·884
5 × 4½	20	·29	·513	5·882	25·025	6·590	2·063	1·058	10·010	2·929
6 × 5	25	·41	·520	7·370	43·690	9·100	2·440	1·110	14·560	3·640
8 × 6	35	·35	·648	10·296	115·058	19·540	3·343	1·378	28·764	6·513
9 × 7	50	·40	·825	14·712	208·128	40·169	3·761	1·652	46·241	11·477
10 × 6	40	·36	·709	11·771	204·803	21·759	4·171	1·360	40·961	7·253
10 × 8	55	·40	·783	16·177	288·688	54·743	4·224	1·840	57·738	13·686
12 × 8	65	·43	·904	19·122	487·769	65·184	5·051	1·846	81·295	16·296
14 × 8	70	·46	·920	20·589	705·575	66·674	5·854	1·800	100·796	16·668
16 × 8	75	·48	·938	22·063	973·909	68·303	6·644	1·759	121·739	17·076
18 × 8	80	·50	·950	23·526	1292·073	69·427	7·411	1·718	143·564	17·357

The properties of British Standard Sections in above table are printed by permission of the British Standards Institution.





a = Sectional Area in square inches.

3·4 a = Weight in lb. per foot (approximately).

c_x c_y Distance of Centre of Gravity from X axis and Y axis.

J = ai² Moment of Inertia.

$= \sqrt{\frac{J}{a}}$ Radius of Gyration.

e_x e_y Distance of outer fibres from X and Y axes.

$Z = \frac{J}{e}$ Modulus of Section.

Safe Loads Uniformly Distributed in Tons on British Standard Steel Beams.
Maximum Stress = $7\frac{1}{2}$ tons per square inch.

Size and Weight per foot.	Span in Feet.																
	4	6	8	10	12	14	16	18	20	22	24	26	28	30	32	36	40
ins. ins. lbs.																	
24×7½×95	—	—	—	112·0	93·8	80·4	70·3	62·5	56·2	51·1	46·9	43·2	40·2	37·5	35·1	31·2	28·1
22×7 ×75	—	—	—	81·2	67·7	58·0	50·8	45·1	40·6	36·9	33·8	31·2	29·0	27·0	25·4	22·5	20·3
20×6¼×65	—	—	—	65·3	54·4	46·7	40·8	36·3	32·6	29·7	27·2	25·1	23·3	21·7	20·4	18·1	16·3
18×8 ×80	—	—	—	76·5	63·8	54·6	47·8	42·5	38·2	34·8	31·9	29·4	27·3	25·5	23·9	21·2	17·2
18×6 ×55	—	—	—	49·8	41·5	35·6	31·1	27·7	24·9	22·6	20·7	19·1	17·8	16·6	15·5	13·8	11·2
16×8 ×75	—	—	—	64·9	54·1	46·3	40·5	36·0	32·4	29·5	27·0	24·9	23·1	21·6	20·2	16·0	12·9
16×6 ×50	—	—	—	41·2	34·3	29·4	25·7	22·8	20·6	18·7	17·1	15·8	14·7	13·7	12·8	10·1	8·2
15×6 ×45	—	—	—	34·9	29·1	24·9	21·8	19·4	17·4	15·9	14·5	13·4	12·4	11·6	10·2	8·0	—
14×8 ×70	—	—	—	53·7	44·7	38·3	33·5	29·8	26·8	24·4	22·3	20·6	19·1	16·7	14·7	—	—
14×6 ×46	—	—	—	33·7	28·0	24·0	21·0	18·7	16·8	15·3	14·0	12·9	12·0	10·4	9·2	—	—
31×5 ×35	—	—	—	23·2	19·3	16·6	14·5	12·9	11·6	10·5	9·6	8·9	7·7	6·7	—	—	—
12×8 ×65	—	—	—	43·3	36·1	30·9	27·0	24·0	21·6	19·7	18·0	15·3	13·2	—	—	—	—
12×5 ×32	—	—	—	19·6	16·3	14·0	12·2	10·9	9·8	8·9	8·1	6·9	6·0	—	—	—	—
10×8 ×55	—	—	—	30·7	25·6	21·9	19·2	17·1	15·3	14·0	12·8	—	—	—	—	—	—
10×6 ×40	—	—	27·3	21·8	18·2	15·6	13·6	12·1	10·9	9·9	9·1	—	—	—	—	—	—
10×4½×25	—	21·7	16·3	13·0	10·8	9·3	8·1	7·2	6·5	6·0	5·4	—	—	—	—	—	—
9×7 ×50	—	—	30·8	24·6	20·5	17·6	15·4	13·7	11·1	—	—	—	—	—	—	—	—
9×4 ×21	24·0	16·0	12·0	9·6	8·0	6·8	6·0	5·3	4·3	—	—	—	—	—	—	—	—
8×6 ×35	—	25·5	19·1	15·3	12·7	10·9	9·5	7·5	6·1	—	—	—	—	—	—	—	—
8×4 ×18	18·5	12·3	9·2	7·4	6·1	5·2	4·6	3·6	2·9	—	—	—	—	—	—	—	—
7×4 ×16	15·0	10·0	7·5	6·0	5·0	4·3	3·2	2·6	—	—	—	—	—	—	—	—	—
6×5 ×25	19·4	12·9	9·7	7·7	6·4	4·7	3·6	—	—	—	—	—	—	—	—	—	—
6×3 ×12	9·3	6·2	4·6	3·7	3·1	2·2	1·7	—	—	—	—	—	—	—	—	—	—
5×4½×20	13·3	8·8	6·6	5·3	3·7	—	—	—	—	—	—	—	—	—	—	—	—
5×3 ×11	7·2	4·8	3·6	2·9	2·0	—	—	—	—	—	—	—	—	—	—	—	—
4¾×1¾×6·5	3·7	2·5	1·8	1·4	—	—	—	—	—	—	—	—	—	—	—	—	—
4×3 ×10	5·1	3·4	2·5	·6	—	—	—	—	—	—	—	—	—	—	—	—	—
4×1¾×5	2·4	1·6	1·2	·78	—	—	—	—	—	—	—	—	—	—	—	—	—
3×1½×4	1·4	·98	·55	—	—	—	—	—	—	—	—	—	—	—	—	—	—

BRITISH STANDARD SECTIONS.

EQUAL ANGLES.

Size. A × B	Standard. Thickness. t	Calculated Weight per Foot.	Sectional Area.	Centre of Gravity.	Moments of Inertia.	Radii of Gyration.	Moduli of Section.
ins.	ins.	lb.	ins.²	ins.	ins.⁴	ins.	ins.³
1×1	·125 ·250	·80 1·49	·234 ·437	·285 ·334	·020 ·035	·292 ·284	·028 ·053
1½×1½	·1875 ·250	1·79 2·34	·53 ·69	·43 ·46	·100 ·130	·450 ·440	·100 ·130
2 ×2	·1875 ·3125	2·43 3·92	·71 1·15	·56 ·61	·260 ·400	·600 ·590	·180 ·290
2½×2½	·250	4·04	1·188	·704	·680	·757	·379
	·375	5·90	1·735	·753	·962	·745	·550
	·500	7·65	2·250	·799	1·206	·732	·709
3 ×3	·250	4·89	1·438	·826	1·203	·915	·553
	·375	7·17	2·110	·877	1·720	·903	·810
	·500	9·35	2·750	·924	2·178	·890	1·049
3½×3½	·250	5·74	1·690	·95	1·940	1·070	·760
	·375	8·45	2·490	1·00	2·800	1·060	1·120
	·500	11·05	3·250	1·05	3·570	1·050	1·460
4 ×4	·375	9·73	2·860	1·12	4·260	1·220	1·480
	·500	12·75	3·750	1·17	5·460	1·210	1·930
	·625	15·68	4·610	1·22	6·560	1·190	2·360
6 ×6	·5 ·625	19·55 24·17	5·75 7·11	1·66 1·71	19·480 23·730	1·840 1·830	4·490 5·540
8 ×8	·625 ·75	32·68 38·89	9·61 11·44	2·20 2·25	58·260 68·580	2·460 2·450	10·050 11·940

UNEQUAL ANTLES.

Size. A × B	Standard. Thickness. t	Calculated Weight per Foot.	Sectional Area.	Centre of Gravity.		Moments of Inertia.		Radii of Gyration.		Moduli of Section.	
2 ×1½	·1875 ·25	2·11 2·76	·62 ·81	·630	·380	·240	·110	·620	·430	·170	·100
				·650	·410	·310	·150	·610	·420	·230	·130
3 ×2	·250	4·04	1·188	·977	·483	1·060	·375	·945	·562	·524	·247
	·3125	4·98	1·460	1·000	·510	·290	·450	·940	·560	·650	·300
	·375	5·90	1·735	1·028	·532	1·505	·528	·931	·551	·763	·359
3½×2½	·250	4·89	1·440	1·090	·600	1·750	·740	1·100	·720	·730	·390
	·3125	6·04	1·780	1·120	·630	2·140	·910	1·100	·710	·900	·480
	·375	7·17	2·110	1·150	·650	2·510	1·060	1·090	·710	1·070	·570
4 ×3	·3125	7·11	2·090	1·240	·750	3·300	1·590	1·260	·870	1·200	·710
	·375	8·45	2·490	1·270	·770	3·890	1·870	1·250	·870	1·420	·840
	·500	11·05	3·250	1·320	·820	4·970	2·370	1·240	·850	1·850	1·090
5 ×4	·375 ·500	11·00 14·45	3·236 4·251	1·507	1·012	7·967	4·532	1·569	1·183	2·281	1·517
				1·558	1·061	10·289	5·828	1·566	1·171	2·989	1·983
6 ×4	·375 ·500	12·28 16·16	3·611 4·752	1·914	·924	13·213	4·743	1·913	1·146	3·233	1·542
				1·967	·975	17·142	6·113	1·899	1·134	4·250	2·020
8 ×4	·625	24·17	7·109	2·822	·895	46·375	7·871	2·554	1·052	9·060	2·535
9 ×4	·75	21·25 31·24	6·25 9·19	3·270	·800	52·460	6·650	2·900	1·030	9·160	2·080
				3·380	·900	75·450	9·870	2·870	1·010	13·430	3·020

BRITISH STANDARD SECTIONS.

Bulb Angles.

Size. $A \times B$	Standard Thickness. Web. t_1 Flange. t_2	Calculated Weight per Foot.	Sectional Area.	Centres of Gravity. c_x	Centres of Gravity. c_y	Moments of Inertia.	Moments of Inertia.	Radii of Gyration. i_x	Radii of Gyration. i_y	Moduli of Section.	Moduli of Section.
ins.	ins.	lb.	ins.²	ins.		ins.⁴		ins.		ins.³	
4 × 2½	·26	6·29	1·849	1·577	·555	3·798	·805	1·433	·660	1·568	·414
5 × 2½	·30	8·49	2·498	2·131	·529	8·094	·968	1·800	·622	2·821	·491
6 × 3	·33	11·37	3·344	2·581	·622	15·865	1·866	2·178	·747	4·640	·785
7 × 3	·38	14·62	4·300	3·147	·616	27·480	2·213	2·528	·717	7·133	·928
8 × 3	·40	17·24	5·070	3·752	·607	42·303	2·435	2·888	·693	9·957	1·018
9 × 3½	·43	21·22	6·241	4·208	·697	66·703	4·147	3·269	·815	13·919	1·480
10 × 3½	·45	24·34	7·159	4·825	·694	94·102	4·526	3·626	·795	18·185	1·613
11 × 3½	·48	28·14	8·276	5·437	·700	130·469	5·054	3·970	·781	23·453	1·805
12 × 4	·50	32·62	9·593	5·911	·784	182·473	7·750	4·361	·899	29·968	2·410
15 × 4	·57	45·40	13·352	7·807	·812	388·085	10·289	5·391	·878	53·949	3·228

Bulb Plates.

Size. A	Standard Thickness. t	Calculated Weight per Foot.	Sectional Area.	Centre of Gravity.	Moments of Inertia.	Radii of Gyration.	Radii of Gyration.	Moduli of Section.	Moduli of Section.
ins.	ins.	lb.	ins.²	ins.	ins.⁴	ins.		ins.³	
7	·35	12·57	3·698	2·447	18·118	2·213	·363	3·980	·448
10	·50	25·66	7·547	3·496	75·462	3·162	·518	11·602	1·308

BRITISH STANDARD SECTIONS.

Channels.

Size. A × B	Standard (Minimum) Thickness. Web. t_1	Flange. t_2	Calculated Weight per Foot.	Sectional Area.	Centre of Gravity. c_y	Moments of Inertia.		Radii of Gyration.		Moduli of Section.	
ins.	ins.		lb.	ins.²	ins.	ins.⁴		ins.		ins.³	
3×1½	·20	·28	4·60	1·352	·476	1·823	·261	1·161	·439	1·215	·255
4×2	·24	·31	7·09	2·085	·599	5·063	·703	1·558	·581	2·532	·502
5×2½	·25	·38	10·22	3·006	·773	11·873	1·641	1·987	·739	4·749	·950
6×3	·25	·38	12·41	3·650	·890	21·271	2·825	2·414	·880	7·090	1·339
7×3	·26	·42	14·22	4·182	875	32·750	3·255	2·798	·882	9·357	1·531
8×3½	·32	·52	20·21	5·944	1·045	60·573	6·370	3·192	1·035	15·143	2·595
9×3½	·34	·54	22·27	6·549	1·003	82·617	6·899	3·552	1·026	18·359	2·763
10×3½	·36	·56	24·46	7·193	·965	109·520	7·420	3·902	1·016	21·904	2·927
12×3½	·38	·50	26·37	7·76	·97	159·73	7·15	4·54	·96	26·62	2·68
12×4	·40	·60	31·33	9·214	1·055	200·086	12·120	4·660	1·147	33·348	4·116
15×4	·41	·62	36·37	10·696	·967	349·095	13·338	5·713	1·117	46·546	4·398
17×4	·48	·68	44·34	13·041	·920	520·178	15·263	6·316	1·082	61·197	3·955

T Bars.

Size. Flange, Web. B × A	Standard Thickness. t	Radii. Root. r_1	Toe. r_2	Calculated. Weight per Foot. w	Sectional Area. a	Centre of Gravity. c_x	c_y	Moments of Inertia. J_x	J_y	Radii of Gyration. i_x	i_y	Moduli of Section. Z_x	Z_y
ins.	ins.	ins.		lb.	ins.²	ins.		ins.⁴		ins.		ins.³	
1½×1½	·250	·21	·15	2·36	·693	·459	0	·135	·067	·442	·312	·130	·090
2×2	·250	·24	·17	3·21	·945	·579	0	·337	·157	·597	·408	·237	·157
2½×2½	·250	·27	·19	4·07	1·197	·697	0	·678	·305	·752	·504	·376	·244
	·375			5·92	1·742	·749	0	·959	·475	·742	·522	·548	·380
3×3	·375	·30	·21	7·20	2·119	·869	0	1·708	·813	·898	·619	·801	·542
4×3	·375	·33	·23	8·49	2·498	·766	0	1·860	1·911	·863	·875	·833	·956
	·500			11·09	3·262	·816	0	2·366	2·595	·852	·892	1·083	1·298
5×4	·375	·39	·27	11·06	3·250	1·000	0	4·470	3·700	1·170	1·070	1·490	1·480
	·500			14·50	4·266	1·052	0	5·773	5·024	1·163	1·085	1·958	2·009
6×4	·500	·42	·29	16·22	4·772	·968	0	6·070	8·642	1·128	1·346	2·002	2·881
	·625			19·99	5·878	1·018	0	7·331	10·932	1·117	1·364	2·459	3·644
6×6	·500	·48	·34	19·62	5·772	1·630	0	19·043	8·556	1·816	1·218	4·357	2·852
	·625			24·23	7·126	1·687	0	23·309	10·866	1·809	1·235	5·404	3·622

SECTION VIII

SURVEYING—SURVEYING INSTRUMENTS—ASTRONOMY—ASTRONOMY IN FIELDWORK (pp. 249-274)

(Revised by Reginald Ryves, M.Cons.E., and Percy M. Ryves, F.R.A.S.)

SECTION VIII

SURVEYING.

(Revised by Reginald Ryves, M.Cons.E., and Percy M. Ryves, F.R.A.S.)

Chaining.

Chains.—The surveying chain is 22 yards = 66 feet long. It consists of 100 links, and each link = 7·92 inches. This chain is known as 'Gunter's' chain.

Another chain is known as the '100-foot' chain, and consists of 100 links, each 1 foot long. It is used almost exclusively for work done abroad.

A third form is the 'Decimetre' chain; this is divided into 100 links, each of which measures a decimetre = ·328 foot.

Chain Measurement.—The most careful measurement that can be made with the common surveying chain is liable to an error of $\frac{1}{3000}$ of the distance measured, and an ordinary measurement across fields, &c., to an error of $\frac{1}{500}$.

Chain measurement alone will not suffice for a greater extent than a square mile on a scale of 6 inches to a mile; or than $\frac{1}{16}$ square mile on a scale of 20 inches to a mile.

Arrows.—With each chain 10 arrows or pins of strong wire about 1½ feet long, and preferably having a piece of red cloth tied to the ring of each, are carried.

Ranging.

Poles and Ranging Rods.—The former are poles of red pine about 15 feet high, 2 inches thick at the bottom and ¾ inch at the top; each should be shod with an iron shoe. The ranging rods are about 6 feet high, and have a white and red flag about 9 inches square at the top.

Offset Staff.—A narrow slip of deal about 1½ inches wide, 1 inch thick, and 10 links long, divided into links. It is used for the purpose of measuring small distances from the line to hedges, &c.

The statute acre in England consists of 10 square chains, that is, of 10 square blocks of land whose length and breadth are 1 chain each; or, taking 10 blocks as one whole block, then an acre is equal to the area of a rectangle whose base is 10 chains and perpendicular 1 chain; or to any other rectangle, the product of whose base and perpendicular also equals 10 chains: thus, 4 chains base with 2½ chains perpendicular, and 2 chains base with 5 chains perpendicular, are each equal to 1 acre.

With any quantity less than 1 chain, to bring feet into links add ½ more; to bring links into feet take ⅓ less.

MENSURATION.

To find the Area of a Triangle when the Base and Perpendicular Height are given.—**Multiply the base by half the height, or *vice versa*.**

To find the Area of a Triangle when the Three Sides are given.—**Take half the sum of the sides, subtract each side severally from this sum, then multiply this and the three remainders together, and take the square root for the area.**

To find the Area of a Rectangular Figure.—**Multiply the length by the breadth the product will be the area.**

To find the Area of a Trapezoid.—**Multiply half the sum of the two parallel sides by the distance between them.**

To find the Area of a Parallelogram whose Angles are not Right Angles.—**Multiply the length of any one of the sides by the perpendicular let fall on it from the angle opposite that side.**

To find the Area of a Trapezium.—**Divide the trapezium into two triangles, and find the area of the latter by the first rule.**

To find the Area of an Irregular Polygon.—**Divide the polygon into trapeziums and triangles, and find the sum of the areas of each.**

To find the Area of an Irregular Figure.—**Draw the figure on fine cardboard or thin sheet-metal, cut the same carefully out, and weigh it with an accurate balance. Then this weight, compared with the weight of a piece of the cardboard or metal of a definite size—say 1 inch square—gives at once the area required.**

Calculation of Areas from Co-Ordinates.

If necessary adjust the traverse so that the co-ordinates correctly close the figure.

Consider each line round the figure, not the stations.

Take the total, or reduced, departures of the stations at each end of one line, add them algebraically, and divide by two. This gives the 'middle departure' of that line.

Repeat for all the lines of the figure.

It makes no difference from what initial point the departures are reckoned, whether from one of the stations on the figure, or from another point either inside or outside the figure.

Taking each line round the figure in order—it makes no difference which way round—take the difference between the reduced latitudes of the stations at each end of the line. The result is the same as the actual amount of Northing or Southing made by that particular line.

Multiply these results for each line by the middle departure of the same line, paying attention to the algebraic sign of each factor, and add all the results together algebraically.

The result is that the area of the figure in squares of the units employed in measuring the length of the sides.

If these are feet, then

divide by 27,878,400 (Log 7·445 2678) for sq. miles; divide by 43,560 (Log 4·639 0878) for acres.

In the logarithmic computation of middle departures by latitudes, it is convenient to use the arithmetical complements of the logs given in preceding paragraph, adding all three logs, and thus avoiding taking out differences.

The terms latitude and departure are mutually interchangeable throughout the above.

TRIANGULATION.

For large surveys a base line is accurately measured and triangles built upon it, the sides being for 6 inches scale from ¾ to 1½ mile. Each point should, if possible, be fixed by three distances.

EXAMPLE.—AB (fig. 1) is the base line: C, D, E, F are convenient points to be fixed by triangulation.

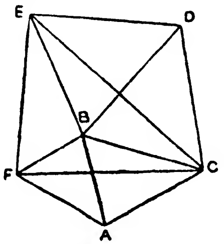


FIG. 1.

AC, AF, BF and BC being obtained, two values of FC can then be found from FBC and FAC, and then two values of FE from FEB and FEC, and so on. In plotting the triangulation the long lines EF, FC, EC should be first laid down.

AB = 5,230 feet measured by chain.

$$\left.\begin{aligned} BAC &= 55^\circ\ 30'\ 15'' \\ CBA &= 64^\circ\ 21'\ 25'' \end{aligned}\right\}\text{ measured by instrument.}$$

$$BCA = 60^\circ\ 8'\ 20''$$

$$BC = \frac{AB \sin BAC}{\sin BCA}.$$

$$\begin{aligned} \text{Log } AB &= 3{\cdot}7185017 \\ \text{Log sin } BAC &= 9{\cdot}9160154 \\ &\overline{\ 13{\cdot}6345171\ } \\ \text{Log sin } BCA &= 9{\cdot}9381368 \\ \text{Log } BC &= 3{\cdot}6963803 \\ BC &= 4970 \end{aligned}$$

The triangles are then broken up into smaller ones and measured by the chain.

Simson's Rule for Area Computation.

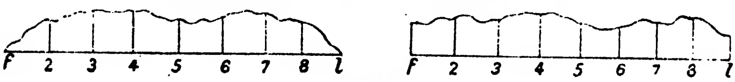


FIG. 2.

Divide base into any even number of equal parts, and measure length of perpendiculars or ordinates through each division.

If S = sum of lengths of first (f) and last (l) ordinates; E = sum of lengths of intermediate even ordinates; N = sum of lengths of intermediate odd ordinates; d = common distance between ordinates; then

$$\text{Area} = \frac{S + 4E + 2N}{3} \times d.$$

Rate of Surveying.

The average daily progress of a good surveyor in England, surveying with one chainman, for the 6-inch-to-a-mile scale, is, according to Frome—

Close large village	about	5 acres.
Villages and surrounding fields, etc.	"	14 acres.
Close country, gentlemen's houses, etc.	"	20 acres.
Medium country, ordinary fields, and scattered farms	"	30 to 32 acres.
Open moorland, with roads, streams, boundaries, cart tracks, etc. (no fields)	"	55 acres.

A survey of a piece of country of about 6 square miles, on a scale of 6 inches to a mile (which would comprise measuring and levelling base line, triangulating, finding variation of the compass, traversing the principal roads, filling in detail by prismatic compass, contouring to every 25 feet, and completing fair plan), would take a good surveyor thirty working days.

A greater area takes less time in proportion, for the base is the same, as also variation of compass and part of the triangulation.

TO ASCERTAIN THE DISTANCE OF AN INACCESSIBLE OBJECT.

To measure a Base Line across a River.

Let DA (fig. 3) be the direction of the line which has been measured up to the river. It is required to ascertain the distance BA at once upon the ground, so as to continue the measurement of the line.

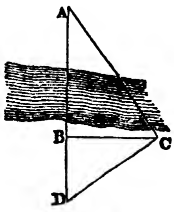


FIG. 3.

On the line DA take any point, B, whence a perpendicular, BC, can be taken, which will be free from obstruction, so that BC can be accurately measured; carry the range on across the river, and at A set up a flag; on BC take any point, C, whence A can be seen; and at C erect a perpendicular to AC, intersecting the line AD on D.

Measure DB; then

$$AB = \frac{BC^2}{BD}.$$

These angles can either be taken by a cross-staff or by the chain, with the distances of 30 links, 40 links, and 50 links; 50 being the hypotenuse of a right-angled triangle when the base and perpendicular are respectively 30 and 40.

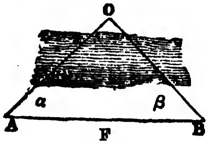


FIG. 4.

To measure the Width of a River with the Base Line alongside of it.

Take at the ends of the base line, AB (fig. 4), with a theodolite, the angles α and β, made between the base line and a flag, O, placed at the edge, on the other side of the river; then width OF is

$$OF = AB\,\frac{\tan\alpha\tan\beta}{\tan\alpha + \tan\beta}.$$

To find the Distance of one Object from another, where a River divides them, without using the Theodolite.

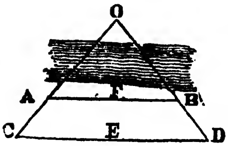


FIG. 5.

Let O and A (fig. 5) be the given objects, and AO the distance required. From A draw AB at *any* angle to AO, and produce OA to C, measuring AC about one-third length of AB; from C measure CD, parallel to AB, OBD being in one straight line. To make CD parallel to AB, at A and B erect equal perpendiculars, which can be done either by a cross-staff or by means of the unit proportion of the sides of a right-angled triangle, viz. 3, 4, and 5, or any multiples of them whatever. Then

$$AO = \frac{AB \cdot CA}{CD - AB}.$$

If the perpendicular distance OF be required, then this can be obtained by the formula—

$$OF = FE\,\frac{OA}{AC}.$$

Or we can so select the point A as to have OAC at right angles to the river, and make AB and CD at right angles to OAC, when AO, given by the foregoing formula, gives, of course, the required perpendicular distance.

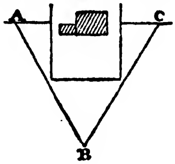


FIG. 6.

To continue the Measurement and Direction of a given Line when any Obstacle stands in the way, which cannot be crossed but can be avoided by going to the right or left.

At any point, A (fig. 6), on the given line, AC, take an angle with the given line, of 120° if you would turn to the left, or 240° if to the right, as in the present case, and proceed measuring to B till an angle of 60° made with this line towards the first line, AC, will carry you clear of the obstacle. Take this angle, ABC, 60°, and measure BC the same distance as AB: the point C will then be on the given line, and AC will be equal to AB or BC. By taking an angle of 240° with the line BC, the range of the line can be continued.

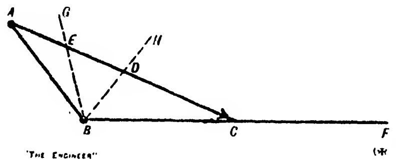


FIG. 7.

Line and Angle Bisector.

The following simple contrivance, based on Euclid, Proposition 3, Book 6, and devised by H. R. Kempe, may be found useful to draughtsmen.

AB and AC are two links hinged at A, AB being pivoted at B; AC is twice as long as AB. E and D are points on AC such that D is midway between A and C, and E is midway between A and D.

If, now, the end C of the link AC be slid along BF, then HB will always bisect GBF, and EB will always equal $\frac{BC}{2}$.

General Remarks.

1. Avoid taking bearings and distances along a circuitous boundary line like *abc* (fig. 8), but run the straight line *ac*, and at right angles to it measure offsets to the crooked line. 2. If it is desired to survey a straight line from *a* to *c*, but the instrument cannot be directed precisely towards *c*, on account of intervening obstacles, first run a trial line, *am*, as nearly in the proper direction as can be guessed. Measure *mc*, and calculate a distance, *y*, say, so that *am* is to *mc* as is 100 ft. to *y*. Lay off *ao* equal to 100 ft., and *os* equal to *y*, and run the final line *asc*. Or, if *mc* is quite small, calculate offsets like *os* for every 100 ft. along *am*. 3. When *c* is visible from *a*, but the intervening ground difficult to measure along on account of marshes, &c., extend the side *ya* to good ground at *t*; then, making the angle *ytd* equal to *yac*, run the line *tn* to that point, *d*, at which the angle *ndc* is found by trial to be equal to the angle *atd*. It will rarely be necessary to make more than one trial for this point *d*;

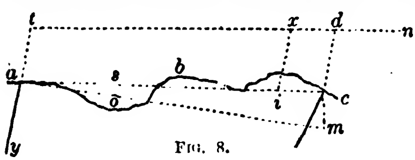


FIG. 8.

for, suppose it to be made at x, see where it strikes ac at i; measure ic, and continue from x, making $xd = ic$. 4. In the case of a very irregular piece of land, or a lake (fig. 9), surround it by straight lines. Survey these, and at right angles to them measure offsets to the crooked boundary. 5. When between two objects, m and n (fig. 10), and it is wished to place oneself in range with them, lay a straight rod, cb, on the ground, and point it to one of the objects, m; then go to the end c, and see if it points to the other object; if it does not, then, by successive trials, find the position, ed, in which the rod points to both objects, and consequently is in range with them. If no rod is at hand, two stones will answer, or two chain-pins. A plumb-line (a pebble tied to a piece of thread) will add to the accuracy of ranging the rod, or stones, &c.

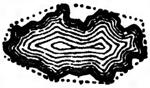


FIG. 9.

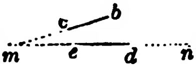


FIG. 10.

TO MEASURE THE HEIGHT OF AN OBJECT.

(1) When the Base is Accessible.

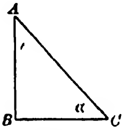


FIG. 11.

Measure any distance from the base of the object as nearly equal as possible to the height, and take the angle of elevation by the theodolite.

Let BA (fig. 11) be the object; measure BC, and take the angle a; then

$$BA = BC \tan a.$$

(2) When the Object is Inaccessible.

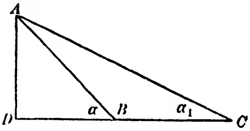


FIG. 12.

If the object be inaccessible, so that DB *cannot be measured*, range DB (fig. 12) on to C, and taking BC nearly equal to DB, measure BC, and at B and C take the angles of elevation, a and a_1; then

$$AD = BC \tan a_1 \left(\frac{\tan a}{\tan a - \tan a_1}\right).$$

Chaining on Slopes.

A = angle of slope with horizon. L = length of line chained on slope.
l = length of line reduced to the horizontal.

$$l = L \cos A.$$

TABLE OF DEDUCTIONS OR ADDITIONS TO BE MADE PER 100 FEET IN CHAINING OVER SLOPING GROUND

In order to reduce the inclined measurements to horizontal ones.

Slope in Deg.	Deduct Feet.	Rise in 100 ft. Hor.	Slope in Deg.	Deduct Feet.	Rise in 100 ft. Hor.	Slope in Deg.	Deduct Feet.	Rise in 100 ft. Hor.	Slope in Deg.	Deduct Feet.	Rise in 100 ft. Hor.
¼	·001	·436	5¼	·420	9·189	10¼	1·596	18·08	15¼	3·521	27·28
½	·004	·873	½	·460	9·629	½	1·675	18·53	½	3·637	27·73
¾	·009	1·309	¾	·503	10·07	¾	1·755	18·99	¾	3·754	28·20
1	·015	1·746	6	·548	10·51	11	1·837	19·44	16	3·874	28·67
¼	·024	2·182	¼	·594	10·95	¼	1·921	19·89	¼	3·995	29·15
½	·034	2·619	½	·643	11·39	½	2·008	20·35	½	4·118	29·62
¾	·047	3·055	¾	·693	11·84	¾	2·095	20·80	¾	4·243	30·10
2	·061	3·492	7	·745	12·28	12	2·185	21·26	17	4·370	30·57
¼	·077	3·929	¼	·800	12·72	¼	2·277	21·71	¼	4·498	31·05
½	·095	4·366	½	·856	13·17	½	2·370	22·17	½	4·628	31·53
¾	·115	4·803	¾	·913	13·61	¾	2·466	22·63	¾	4·760	32·01
3	·137	5·241	8	·973	14·05	13	2·563	23·09	18	4·894	32·49
¼	·161	5·678	¼	1·035	14·50	¼	2·662	23·55	¼	5·030	32·98
½	·187	6·116	½	1·098	14·95	½	2·763	24·01	½	5·168	33·46
¾	·214	6·554	¾	1·164	15·39	¾	2·866	24·47	¾	5·307	33·95
4	·244	6·993	9	1·231	15·84	14	2·970	24·93	19	5·448	34·43
¼	·275	7·431	¼	1·300	16·29	¼	3·077	25·40	¼	5·591	34·92
½	·308	7·870	½	1·371	16·73	½	3·185	25·86	½	5·736	35·41
¾	·343	8·309	¾	1·444	17·18	¾	3·295	26·33	¾	5·882	35·90
5	·381	8·749	10	1·519	17·63	15	3·407	26·79	20	6·031	36·40

The foregoing table gives deductions or additions to be made every 100 ft. as actually chained along sloping ground in order to reduce the sloping measurements to horizontal ones. Even when it is so nearly level that the eye cannot detect the slope, an over-measurement of an inch or two in 100 ft. may readily occur. It is plain that if we measure all the undulations of the ground we shall get greater totals than if we measure horizontally, as is *supposed* always to be done; but since few surveyors pretend to measure horizontally until the slope becomes apparent to the eye, their lines are usually too long by from 1 to 2 inches in 100 ft. To counteract this to some extent chains are frequently made from 1 to 2 inches longer than 100 ft.; and for ordinary purposes the precaution is a good one. When greater accuracy is required the chainmen should be attended by a third person, with a rod and slope-level, for taking the inclinations of the ground. These deductions being made, the remainder will be the actual horizontal distance.

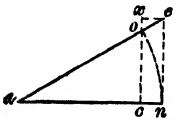


FIG. 13.

For example, in fig. 13. each 100 ft. *ao* measured up or down the slope *ae* plainly corresponds to the shorter horizontal distance, *ac*, the difference or deduction being *cn*. Taking *ao* as radius, then *ac* is the cosine, and *cn* the versed sine of the angle *ean* of the slope. Therefore *ae* multiplied by the nat. cosine of the angle *ean* gives the reduced horizontal distance, *ac*, which, taken from *ao*, gives the deduction *cn* of our table.

But if, while chaining along the slope *ae*, we wish to drive stakes that shall correspond with horizontal distances, *an*, of 100 ft., it is evident that we must add *cn* to each 100 ft. *ao*, as shown at *xe*, and the stake must be driven at *e* instead of at *o*. Observe that $xe = cn$ must be measured horizontally.

When the ground is very sloping all this calculation may be avoided, where great accuracy is most required, by actually holding the chain horizontal, as nearly as can be judged by eye, and finding, by means of a plumb-line, where its raised end would strike the ground. A whole chain at a time cannot be measured in this way, but shorter distances must be taken, as the ground requires—at times, on very steep ground, not more than 5 or 10 ft.

The following short rules may be used for finding the approximate correction to be made :—

(1.)

Square the number of degrees in the slope, multiply by $1\frac{1}{2}$, and obtain correction in hundredths of a foot.

Example.—10° slope. $10^2 \times 1\frac{1}{2} = 150$, or 1·50 ft. correction.

(2.)

Square the rise in feet per 100 ft. horizontal, and divide by 2; this gives correction in hundredths of a foot.

Example.—10 ft. rise in 100. $10^2 \div 2 = 50$, or 0·50 ft. correction.

SCALES FOR SURVEYS.

Ordnance Survey Scales.

VARIOUS SCALES OF MAPS OF GREAT BRITAIN.

Natural Scales.	Inches to one Statute Mile	Statute Miles to one Inch.	Chains " to " one Inch.	REMARKS.
$\frac{1}{500}$	126·720	0·0079	$0{\cdot}6\dot{3}\dot{1}$	Town maps.
$\frac{1}{528}$	120·000	$0{\cdot}008\dot{3}$	$0{\cdot}\dot{6}$	
$\frac{1}{1056}$	60·000	$0{\cdot}01\dot{6}$	$1{\cdot}\dot{3}$	
$\frac{1}{2500}$	25·344	0·0395	$3{\cdot}1\dot{5}\dot{6}$	Cadastral, or parish, maps of the United Kingdom.
$\frac{1}{10560}$	6·00	$0{\cdot}1\dot{6}$	$13{\cdot}\dot{3}$	County maps of the United Kingdom and special maps.
$\frac{1}{126720}$	0·50	2·0	160	Contoured, 50 ft., 100 ft., every 100 ft. to 1,000 ft., above that every 250 ft. Roads coloured by classes; water blue. Normal sheets 27 in. by 18 in.
$\frac{1}{253440}$	0·25	4·0	320	A motoring map. Eleven sheets cover England and Wales.
$\frac{1}{15840}$	4·000	0·25	20·0	Special index.
$\frac{1}{63360}$	1·000	1·0	80·0	General map of United Kingdom, and special maps.

Small-Scale Maps of Great Britain.

Aeronautical Maps: 1:500,000. Intended for general navigation use. Railways emphasised; main roads are shown, road junctions specially marked.

'*Civil Air*' *Maps:* Quarter-inch, covering England and Wales in 12 sheets. Railways, main roads, woods, inland water and high ground strongly emphasised. Golf links and race courses clearly marked.

Ten Mile Maps: (1) The layered map. The whole of Great Britain is covered in two sheets. (2) Road map, in two sheets, showing Ministry of Transport road numbers.

United Kingdom: 1:1,000,000. 15·782 miles to an inch. Two sheets. 24½ × 34½ inches each. Orkney and Shetland Islands shown on sheet 1. Hills shaded in brown.

One Inch Maps: Popular edition. Water blue; main roads red; secondary roads brown; minor roads uncoloured. *Contours* at 50 ft. intervals; woods green. Normal size of sheet 27 × 18 inches.

Maps of the India Survey Department.

One mile to one inch: Topographical maps; about one-eighth of the total number has been published (1914).

Four miles to one inch: Prepared in sheets of one degree by one degree.

Thirty-two miles to one inch: The outline engraving of this new map of India and adjacent countries is completed. The hills are shown in the 'layer' system. (*a*) A new edition of the railway, canal, and road map of India. (*b*) A new map of India and adjacent countries, hills shown on the 'layer' system. (*c*) A new edition of the map showing railway stations.

Dip of Horizon.

d = dip of horizon, in seconds. h = height of observer's eye, in feet.

s = distance of horizon, in statute miles. n = distance of horizon, in nautical miles.

$d = 57{\cdot}4\sqrt{h}$; approximate, varying with temperature. $h = {\cdot}666s^2 = {\cdot}86n^2$.

Table of Dip and Distance of Horizon at Various Heights.

h.	s.	n.	d.	h.	s.	n.	d.
Feet.	Miles.	Miles.	′ ″	Feet.	Miles.	Miles.	′ ″
5	2·739	2·411	2 8·35	80	10·959	9·644	8 33·4
10	3·874	3·409	3 1·51	90	11·624	10·229	9 4·54
15	4·745	4·176	3 42·31	100	12·253	10·783	9 34
20	5·480	4·822	4 16·7	150	15·007	13·206	11 43
25	6·126	5·391	4 47	200	17·329	15·249	13 31·76
30	6·711	5·906	5 14·39	300	21·223	18·676	16 34·2
35	7·249	6·379	5 39·58	400	24·507	21·566	19 8
40	7·749	6·819	6 3·03	500	27·399	24·111	21 23·5
45	8·219	7·233	6 25·05	1,000	38·749	34·099	30 15·1
50	8·664	7·624	6 45·88	2,000	54·799	48·223	42 47
60	9·491	8·352	7 24·62	3,000	67·115	59·061	51 23·9
70	10·252	9·021	8 0·24	4,000	77·498	68·198	60 30·3

To reduce a Measured Base to the Level of the Sea.

Let r represent the radius of the earth corresponding to the base b at the level of the sea, and $r + a$ the radius referred to the level of the measured base, B; then

$$r + a : r :: B : b, \quad \text{or,} \quad b = B \times \frac{r}{r+a}, \quad \text{and,} \quad B - b = B - B\frac{r}{r+a} = B\left(1 - \frac{r}{r+a}\right)$$

$$= B\left(\frac{a}{r+a}\right) = B\left(\frac{\frac{a}{r}}{1 + \frac{a}{r}}\right) = \frac{Ba}{r}\left(\frac{1}{1+\frac{a}{r}}\right) = \frac{Ba}{r}\left(1 - \frac{a}{r} + \frac{a^2}{r^2} - \&c.\right) = B\left(\frac{a}{r} - \frac{a^2}{r^2} + \frac{a^3}{b^3} - \&c.\right).$$

But the radius of the earth being very great in comparison with the difference of level, a, we have the correction, $B - b$, or d, say, sufficiently accurate by retaining only the first term. Hence

$d = \frac{Ba}{r}$ is the value to be deducted from the measured base.

Spherical Excess.

$$E, \text{ in seconds} = \frac{S}{r^2 \sin 1''} = \frac{ab \sin C}{2r^2 \sin 1''},$$

S = the area of the triangle. r = the radius of the earth.

$$S = \frac{ab \sin C}{2} = \sqrt{s(s-a)(s-b)(s-c)}, \quad s \text{ being} = \frac{a+b+c}{2}.$$

Between latitudes 45° and 25° the spherical excess amounts to about 1′ for an area of 75·5 square miles.

Hence, if the area in square miles be known, a close approximation to the spherical excess will be had by dividing the area by 75·5. Log mean radius of the earth in yards = 6·8427917. If the three angles of a triangle are assumed to have been equally well determined, the previous determination of the spherical excess is not necessary for the calculation of the sides, though it will be required for estimating the relative value of the observations. For the sides of a spherical triangle may be computed as if they were rectilineal if one-third the excess of the sum of the three angles above 180° is deducted from each of the three observed angles, in which case

$$\text{side } b = \text{side } a \sin\left(B - \tfrac{1}{3}E\right) \div \sin\left(A - \tfrac{1}{3}E\right).$$

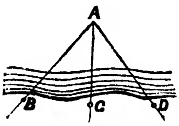


FIG. 14.

Surveying Soundings.

To determine the position of any sounding, A (fig. 14), when afloat, take simultaneously the bearings of three known objects on shore, B, C, D, plot the angles on tracing-paper, and move them on plan until the lines cut the points B, C, D.

Another method.—B, C, and D being known points on shore, and the angles BAC, CAD having been measured—

Double the angle BAC, and deduct this doubled angle from 180°. Halve the remainder, and set off from B and C angles, each equal to the half remainder—that is, BCE, CBE; then BE = EC. With E as a centre and radius EC, describe the circle BCA. Then the point A will be in the circumference of this circle.

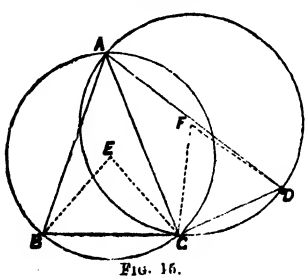


FIG. 15.

In like manner, double the angle CAD and proceed as before; the point A will be in the circumference of the circle CAD. Being in the circumference of the two circles, the point A will therefore be at the intersection of the two.

Suppose the angle BAC to have been 40°, CAD 30°, the double of BAC is 80°; then 180° − 80° = 100°, which divided by 2 gives 50°. Set off from B and C angles equal to 50°. In like manner, 2CAD = 60°, and 180° − 60° = 120°; half 120° = 60°; from C and D set off 60°. With E and F as centres (fig. 15), and radii EB, FC, describe the two circles, whose intersection determines the position, A, of the observer.

To reduce a Sounding to Low Water.

h = vertical rise of tide, in feet. h' = height of tide above low water, in feet, at the time t.
= time between high and low water. t' = time from time of sounding to low water, in hours

$$\frac{h}{2}\left(1 \mp \cos \frac{180t'}{t}\right) = h'; \quad - \cos \text{ when } \frac{180t'}{t} < 90^\circ, \text{ and } + \cos \text{ when } > 90^\circ.$$

EXAMPLE.—Low water occurring at 3.45, and high water at 10.15 P.M., a sounding taken at 5.30 P.M. was 18·25 feet; what was the depth at low water, the vertical rise being 10 feet?

$$h = 10\,\textit{feet};\ t = 5h.\ 30m. - 3h.\ 45m. = 1h.\ 45m. = 1{\cdot}75\ \textit{hours}.$$
$$t = 10h.\ 15m. - 3h.\ 45m. = 6h.\ 30m. = 6{\cdot}5\ \textit{hours}.$$

Then

$$\frac{10}{2}\left(1 \mp \cos \frac{180 \times 1{\cdot}75}{6{\cdot}5}\right) = 5(1 - 48^\circ\ 27') = 5 \times (1 - {\cdot}662) = 1{\cdot}69\,\textit{feet}.$$

Sounding	18·25 *feet.*
Reduction	1·69
	16·56

SURVEYING INSTRUMENTS.

ADJUSTMENT OF THE THEODOLITE.

(*R. T. Hancock.*)

The first and most important adjustment consists in correctly levelling the instrument with its vertical axis pointing truly to the zenith, every time it is set up. Other errors in adjustment can be corrected or compensated by special methods of observation, but correct results are impossible if the theodolite is out of level. This adjustment must precede all others.

Set the instrument up firmly with its legs well spread and bring it approximately level by means of the base adjusting screws, with the aid of the bubble tubes on the vernier plate. Place the telescope as nearly horizontal as possible, with the vertical arc clamped to zero if desired, and bring it over one pair of the base levelling screws, or parallel to a pair in the case of a tribrach base, and observe the position of one end of the bubble of the telescope level in relation to the gradations on the tube. Select and use one end throughout, say, the end of the bubble nearest the eyepiece. Revolve the upper part of the theodolite through 180°, bring the bubble back half way, if it has moved, by means of the base levelling screws. Check by turning the instrument back to its original position. If the bubble remains in its mean position, turn the upper part of the instrument through 90°, so that telescope lies over the other pair of levelling screws, or the remaining one of the tribrach, and bring the bubble to the same position by these. Finally check by turning the instrument slowly through 360°, and see that the bubble remains unmoved. If not, repeat adjustment from the beginning. The bubble may not finally occupy the centre of its tube: this is unimportant. What is important is that it should not move when the instrument is revolved about its vertical axis. If the bubbles of the levels on the vernier plate are not well centered when the instrument is thus levelled, it will be convenient to make them so now by means of the turret screws at the ends of the cases, so that a close approximation to true levels may be obtained when setting up on future occasions. It should make no difference in levelling up whether the two plates revolve together or whether the lower is clamped and the upper free. If it does, it shows that the cones require regrinding.

Adjustment for parallax consists in moving the eyepiece in or out, altering the focus of the object glass to correspond, until the crosshairs of the diaphragm do not appear to move relative to the point viewed when the eye is moved slightly from side to side of the lens opening. Care should be taken not to revolve the eyepiece tube in its casing, as this affects the optical centering of the system of lenses. For this reason the lens cell and its casing are usually provided with corresponding marks by the maker.

The transit axis upon which the telescope swings should be accurately at right angles with the vertical axis of the instrument. The adjustment of this is effected by sliding bearings at the top of the standards, but in cheaper instruments no provision for adjustment in this respect is made, the makers supposedly securing it once and for all by correctly proportioning the height of the standards. Direct the telescope to some high object, such as the finial of a spire. Depress the telescope and mark on the ground the point where the intersection of the hairs falls. Transit the telescope, sight up again, and mark where the intersection falls on the ground the second time. Raise or lower one bearing so that the intersection falls one-quarter of the distance between the marks from the second mark. Check, and repeat if necessary.

The collimation line may be defined as a line joining the optical centre of the object glass and the intersection of the crosshairs. To make it at right angles to the axis on which the telescope swings, set up, level, and sight to a well-defined mark about 100 feet away; transit the telescope, and mark where the intersection falls in the opposite direction, at about the same distance away and on the same level as near as may be. Revolve the plates through 180° about, and sight to the original mark. Again transit, and mark the new point of intersection, if altered, near the former one. Measure a point one-fourth of the distance between the two from the latter, and move the diaphragm carrying the crosshairs by means of the adjusting screws holding it in the telescope until the intersection covers the measured mark. It will be necessary to loosen all four screws to do this, but care should be taken not to move the diaphragm up or down more than can be helped, and to keep the crosshairs vertical and horizontal. Check by transiting back again to the original mark, which will now not fall under the crosshairs; bring it under the intersection by the slow motion tangent screw, and transit back again. The intersection should now fall half way between the two original marks.

Having performed the foregoing adjustment we now have the vertical axis of the instrument pointing truly to the zenith, the horizontal axis on which the telescope swings truly at right angles to the vertical axis, and the line of collimation truly at right angles to the horizontal axis. The collimation line then lies wholly in a vertical plane cutting the horizontal axis at right angles, but does not necessarily lie parallel to the axis of the level on the telescope in that plane. It can be made to do so by altering the lie of one or the other, and in order to avoid disturbing the adjustment just completed, it is better to bring the axis of the telescope-level parallel to the collimation line as it now lies. This is why it was recommended not to disturb the up-and-down position of the collimation line more than can be helped, so as to avoid the necessity for much alteration in the telescope level. This adjustment is not really required for the theodolite, unless it is intended to use it occasionally as a level, and the surveyor who desires to effect it can proceed as described below for the level, except that after setting up beyond the levelled pegs, the telescope is lowered or raised by the slow-motion tangent screw of the vertical arc till the crosshairs read the same on both pegs; the bubble of the telescope level is then brought to the centre of its run by altering the capstan-headed screws at each end of its casing.

Subtense Measurements. Tacheometers.

It is quite common at present to place a pair of webs—or much better, a pair of points—in the diaphragm of a theodolite or level, these being at such a distance from each other that they will by vision correspond to a measurement (upon a distant levelling staff), which is proportional to the base of the subtense angle, so as to show thereby, in the space between the lines or points, the true distance of the instrument from the staff. FIG. 16 shows the form these points appear in focus of the eye piece. By the employment of this system chain measurements are entirely dispensed with, and the system has the merit of a simple sight measurement, as upon a rule, without any calculation. Theoretically it is scarcely as exact as the work of the Omnimeter, as there is great difficulty of exact adjustment of the points without the use of complicated optical means, but practically, as testified in some recent work by the Surveyor-General of Lagos and Nigeria, a good instrument in correct adjustment may be read within less than ·001 of total distance, which is nearer than is possible by chaining over uneven ground. One merit over the Omnimeter is that the instrument is not touched by the hand during the observation of the points, cut by vision upon the staff, which give the distance.

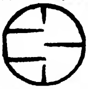


FIG. 16.

THE DUMPY LEVEL.

This is a perfect knock-about instrument. If stations are taken at equal distances apart, say one chain, this form of level will act correctly although out of adjustment. Thus with the level

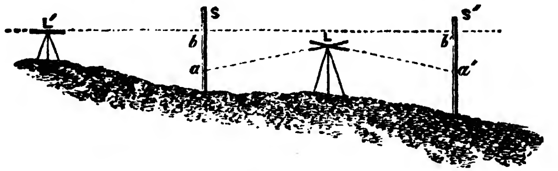


FIG. 17.

at L (fig. 17), place a staff vertically at one chain distance from the instrument at S. The inaccurate instrument will read the staff at *a*. Now place the staff vertically at S′. It will read at *a′*. The distances and verticality being equal, *a* and *a′* will be level. If we now place the level at some distant position nearly in line L′ and adjust the level by the bubble screw to read equal distances above *a* and *a′* and *b* and *b′*, the level must be true.

Adjustment of the Level.

(*R. T. Hancock.*)

Select a fairly level piece of ground, set up the instrument, carefully levelled up as described for the theodolite (p. 257), and drive a peg at a distance of 100 feet or so away. Hold the levelling staff on the peg and note the exact reading. On the opposite side of the instrument, and exactly the same distance away, set another peg, driving it down till the staff held on it reads exactly the same as on the first peg. The tops of the pegs are then on the same level, independent of instrument error. Set up on the prolongation of the line beyond one of the pegs, level up carefully, and make the crosshairs read the same on both pegs by raising or lowering the diaphragm cell by means of the screws holding it in the telescope tube. The collimation line is now horizontal.

It is not necessary to level the pegs if the difference in level is noted, and the crosshairs are altered to read the same difference. To facilitate the adjustment in this case, set up beyond the pegs at a distance which bears some simple relation to the distance between the pegs, make a sketch of the position of instrument and staffs, insert readings and distances, and calculate the amount it is necessary to alter the reading on either staff (by moving the crosshairs), on the principle of similar triangles.

The telescope bubble may not finally occupy the centre of its tube when the collimation has been put horizontal as described. This is unimportant, as the instrument can always be correctly levelled up if the procedure as described for the theodolite is followed, in fact the central position of the bubble should never be relied on as a test for levelling up, even if it has once been secured by adjustment. It may be brought to the centre of its run by adjusting the screws holding the level to the telescope, care being necessary not to disturb the instrument.

The Plane Table.

This instrument is now generally used in new countries and in our colonies, for filling in details of work with the theodolite. It has the great merit that an intelligent native without technical education can be instructed to do valuable detail work with it. In its simplest form it consists of a sighted rule placed on a small board supported by a light tripod, the work being performed on the surface of paper. Many civil engineers find its work extremely convenient, because this work being made most definite by sketches upon the paper used, from these sketches they have a certainty of position in making maps from the same; so that they employ plane tables with all the refinement of means of observation possessed by the theodolite.

Prismatic Compass.

Details of small surveys or filling-in work are made by this compass, which is sighted for observation of direction, so that the eye may also see the compass ring by reflection through a prism at the instant of sighting. A reflector is commonly fixed to the forward sight so that distant objects may be seen up or down hill. The instrument is generally held in the hand for work, but where greater refinement is desired it is supported on a tripod stand. 2½ in. or 3 in. instruments are used by hand; 3½ in., 4 in., and 4½ in. are each set on a stand.

To Find the Variation of the Compass.

1. By the Pole Star (Polaris).

The pole star is on the meridian, either above or below the pole, when a vertical plane passes through it, and the bright star at the root of the tail of the Great Bear ε Ursæ Majoris (Alioth), Thus, during the twenty-four hours, two opportunities will be presented of finding the pole star on the meridian. Set up a fine plumb-line in a convenient place, free from currents of air, and the observer, placing himself behind the thread, waits until the moment when the two stars are hidden by the thread. If the bearing of the pole star be then read, it will show the variation of the compass. But as it is not very easy to read a compass at night, it is best to fix a stick with a light on it in the line of the pole star, and as far off as possible, the line joining the observer's position, and the light is the meridian line, and its bearing may be read in the morning.

2. By a Single Altitude of the Sun.

1. Compare watch with chronometer to get correct mean time.
2. Note readings of barometer and thermometer.
3. Adjust theodolite and note index error.
4. Set up instrument at . . . . Δ station; make the temporary adjustments, and set instrument at 360° on . . . . Δ station.

5. About three hours from 12 o'clock, morning or afternoon, unclamp upper horizontal plate and vertical arc. See that the dark-eye piece is down, and move upper part of instrument till the sun is in the field of object glass. Clamp upper horizontal plate and vertical arc, and with the two tangent screws follow (morning observation) the sun's movement till the sun's upper limb just touches the intersection of the cross-hairs. Note the time by watch, and read on horizontal arc the azimuth, and on vertical arc the altitude of $\overline{\odot}$, sun's upper limb.

6. Unclamp horizontal plate, and either with a slight motion, or, if plate be clamped, wholly with tangent screw, follow the sun's motion in azimuth till the sun's lower limb ($\underline{\odot}$) becomes tangent to the intersection of cross-hairs, the vertical arc being untouched. Note the time and the reading on the horizontal arc.

7. The mean of the times and the mean of the azimuths will give the time and azimuth at which the vertical reading of the arc denotes the apparent altitude of the sun's centre $\odot$.

8. Correct apparent altitude, $\odot$.

Correction 1. For index error of instrument. Correction 2. For refraction, modified by barometric and thermometric readings; corrections to be found in tables in Frome's 'Surveying,' Drayson, or Chambers. Correction 3. For parallax. To be found in tables in 'Nautical Almanac,' and modified to noted altitude by formula:—Sin par. in alt. = sin hor. par. × cos. alt.

9. Apparent altitudes thus corrected gives true alt. $\odot$.

10. The true alt. $\odot$ subtracted from 90° gives the zenith distance ZS.

11. As the altitude of the elevated pole is equal to the latitude of the place, 90° minus latitude = zenith distance of the pole, viz. PZ.

12. The declination of the sun is given in the 'Nautical Almanac' for each day at apparent noon. The variation of declination for each hour is also given, and the direction of declination, N. or S. Hence the declination at the time of observation is obtained. If the time of observation be 9 A.M., and declination increasing, the correction for three hours' variation must be subtracted from the declination at apparent noon, *vice versa* if the declination be decreasing.

13. Then we have the three sides of the astronomical triangle to ascertain the angle Z, which is the measure of the angular deviation of the sun from the true meridian.

14. Compute the angle Z by the formula

$$\cos\frac{Z}{2} = \sqrt{\frac{\sin S \sin (S - PS)}{\sin PZ \sin ZS}}$$

where $S = \frac{1}{2}$ sum of the three sides of the triangle.

15. The magnetic bearing of the Δ station from which the sun's azimuth was taken should be noted two or three times with great care. If possible a stand to be used for the compass.

16. Then, knowing the sun's azimuth from a station and the direction of that station as regards the magnetic north, we get the deviation of the sun from the magnetic north. The computed angle, Z, is the deviation of the sun from the true north, and the difference of the two deviations is the variation of the compass.

17. Note that for afternoon observations the computed angle, Z, is to be subtracted from 360° before comparison with sun's magnetic bearing.

Magnetic Elements.

At Abinger Magnetic Station, near Leith Hill, Surrey (in connection with the Royal Observatory, Greenwich).

Mean values October 1948.

Declination, West, 9° 33′. Inclination (or dip) 66° 46′. Horizontal intensity, 0·1858 gauss. Vertical intensity, 0·4327 gauss. Average annual decrease in declination, 8′ E.

Declination at Greenwich, 14·0′ less than at Leith Hill. On any particular day, and even at different times of the same day, the departures from these values may be considerable. It is meaningless to give estimated values to decimals of a minute or five significant figures of intensity.

For other places in the British Isles, very rough values may be found by increasing 0·5° for each degree of longitude, W., and increasing 0·3° for each degree of latitude, N.; but local variations are considerable.

To Determine the South by Means of a Watch.

Point the hour hand of the watch at the sun, then half-way round (see note *a*) between the hour hand and 'XII' on the dial points 'south.' Thus, suppose it to be 8 P.M., point the hour hand at the sun, then the 'IIII' of the dial points to south. In southern latitudes, to find 'north,' point 'XII' at the sun, then half-way round between 'XII' and the hour is due 'north.'

This method is accurate only about the times of the equinoxes. The allowance for the actual times of the sun's southing (see Whitaker's Almanack) increase to and decrease from about 15 minutes at the times of the solstices.

(*a*) The half-way round is clockwise in the forenoon and counterclockwise in the afternoon.

To Determine a True East and West.

Set up a *vertical* rod (any size) on a flat floor. Describe a circle round the base about twice (this varies with the latitude) the height of the rod. Some time in the forenoon mark the point where the shadow of the rod just touches the circle, and repeat this in the afternoon on the other side of the circle; mark this point also. A line through these points will be true east and west.

As a guide to the true north, the following notes as to the whereabouts of the pole star at convenient hours in the evening or night will be useful, and correct enough for the purpose for the next hundred years.

The times given (Local mean time) are for the first day of each month, and become about 4 minutes (3 mins. 59·34 secs.) *less* every day.

Month.	1° W.	True N.	1° E.	Month.	1° W.	True N.	1° E.
January .	9·45	6·30	..	July . .	..	..	9·45
February .	7·45	..	..	August .	..	..	7·45
March . .	11·15	..	..	September .	..	..	11·15
April . .	9·15	12·30	..	October .	..	12·30	9·15
May . .	7·15	10·30	..	November .	..	10·30	7·15
June . .	..	8·30	11·45	December .	..	8·30	5·15

Equation of Time.

TABLE SHOWING TIME TO BE ADDED TO OR SUBTRACTED FROM SUNDIAL TIME IN ORDER TO OBTAIN TRUE OR MEAN TIME. (LOCAL.)

Date.	Minutes (approximate).	Date.	Minutes (approximate).	Date.	Minutes (approximate).	Date.	Minutes (approximate).
Jan. 1	+ 4	Mar. 28	+ 5	Aug. 9	+ 5	Oct. 27	−16
„ 3	+ 5	April 1	+ 4	„ 15	+ 4	Nov. 15	−15
„ 5	+ 6	„ 4	+ 3	„ 20	+ 3	„ 20	−14
„ 7	+ 7	„ 7	+ 2	„ 24	+ 2	„ 24	−13
„ 9	+ 8	„ 11	+ 1	„ 28	+ 1	„ 27	−12
„ 12	+ 9	„ 15	0	„ 31	0	„ 30	−11
„ 15	+10	„ 19	− 1	Sept. 3	− 1	Dec. 2	−10
„ 18	+11	„ 24	− 2	„ 6	− 2	„ 5	− 9
„ 21	+12	„ 30	− 3	„ 9	− 3	„ 7	− 8
„ 25	+13	May 13	− 4	„ 12	− 4	„ 9	− 7
„ 31	+14	„ 29	− 3	„ 15	− 5	„ 11	− 6
Feb. 10	+15	June 5	− 2	„ 18	− 6	„ 13	− 5
„ 21	+14	„ 10	− 1	„ 21	− 7	„ 16	− 4
„ 27	+13	„ 15	0	„ 24	− 8	„ 18	− 3
Mar. 4	+12	„ 20	+ 1	„ 27	− 9	„ 20	− 2
„ 8	+11	„ 25	+ 2	„ 30	−10	„ 22	− 1
„ 12	+10	„ 29	+ 3	Oct. 3	−11	„ 24	0
„ 15	+ 9	July 5	+ 4	„ 6	−12	„ 26	+ 1
„ 19	+ 8	„ 11	+ 5	„ 10	−13	„ 28	+ 2
„ 22	+ 7	„ 28	+ 6	„ 14	−14	„ 30	+ 3
„ 25	+ 6			„ 19	−15		

Length of a Degree of Longitude and Latitude.

Lat.	Degree of Longitude.		Degree of Latitude.		Lat.	Degree of Longitude.		Degree of Latitude.	
$\phi°$	Stat. Miles.	Naut. Miles.	Stat. Miles.	Naut. Miles.	$\phi°$	Stat. Miles.	Naut. Miles.	Stat. Miles.	Naut. Miles.
0	69·174	60·000	68·711	59·598	45	48·996	42·503	69·063	59·895
1	69·162	59·991	68·711	59·598	46	48·136	41·752	69·077	59·916
2	69·132	59·964	68·712	59·599	47	47·261	40·993	69·089	59·926
3	69·080	59·918	68·713	59·600	48	46·372	40·222	69·101	59·937
4	69·006	59·854	68·715	59·602	49	45·469	39·439	69·113	59·947
5	68·911	59·774	68·718	59·604	50	44·552	38·643	69·125	59·958
6	68·798	59·674	68·720	59·606	51	43·621	37·836	69·137	59·968
7	68·661	59·557	68·723	59·609	52	42·677	37·017	69·149	59·978
8	68·505	59·420	68·725	59·611	53	41·719	36·186	69·160	59·988
9	68·327	59·265	68·730	59·615	54	40·749	35·345	69·172	59·998
10	68·130	59·094	68·734	59·618	55	39·766	34·492	69·183	60·008
11	67·912	58·905	68·739	59·622	56	38·771	33·629	69·195	60·018
12	67·673	58·698	68·744	59·627	57	37·765	32·777	69·206	60·028
13	67·413	58·473	68·749	59·631	58	36·746	31·896	69·216	60·037
14	67·133	58·230	68·755	59·637	59	35·716	30·979	69·227	60·046
15	66·832	57·969	68·761	59·642	60	34·675	30·076	69·238	60·056
16	66·511	57·690	68·767	59·647	61	33·623	29·163	69·248	60·064
17	66·172	57·396	68·774	59·653	62	32·561	28·242	69·258	60·073
18	65·810	57·080	68·780	59·658	63	31·488	27·312	69·268	60·082
19	65·428	56·751	68·789	59·666	64	30·406	26·374	69·277	60·089
20	65·028	56·404	68·796	59·672	65	29·315	25·427	69·287	60·098
21	64·608	56·040	68·804	59·679	66	28·215	24·479	69·296	60·106
22	64·167	55·654	68·813	59·687	67	27·106	23·511	69·303	60·112
23	63·709	55·260	68·822	59·695	68	25·988	22·594	69·313	60·121
24	63·228	54·843	68·830	59·702	69	24·862	21·565	69·321	60·128
25	62·731	54·411	68·840	59·710	70	23·729	20·582	69·329	60·135
26	62·212	53·961	68·849	59·718	71	22·589	19·598	69·336	60·140
27	61·676	53·496	68·859	59·727	72	21·441	18·597	69·343	60·147
28	61·119	53·013	68·869	59·736	73	20·287	17·596	69·350	60·153
29	60·550	52·520	68·879	59·744	74	19·126	16·590	69·357	60·158
30	59·957	52·005	68·890	59·754	75	17·960	15·562	69·363	60·164
31	59·347	51·476	68·901	59·763	76	16·788	14·561	69·368	60·168
32	58·718	50·933	68·911	59·772	77	15·611	13·540	69·374	60·174
33	58·071	50·372	68·923	59·782	78	14·429	12·515	69·379	60·177
34	57·409	49·795	68·935	59·792	79	13·242	11·486	69·383	60·181
35	56·726	49·203	68·946	59·802	80	12·051	10·453	69·387	60·185
36	56·027	48·597	68·957	59·812	81	10·857	9·417	69·391	60·188
37	55·312	47·976	68·968	59·822	82	9·659	8·378	69·394	60·191
38	54·579	47·232	68·980	59·832	83	8·458	7·336	69·397	60·193
39	53·829	46·583	68·992	59·842	84	7·254	6·292	69·400	60·196
40	53·064	46·027	69·004	59·853	85	6·049	5·217	69·402	60·198
41	52·282	45·348	69·016	59·863	86	4·842	4·199	69·404	60·200
42	51·483	44·655	69·028	59·874	87	3·632	3·151	69·405	60·200
43	50·670	43·950	69·041	59·885	88	2·422	2·102	69·406	60·201
44	49·840	43·230	69·053	59·895	89	1·211	1·051	69·406	60·201
45	48·996	42·503			90	0·000	0·000		

Length of a degree of longitude at $\phi°$ latitude = $\cos \phi_1°$ × 69·174 statute miles; where $\tan \phi_1° = b \div a \tan \phi°$; a = equatorial radius (3963·36 miles), and b = polar radius (3950·06 miles); $b \div a$ = 3950·04 ÷ 3963·36 = 0·99664.

Example: What is the length of a degree of longitude at 50° latitude?

tan 50° = 1·1917536, therefore,

$\tan \phi_1°$ = ·99664 × 1·1917536 = 1·1877493 = tan 49° 54′ 18″; cos 49° 54′ 18″ = ·6440569, therefore length of degree = ·6440569 × 69·174 = 44·552 statute miles.

Length of a degree of latitude at $\phi°$ latitude ($\phi°$ being at the middle of the arc) = 69·174 × ·99330 + ·01005 $\sin^2 \phi°$) = 68·711 + ·6952 $\sin^2 \phi°$ statute miles.

Example: What is the length of a degree of latitude between 30° and 31°? $\phi°$ = 30° 30′, sin 30° 30′ = ·5075384, therefore length of degree = 68·711 + (·6952 × ·5075384 × ·5075384) = 68·711 + ·179 = 68·890 statute miles. (*H. R. Kempe.*)

APPROXIMATE DISTANCE OF OBJECTS WHEN FIRST SEEN AT SEA, THE OBSERVER'S EYE BEING SUPPOSED AT SEA LEVEL.

Height. Feet.	Distance. Naut. Miles.	Height. Feet.	Distance. Naut. Miles.	Height. Feet.	Distance. Naut. Miles.	Height. Feet.	Distance. Naut. Miles.	Height. Feet.	Distance. Naut. Miles.	Height. Feet.	Distance. Naut. Miles.
1	1·1	22	5·4	43	7·5	120	12·6	300	19·9	520	26·2
2	1·6	23	5·5	44	7·6	125	12·9	310	20·2	540	26·7
3	2·0	24	5·6	45	7·7	130	13·1	320	20·6	560	27·2
4	2·3	25	5·7	46	7·8	135	13·4	330	20·9	580	27·7
5	2·6	26	5·9	47	7·9	140	13·6	340	21·2	600	28·1
6	2·8	27	6·0	48	8·0	145	13·8	350	21·5	620	28·6
7	3·0	28	6·1	49	8·1	150	14·1	360	21·8	640	29·1
8	3·2	29	6·2	50	8·2	160	14·5	370	22·1	660	29·5
9	3·4	30	6·3	55	8·5	170	15·0	380	22·4	680	29·9
10	3·6	31	6·4	60	8·9	180	15·4	390	22·7	700	30·4
11	3·8	32	6·5	65	9·3	190	15·8	400	23·0	720	30·8
12	4·0	33	6·6	70	9·6	200	16·3	410	23·2	740	31·2
13	4·1	34	6·7	75	9·9	210	16·7	420	23·6	760	31·7
14	4·3	35	6·8	80	10·3	220	17·0	430	23·8	780	32·1
15	4·4	36	6·9	85	10·6	230	17·4	440	24·1	800	32·5
16	4·6	37	7·0	90	10·9	240	17·8	450	24·4	820	32·9
17	4·7	38	7·1	95	11·2	250	18·1	460	24·7	840	33·3
18	4·9	39	7·2	100	11·5	260	18·5	470	24·9	860	33·7
19	5·0	40	7·3	105	11·8	270	18·9	480	25·2	880	34·1
20	5·1	41	7·4	110	12·1	280	19·2	490	25·5	900	34·5
21	5·3	42	7·5	115	12·3	290	19·6	500	25·7	920	34·9

Example: From what distance will a hill 300 feet high be visible to an observer 20 feet above the sea?

To an observer on the summit of the hill the horizon would be 21·2 miles distant, and to the observer in the example it would be 5·5 miles distant, hence two such observers facing one another would just see each other at a distance of 21·2 + 5·5 = 26·7 miles, which is therefore the distance the hill will be visible.

ASTRONOMY.

The Earth and Sun.

Equatorial radius of the earth = 6,378,200 metres (3963·30 miles) (*Helmert*) = 6,378,388 metres (3963·42 miles) (*Hayford*); mean value, 6,378,294 metres (3963·36 miles). Polar radius of the earth = 6,356,818 metres (3950·01 miles) (*Helmert*) = 6,356,909 metres (3950·06 miles) (*Hayford*); mean value, 6,356,864 metres (3950·04 miles.

Equatorial horizontal parallax of sun (angle under which the equatorial radius of the earth would appear at the sun's centre) = 8″·80. Mean distance of sun from earth $= \dfrac{3963{\cdot}23}{\text{sine } 8''{\cdot}80}$ $= \dfrac{3963{\cdot}23}{{\cdot}0000426664}$ = 92,890,000 miles. Maximum distance (1st July) = 93,950,000 miles. Minimum distance (1st Jan.) = 90,950,000 miles.

Angle under which the radius of the sun would appear at the earth's centre, at the earth's mean distance from the sun = 16′ 1″·18 (*Auwers*).

Radius of sun $= 3963{\cdot}23\ \dfrac{\text{sine } 16'1''{\cdot}18}{\text{sine } 8''{\cdot}8} = 3963{\cdot}23\ \dfrac{{\cdot}0045699}{{\cdot}000042664}$ = 424,500 miles.

None of the foregoing numerical values can be accepted as absolute, as authorities are not agreed on the subject.

Identification of Stars.

In stellar observations for time, latitude, or azimuth, the star selected for observation may be conveniently found from the following notes, which describe the relative positions of the larger stars in the constellations. The angular distance between two stars can be closely estimated by comparison with half the distance between the observer's zenith and the horizon—*i.e.* 45°.

(*a*) In the constellation *Ursa Major* the two well-known stars α and β (THE POINTERS) point directly to POLARIS, the Pole star. The latter also forms the end or tail of *Ursa Minor*.

(*b*) If the curve of the tail of *Ursa Major* be continued, about 30° along it will be found ARCTURUS (α *Bootes*).

(*c*) A line from POLARIS at right angles to the line from THE POINTERS gives 50° from the former, CAPELLA (*α Auriga*).

(*d*) A line from POLARIS through the last star but one of the tail of *Ursa Major* passes through SPICA, 30° beyond ARCTURUS.

(*e*) The POINTERS' line continued through POLARIS for 60° indicates THE GREAT SQUARE OF PEGASUS. The star in the corner opposite to the Pole is MARKAB.

(*f*) Midway between the SQUARE and POLARIS is the constellation *Cassiopeia* (five stars grouped like W).

(*g*) A diagonal through the SQUARE S.E. to N.W. produced 40° gives DENEB (*α Cygni*).

(*h*) 25° further and 10° to the right the same line gives VEGA (*α Lyræ*), a large white star.

(*j*) 35° south of DENEB and VEGA is ALTAIR (*α Aquilae*). This star is between two companions, and forms the apex of an isosceles triangle with DENEB and VEGA.

(*k*) A line from POLARIS through CAPELLA will touch RIGEL, 65° from CAPELLA. This line passes between ALDEBARAN (*α Tauri*), 10° west and 30° from CAPELLA, and BETELGEUSE, 10° east and 40° from CAPELLA. RIGEL and BETELGEUSE are diagonal stars in *Orion*.

(*l*) A line from ALDEBARAN through *Orion's Belt* passes near SIRIUS (*α Canis Majoris*) at an equal distance from ALDEBARAN. This is the brightest star in the heavens.

(*m*) 50° east of BETELGEUSE, forming nearly an equilateral triangle with BETELGEUSE and SIRIUS, is PROCYON.

(*n*) 30° north of PROCYON is CASTOR, and 5° S.E. of CASTOR is POLLUX.

(*o*) The apex of an isosceles triangle with CASTOR and PROCYON, 45° east, is REGULUS (*α Leonis*).

(*p*) Eastward from PROCYON to REGULUS 30° is DENEBOLA. DENEBOLA forms an equilateral triangle with ARCTURUS and SPICA.

ASTRONOMY IN FIELD WORK.

(*Fred. Simpson, A.M.I.C.E.*)

Choice of Instrument.

Choice of Instrument.—The most suitable instrument is the Transit Theodolite provided with a delicate level attached to the vernier arms of the vertical circle, and a sensitive striding level for use on the pivots of the horizontal axis. The adjustments must be made as perfectly as possible, and the residual index error carefully ascertained. The value of a division of each of the sensitive levels should be determined, and the position of the bubbles should be noted at the time of observing, and an allowance made for any variation in their readings. The most suitable form of diaphragm for astronomical work is that with simple vertical and horizontal cross hairs. For night work on stars it is necessary to illuminate the cross hairs, if the instrument is not provided with an illuminated axis; a lamp held a little distance in front of, and a little on one side of the object glass, may be used for this purpose.

With the ordinary eyepiece, altitudes exceeding 60° cannot be conveniently taken; for greater altitudes some form of diagonal attachment to the eyepiece is required. A makeshift arrangement can be improvised with a small piece of mirror held in the hand, or provisionally attached in front of the eyepiece at an angle of 45° to the axis of the telescope.

In observing the sun, a very convenient method, less trying to the eyes, is to cast the image of the sun and cross hairs on to a piece of white paper or card (to act as a screen) held about

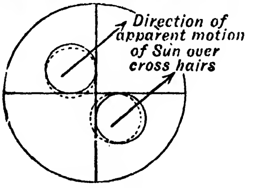


FIG. 18.

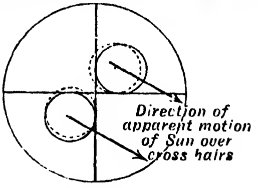


FIG. 19.

three or four inches from the eyepiece; the telescope must be focused for long-distance vision, with the cross hairs in focus; the telescope is then directed to the sun, and the eyepiece drawn out until perfect images of the sun and cross hairs are projected on the screen, the diameter of the image of the sun being smaller as the eyepiece is withdrawn. If the telescope inverts with direct vision, the image on the screen will be erect.

The limbs of the sun are observed when tangential to the respective cross hairs; the image of the sun should be allowed to leave one cross hair clamped a little in advance and followed with the other cross hair, using the tangent motion, so that the moving hair is brought tangential to the other limb as the first limb is just leaving the fixed hair; in this way only one hand is occupied with the tangent motion. The correction for semi-diameter must be applied to the readings in altitude and azimuth. In the case of the altitudes, the correction for semi-diameter is applied as taken from the Almanac; in that of the azimuths, the correction to be applied to the horizontal circle reading is approximately the semi-diameter, taken from the Almanac, multiplied by the secant of the angle of altitude. These corrections apply to observations taken on any one limb of the sun to reduce the reading to the centre of the sun; but by taking the means of pairs of observations, on the upper and lower limbs for altitude, and on the right and left limbs for azimuth, the necessity for correction for semi-diameter is eliminated.

In a double observation on upper and lower and right and left limbs, a pair of observations are made, one in which the horizontal hair is fixed in advance, the vertical hair being moved by the tangent motion of the horizontal circle; the other observation is then made with the vertical hair fixed in advance, the tangent motion of the vertical circle being then used to follow the lower or upper limb with the horizontal cross hair, up to the time of the vertical hair, with the right or left limb which is being observed.

The cross hair which is being moved should always be kept slightly on the disc of the sun, up to the time of contact with the other hair, as it becomes indistinct when off the disc.

The means of the circle readings of the altitudes and azimuths taken in this way will be thus reduced to the centre, and will correspond to the mean of the times of observations, and no correction for semi-diameter is required.

Altitudes require correction for index error (+ or −), for refraction corresponding to altitude (always −), parallax (in case of sun or planets) corresponding to altitude (always +); semi-diameter in case of sun or planets, + if lower limb is observed, − if the upper; this correction is eliminated if observations on upper and lower limbs are combined.

Refraction.—The mean refraction values are near enough for ordinary purposes when the altitude observed is over 20°, but for lower altitudes a correction may be necessary, corresponding to the altitude, temperature, and barometric pressure; refraction decreasing with higher temperature and increasing with higher barometric pressure.

Parallax.—The horizontal parallax of the sun is approximately 8·8″, and for any given altitude = 8·8″ × cosine of altitude, in seconds.

Declination as taken from the Almanacs must be corrected to correspond to the time of observation referred to time at the principal meridian adopted in the Almanac; generally it is sufficient to know his time to within ten minutes in spring or autumn, or within thirty minutes in summer or winter.

To Find Latitude.

Latitude may be deduced from a single altitude of a star, or the upper or lower limb of the sun, when crossing the meridian. This altitude is corrected for index error and refraction in the case of a star; in that of the sun a further correction is required for parallax and semi-diameter to reduce the altitude to the sun's centre. The declination is corrected to the time of observation.

In the case of the sun, and of stars with declination less than the latitude if of the same name:—
Latitude = 90° − (altitude + declination), if declination is of the opposite name to latitude.
Latitude = 90° − (altitude − declination), if declination is of the same name as latitude.
In case of stars with declination of the same name, but greater than latitude:—
Latitude = (altitude + declination) − 90° at the upper transit.
Latitude = altitude + (90° − declination) at lower transit of circumpolar stars.

More exact determination of the meridian altitude can be obtained from a series of ex-meridian altitudes taken about the time of the star or sun's meridian passage, provided three altitudes are taken at equal intervals of time, or equal intervals of azimuth (it is immaterial what intervals are taken, but they must be equal and not extend more than about twenty minutes on either side of the meridian passage).

The corrected altitude at the meridian may be obtained by the following reduction:—

Subtract the lowest altitude from the other two and let these differences, reduced to seconds, be denoted by d_1 and d_2; then the meridian altitude will be = lowest altitude + $\dfrac{(4d_1 - d_2)^2}{8(2d_1 - d_2)}$ seconds.

Example.—Three altitudes of the upper limb of the sun were taken at intervals of 6 minutes as follows:

	° ′ ″	° ′ ″	° ′ ″
	38 10 35	38 12 40	38 12 30
+ correction	2 17	− 38 10 35	× 38 10 35
= altitude on meridian, upper limb	38 12 52	2 05 = 125′	1 55 = 115

$$\frac{(4 \times 125 - 115)^2}{8(2 \times 125 - 115)} = 137'' = 2'\ 17''$$

To Find Time and Azimuth.

Time and Azimuth **from sun or star altitudes.**

Let a = altitude observed (corrected for under error and refraction in case of star, as well as for parallax and semi-diameter in case of sun);

l = latitude

p = polar distance = 90° − declination, if latitude and declination are of the same name, = 90° + declination, if latitude and declination are of different names.

$s = \dfrac{a + l + p}{2}$;

then,

$$\log \sin \tfrac{1}{2} \text{ Hour Angle} = \frac{\log \sec l + \log \operatorname{cosec} p + \log \cos s + \log \sin (s - a)}{2}$$

$$\log \cos \tfrac{1}{2} \text{ Azimuth} = \frac{\log \sec a + \log \sec l + \log \cos s + \log \cos (p - s)}{2}$$

The Hour Angle in arc is reduced to time; if from a P.M. observation of the sun this will be Apparent Time P.M., but if from an A.M. observation this Hour Angle in time should be deducted from 12, the result being Apparent Time A.M. To arrive at Mean Time, the Equation of Time must be added or subtracted as directed in the Almanac, this Equation of Time being first corrected to correspond to the approximate time of the observation at the principal meridian to which the Almanac refers.

In star observations for time, the Hour Angle found and reduced to time is deducted from or added to the Right Ascension of the star (deducted from if before the meridian transit, but added to if after), this gives the Sidereal Time of observation. Take the Right Ascension of the Mean Sun for the previous noon from the Almanac, and correct this to the meridian of the place of observation; this deducted from the Sidereal Time of observation, and the difference reduced to Solar Time, will give the Mean Time of the observation.

		h. m. s.
E.g., suppose Hour Angle found after transit of meridian .		4 30 24
Right Ascension of star		8 10 43
Sidereal Time at observation		12 41 07
	h. m. s.	
R.A. Mean Sun	1 35 28	
Correction for longitude 8 h. 54 m. West	+ 38	
		1 36 06
Interval in Sidereal Time from Mean Noon		11 05 01
Correction of interval, Sidereal to Mean Time . .		− 1 49
Mean Time of observation		11 03 12

The horizontal circle reading of the azimuth found as above should be corrected for the inclination of the horizontal axis of the vertical circle as denoted by the reading of the cross striding level; the difference of level in seconds of arc multiplied by the tangent of the altitude taken will give the correction to the horizontal circle reading this being added if the left-hand pivot is highest, and subtracted if the right-hand pivot is highest. The circle reading of some distant well-defined object should be used as a Reference Point to which the azimuth observed will then be connected.

In a series of observations where the sun has been utilised in the daytime, and it is desired to connect up a series of observations on stars, the adjustment of the instrument should be carefully gone over after sundown, and when the instrument has attained an even temperature, the Reference Point being utilised to check the results of the day observations with those of the night.

The degree of accuracy of the results will depend upon the care taken to eliminate slight errors in the data assumed or instrumental defects. In the case of observations for time, the taking of the mean of a combination of A.M. and P.M. observations will to a great extent eliminate the slight errors that may be due to instrumental defects, incorrect latitude assumed, or almanac corrections to date. The same remarks apply also to Azimuth except that the cross level error can only be eliminated by observations taken with reversed positions of vertical axis. In observations for latitude the errors of the instrument can be to a great extent corrected by taking the mean of meridian observations on stars to the north and to the south of the zenith.

Example (in time and azimuth; P.M. observation). Data: latitude (l) 53° 29′ 05″ N.; corrected declination, 0° 35′ 13″ S.; corrected altitude of the sun's centre (a) 20° 37′ 18″; polar distance (p) 90° + 0° 35′ 13″ = 90° 35′ 13″.

Time.

	h. m. s.	
a =	20 37 18	
l =	53 29 05	log sec 0·22546
p =	90 35 13	log cosec 0·00002
	2)164 41 36	
s =	82 20 48	log cos 9·12444
$s-a$ =	61 43 30	log sin 9·94482
		2)19·29474
½-hour angle	26 21 30	log sin 9·64787
	2	
hour angle	52 43 00 in arc	
	4	
hour angle	3 30 52 in time	

Azimuth.

	° ′ ″	
a =	20 37 18	log sec 0·02876
l =	53 29 05	log sec 0·22546
p =	90 35 13	
	2)164 41 36	
	82 20 48	log cos 9·12444
$(p-s)$ =	8 14 25	log cos 9·99549
		2)19·37415
	60 53 30	log cos 9·68707
	2	
Azimuth N.	121 47 00 W	

To Find Azimuth from Greatest Elongation of Circumpolar Stars.

At instant of greatest elongation :—

$$\sin \text{Azimuth} = \cos d \sec l;$$
$$\cos \text{Hour Angle} = \tan l \cot d;$$
$$\sin \text{Altitude} = \sin d \operatorname{cosec} l.$$

The Hour Angle, reduced to time, is added to the Right Ascension of the star if the Greatest Elongation is to the west of the meridian, or subtracted if to the east; this will give the Sidereal time of the Greatest Elongation. The sidereal time of the mean sun's passage of meridian for the succeeding noon subtracted from this will give the sidereal interval of time from noon, which, reduced to mean time, will give the Mean Time of Greatest Elongation. The altitude at Greatest Elongation is useful in setting the instrument for the observation.

Without knowledge of the latitude the Azimuth may be obtained from two observations on stars, one to the east, the other to the west of the meridian, and from the horizontal angle between them, which will be the sum of their Azimuths. Let this angle be denoted by $(A + A_1)$, and let d and d' be the respective declinations of the stars used, then,

$$\tan\left(\frac{A - A_1}{2}\right) = \tan\frac{d + d_1}{2} \times \tan\frac{d - d_1}{2} \times \tan\left(\frac{A + A_1}{2}\right),$$

giving the half difference of Azimuths, which added to and subtracted from the half sum of Azimuths will give A and A_1 respectively. The latitude can then be computed from one of the Azimuths and found by formula

$$\cos l = \cos d \operatorname{cosec} A.$$

Example.—Taking stars β Crucis (d_1 = 59° 06′ 30″ S.); γ Hydri (d = 74° 33′ 59″ S.); angle between their Greatest Elongations 57° 33′ 33″ ($= A + A_1$).

	° ′ ″	
$\frac{d + d_1}{2}$ =	66 50 14½	l tan 0·36874
$\frac{d - d_1}{2}$ =	7 43 44½	l tan 9·13264
$\frac{A + A_1}{2}$ =	28 46 46½	l tan 9·73980
$\frac{A - A_1}{2}$ =	9 53 05	l tan 9·24118
	± 28 46 46½	
Greatest Elongation γ Hydri	18 53 41½	
β Crucis	38 39 51	

The latitude may then be calculated as follows, from γ Hydri :—

Azimuth	18° 53′ 41½″	*l* cosec	0·48968
d	74 33 59	*l* cos	9·42508
		l cos	9·91476
latitude =	34 43 55 S.		

To reduce Mean Solar Time intervals to Sidereal Time intervals (approximate).

Reduce interval into hours and decimals of an hour, multiply by 10 and diminish product by $\frac{1}{70}$; the result will be correction in seconds to be *added* to Mean Time interval.

	h. m. s.		
EXAMPLE.	16 30 18 M.T.	$16{\cdot}505 \times 10 = 165{\cdot}05$	
Correction +	2 42·7	$-\frac{1}{70}$	2·36
	16 33 00·7 Sidereal T.		162·69 sec
			= 2 m. 42·69 sec.

To reduce Sidereal to Mean Time intervals.

Reduce intervals into hours and decimals of an hour, multiply by 10 and diminish product by $\frac{1}{60}$; the result will be correction to be *subtracted* from Sidereal Time interval.

	h. m. s.		
EXAMPLE.	16 33 00·7 Sid. T.	$16{\cdot}5502 \times 10 = 165{\cdot}50$	
Correction −	2 42·7	$-\frac{1}{60}$	2·76
	16 30 18 M.T.		162·74 sec.
			= 2 m. 42·74 sec.

The error in using these approximations will be less than $\frac{1}{10}$ second in a 24-hour interval.

To Find Longitude by Moon Culminating Stars.

This is a convenient method for determining the longitude when the Mean Time at Greenwich cannot be obtained by wireless time signals, the only requirements being a well-adjusted transit theodolite, a watch the rate of which can be depended upon for a short interval of time but the actual error of which need not be known, and a Nautical Almanac.

The moon varies comparatively rapidly in Right Ascension (R.A.), but its R.A. at the instant of its transit across the plane of the meridian can be accurately determined by comparing its time of transit with that of a star or stars which pass the meridian about the same time.

The theodolite is adjusted with its line of collimation in the plane of the meridian by the aid of one of the methods before described for obtaining azimuths; the time of transit, over the centre thread, of the selected stars of reference and the moon's bright limb are carefully noted.

The intervals (reduced to Sidereal Time) between the times of transit of the respective stars and the time of transit of the moon's bright limb, added to the R.A. of the respective stars (in the case of stars which transit before the moon), or subtracted from the R.A. of any stars which transit after the moon, will give the R.A. of the moon's bright limb at the instant of its transit over the meridian.

We have, under the heading of 'Moon at Transit at Greenwich,' the R.A. of the moon's bright limb at its transit at Greenwich, and also the variation of the Moon's R.A. in one day expressed in seconds; the difference between the R.A. of the moon's bright limb when passing the meridian of Greenwich and the R.A. as found when passing the meridian at the place of observation gives the change of R.A., and this difference (reduced to seconds), being divided by the variation corresponding to one hour difference of longitude, will give the difference of longitude between Greenwich and the place of observation expressed in hours and fractions of an hour.

From the table 'Apparent places of Stars' in the Nautical Almanac there must be selected convenient pairs of stars which transit within half an hour or so of the time of transit of the moon, and which have a declination nearly that of the moon, so that a slight movement of the telescope of the instrument in altitude will bring them within its field. As these stars are generally of small magnitude and not easily distinguishable it is advisable to (approximately) calculate the altitude at which they cross the meridian; the instrument may then be set in advance so that the star will cross approximately in the centre of the field. The moon's right ascension at the moment of transit will be found in the Nautical Almanac under the heading: 'Moon: at Transit at Greenwich.'

When the longitude of the place of observation differs considerably from that of Greenwich the 'variation for one hour' given in the tables should be corrected by interpolation to correspond to the approximate middle interval of time between the transit at Greenwich and that of the observed transit.

Let

t_1 t_2 be the observed times of transit of the star and the moon's bright limb;

RA_1 RA_2 be the Right Ascensions of the star and the moon's bright limb taken from the catalogue;

c the hourly variation (corrected, if necessary, to the middle time);

then the longitude in time $= \dfrac{(t_1 - t_2) - (RA_1 - RA_2)}{c}$,

the interval $(t_1 - t_2)$ if in Mean Time must be reduced to Sidereal Time,

$(RA_1 - RA_2)$ must be expressed in seconds,

the longitude will be expressed in hours and fractions of an hour, and to the west if positive and to the east if negative.

Timing Observations.

When working with an assistant the time may be noted at the instant a signal is given by the observer, but when observing alone it is difficult to note the exact time of contact or transit on watch or chronometer when the hands are employed in manipulating the instrument and the eye engaged in taking the sight.

In this case a convenient method is to use a watch whose time of beat is known (in modern watches usually $\frac{1}{5}$ second); this watch should be in such a position that the beats are distinctly heard, for example, suspended over the ear from the cap, in a small bag or watch pocket. The beats are then counted from the instant of contact or transit and continued until the seconds finger of the standard timepiece can be noted on some convenient second; this time noted, less the number of beats of the auxiliary watch, reduced to seconds and fractions, will be the actual time of the observation.

As probably some little time may elapse after the observation before the standard timepiece can be read, especially at night where a lamp has to be used, this extended counting of beats of say $\frac{1}{5}$ second becomes difficult when arriving at the double syllable numbers; but by adding a 'dummy' syllable to the first twelve counted and then continuing with 'thir-teen,' 'four-teen,' and so on, giving one syllable to each beat, the counting of $\frac{1}{5}$ seconds becomes quite easy, even, if carried to 100 beats and over, e.g. taking the dummy syllable as 'and,' and starting the instant of observation with 'nought,' the counting would continue:—'nought-and,' 'one-and,' 'two-and,' 'three-and,' . . . ''leven-and,' 'twelve-and,' 'thir-teen,' 'four-teen,' . . . 'twen-ty, 'twent'-one,' and so on. Then taking the first syllable as units and the second syllable as half, the counted beats multiplied by 4 and divided by 10 will give the required intervals in seconds and tenths, in the case of the auxiliary watch beating fifths of seconds.

Taking examples, say the beats counted between the instant of observation and the time noted on the timepiece was on the first syllable to thirty-seven, the time at that beat being 3 hrs. 47 mins. 15 secs.; then the interval to be deducted would be $\dfrac{37 \times 4}{10} = 14 \cdot 8$ seconds, and the corrected time of the observation would then be 3 hrs. 47 mins. 15 secs. $- 14 \cdot 8$ secs. $=$ 3 hrs. 47 mins. $0 \cdot 2$ secs. In another case, say the second syllable of 'nine-and' corresponded to 6 hrs. 22 mins. 55 secs. on the timepiece, the correction would then be $\dfrac{9\frac{1}{2} \times 4}{10} = 3 \cdot 8$ seconds, and the corrected time of observation 6 hrs. 23 mins. 55 secs $- 3 \cdot 8$ secs. $=$ 6 hrs. 23 mins. $51 \cdot 2$ secs.

This method of measuring short intervals of time is useful in other instances; for example, in estimating distacces by the velocity **of sound, etc.**

TABLE OF MEAN REFRACTIONS.

Corresponding to altitudes, A; at 50° F. and Barometer 29½ ins.

A	0′	5′	10′	15′	20′	25′	30′	35′	40′	45′	50′	55′
°	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″
0	34 54	33 51	32 49	31 50	30 52	29 57	29 3	28 12	27 28	26 35	25 50	25 6
1	24 25	23 45	23 7	22 30	21 56	21 23	20 51	20 21	19 52	19 24	18 58	18 33
2	18 7	17 45	17 23	17 1	16 41	16 20	16 1	15 42	15 23	15 5	14 48	14 31
3	14 15	13 59	13 44	13 29	13 15	13 1	12 48	12 36	12 24	12 12	12 1	11 50
4	11 39	11 28	11 18	11 8	10 59	10 49	10 40	10 30	10 21	10 12	10 3	9 55
5	9 47	9 39	9 31	9 23	9 16	9 9	9 2	8 55	8 48	8 54	8 36	8 29
6	8 23	8 17	8 12	8 6	8 0	7 55	7 49	7 44	7 39	7 34	7 29	7 24
7	7 20	7 15	7 11	7 6	7 2	6 57	6 53	6 49	6 45	6 42	6 37	6 33
8	6 30	6 26	6 22	6 18	6 15	6 12	6 8	6 5	6 2	5 59	5 55	5 52
9	5 49	5 46	5 43	5 40	5 38	5 35	5 32	5 29	5 26	5 24	5 21	5 19
10	5 16	5 14	5 11	5 9	5 6	5 4	5 2	4 59	4 57	4 55	4 53	4 51
11	4 49	4 46	4 44	4 42	4 40	4 38	4 36	4 34	4 32	4 30	4 29	4 27
12	4 25	4 23	4 21	4 20	4 18	4 16	4 15	4 13	4 11	4 10	4 8	4 7
13	4 5	4 3	4 2	4 0	3 59	3 57	3 56	3 54	3 53	3 52	3 50	3 49
14	3 47	3 46	3 45	3 43	3 42	3 41	3 39	3 38	3 37	3 36	3 35	3 33
15	3 32	3 31	3 30	3 29	3 27	3 26	3 25	3 24	3 23	3 22	3 21	3 20
16	3 19	3 18	3 16	3 15	3 14	3 13	3 12	3 11	3 10	3 9	3 8	3 7
17	3 7	3 6	3 5	3 4	3 3	3 2	3 1	3 0	2 59	2 58	2 57	2 57
18	2 56	2 55	2 54	2 53	2 52	2 52	2 51	2 50	2 49	2 48	2 48	2 47
19	2 46	2 45	2 45	2 44	2 43	2 42	2 42	2 41	2 40	2 39	2 39	2 38
20	2 37	2 37	2 36	2 35	2 35	2 34	2 33	2 33	2 32	2 31	2 31	2 30
21	2 29	2 29	2 28	2 27	2 27	2 26	2 26	2 25	2 24	2 24	2 23	2 22
22	2 22	2 21	2 21	2 20	2 20	2 19	2 18	2 18	2 17	2 17	2 16	2 16
23	2 15	2 15	2 14	2 14	2 13	2 13	2 12	2 11	2 11	2 10	2 10	2 9
24	2 9	2 8	2 8	2 7	2 7	2 7	2 6	2 6	2 5	2 5	2 4	2 4

A	0′	10′	20′	30′	40′	50′	A	0′	10′	20′	30′	40′	50′
°	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″	°	′ ″	′ ″	′ ″	′ ″	′ ″	′ ″
25	2 3	2 2	2 1	2 0	2 0	1 59	40	1 9	1 8	1 8	1 8	1 7	1 7
26	1 58	1 57	1 56	1 55	1 54	1 54	41	1 6	1 6	1 6	1 5	1 5	1 4
27	1 53	1 52	1 51	1 50	1 50	1 49	42	1 4	1 4	1 3	1 3	1 3	1 2
28	1 48	1 47	1 47	1 46	1 45	1 45	43	1 2	1 1	1 1	1 1	1 0	1 0
29	1 44	1 43	1 42	1 42	1 41	1 40	44	1 0	59	59	59	58	58
30	1 40	1 39	1 38	1 38	1 37	1 36	45	58	57	57	57	56	56
31	1 36	1 35	1 35	1 34	1 33	1 33	46	56	55	55	55	54	54
32	1 32	1 32	1 31	1 30	1 30	1 29	47	54	54	54	53	53	52
33	1 29	1 28	1 27	1 27	1 26	1 26	48	52	52	51	51	51	50
34	1 25	1 25	1 24	1 24	1 23	1 23	49	50	50	50	49	49	49
35	1 22	1 22	1 21	1 21	1 20	1 20	50	48	48	48	48	47	47
36	1 19	1 19	1 18	1 18	1 17	1 17	51	47	46	46	46	46	45
37	1 16	1 16	1 16	1 15	1 15	1 14	52	45	45	45	44	44	44
38	1 14	1 13	1 13	1 12	1 12	1 12	53	43	43	43	43	42	42
39	1 11	1 11	1 10	1 10	1 10	1 9	54	42	42	41	41	41	41

A	0°	1°	2°	3°	4°	5°	6°	7°	8°	9°
°	″	″	″	″	″	″	″	″	″	″
50	48	47	45	43	42	40	39	37	36	35
60	33	32	31	29	28	27	26	25	23	22
70	21	20	19	18	17	15	14	13	12	11
80	10	9	8	7	6	5	4	3	2	1

SUN'S PARALLAX IN ALTITUDE.

A	0°	10	20°	30°	40°	50°	60°	70°	80°	90°
Parallax	9″	9″	8″	8″	7″	6″	5″	3″	2″	0″

Graphic Method of finding Time and Azimuth from Altitude of Sun or Star.

Describe a circle, fig. 20, of any convenient radius; through the centre O draw diameter MOM_1. From O draw PO making angle POM equal to the latitude; draw OQ at right angles to PO. Through O draw OA, making AOM equal to the corrected altitude, and OD, making DOQ equal to the corrected declination, and on the side of OQ towards P if the declination and latitude are of the same name, or on the opposite side if of different names.

Through A draw AA_1 parallel to MM_1 and through D draw DD_1 parallel to QO, intersecting AA_1 in *s*.

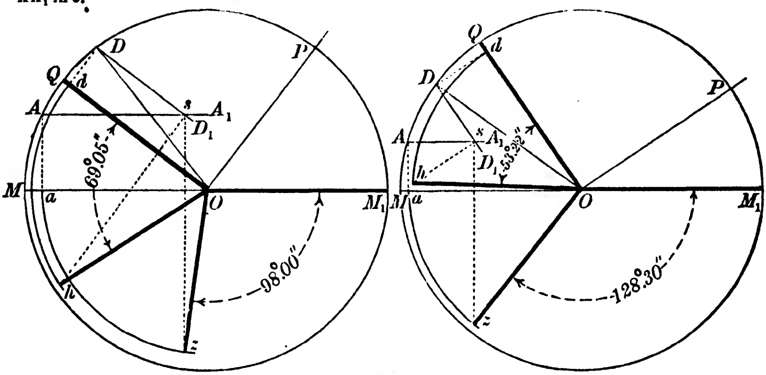


Fig. 20.

Data.	Result.	Data.	Result.
Lat. 53° 30′ N.	Azimuth N. 98° 00′ W.	Lat. 34° 30′ S.	Azimuth S. 128° 30′ W.
Dec. 14° 50′ N.	Hour Angle 69° 05	Dec. 19° 50′ N.	Hour Angle 53° 22′
Alt. 24° 20′ P.M.		Alt. 15° 40′ P.M.	

To find the Azimuth.

Drop a perpendicular from A to OM, intersecting this latter in *a*; from centre O and radius O*a* draw arc *as*, and from *s* draw *sz* perpendicular to MOM_1, intersecting arc *as* in *z*; through *z* draw O*z*. Then the angle M_1Oz will represent the azimuth from the elevated pole east or west of the meridian according as the time (A.M. or P.M.) at which the observation was made.

To find the Hour Angle.

Draw D*d* at right angles to QO, and with centre O and radius O*d* draw arc *dh*; from *s* draw line *sh* perpendicular to QO, intersecting arc *dh* in *h*; draw line O*h*. Then QO*h* will be the hour angle at the time of observation before or after apparent noon.

This diagrammatic method may be utilised for approximate determination of other problems, such as azimuth and time of sun or stars at instant of rising or setting; altitude and azimuth at a given time; or for altitude and time of passing prime vertical.

Geodetic Formulæ.

From measurements made of various meridians and parallels it has been found that the earth has the form of an ellipsoid of revolution with its greatest diameter at the equator and its least between the poles; and that the radius of curvature of the meridian varies with the latitude, being least at the equator and greatest at the poles, so that the length of a degree of latitude is more at the poles than at the equator. The length of a degree of longitude varies also with the latitude, being greatest at the equator and zero at the poles.

In treating with measurements of an extensive character over the surface of the earth, it is necessary to take into account its ellipsoidal shape, and it is convenient to employ certain approximations, so that spherical trigonometry may be used in the solution of the problems involved. The following formulæ enable the radius of a spherical surface to be found that approximates most nearly to the ellipsoidal surface embracing the area under consideration.

In these formulæ

a = the equatorial radius of the earth; e = the eccentricity; ϕ = the mean latitude; A = the azimuth.

Radius of curvature of meridian $= Q = \dfrac{a(1-e^2)}{\sqrt{(1-e^2\sin^2\phi)^3}}$

Radius of curvature of normal section perpendicular to meridian $= N = \dfrac{a}{\sqrt{(1-e^2\sin^2\phi}}$.

Radius of curvature of a normal section which cuts meridian at an angle A

$$= \frac{QN}{Q \sin^2 A + N \cos^2 A}.$$

Radius of curvature of an area of the earth's surface, or mean curvature $= \sqrt{QN}$.

Radius of a parallel of latitude $= N \cos \phi = \frac{a \cos \phi}{\sqrt{1 - e^2 \sin^2 \phi}}$

The logarithms of the most probable values of the constants used above are as follows (taking a in statute miles):—

$$\log. a = 3{\cdot}59806. \quad \log. e^2 = 7{\cdot}83047. \quad \log. (1 - e^2) = 9{\cdot}99853.$$

Adopting the radius which most nearly corresponds to the surface considered at the mean latitude ϕ, any measured length l on this surface may be reduced to arc by one of the following formulæ, ρ being the radius adopted reduced to the same unit as l:—

$\frac{l \times 57{\cdot}2958}{\rho}$ in degrees of arc; $\frac{l \times 3437{\cdot}747}{\rho}$ in minutes of arc; $\frac{l \times 206265}{\rho}$ in seconds of arc.

The table on page 262, 'Length of a Degree of Longitude and Latitude,' is calculated from the accepted dimensions of the earth. Values for intermediate latitudes may be obtained from this table, by interpolating, with sufficient exactitude for most purposes. The length of a degree on a great circle at right angles to the meridian at a given latitude may be obtained by multiplying the length of a degree of longitude (as taken from the table) by the secant of the latitude.

Convergence of Meridians.

In traverses and route surveys, a survey line is referred to a meridian at a certain point crossing the meridian with an azimuth A, this line, if continued in the same direction, having increased or diminished azimuths. The differences of azimuth can be approximately determined by calculating the convergence of the meridian corresponding to a given unit departure at the mean of the latitudes, and multiplying this by the departure as calculated from the traverse. To obtain the azimuth on crossing the second meridian, the convergence obtained in the manner just explained is added to or subtracted from the azimuth crossing the first meridian, being added where the longitude is increased and subtracted where the longitude is diminished.

The convergence, corresponding to various unit lengths of departure, may be calculated by the following formulæ, with sufficient approximation for use in ordinary traverse surveying. The length of a degree of longitude corresponding to the mean latitude is taken in miles from table, page 262, interpolating if necessary; then :—

Log. of convergence in seconds per mile of departure

$= \log. \sin \phi + 3{\cdot}55630 - \log.$ miles per 1° longitude.

Log. of convergence in seconds per 1,000 feet of departure

$= \log. \sin \phi + 2{\cdot}83367 - \log.$ miles per 1° longitude.

Log. of convergence in seconds per 1,000 metres of departure

$= \log. \sin \phi + 3{\cdot}34966 - \log.$ miles per 1° longitude.

To range a parallel of latitude by setting out chords of a given length :—

The chords are ranged making an angle to the meridian of 90° − half the angle of convergence corresponding to the length of the chord at the proposed latitude. The succeeding chords will have an angle of deflection from the preceding chord produced, equal to the convergence corresponding to the length of chord used.

To range a parallel of latitude by means of offsets :—

A line is run, perpendicular to the meridian, then the lengths of offsets are equal to the distances along this perpendicular line multiplied by the sine of half the angle of convergence corresponding to the lengths.

If the offsets are equidistant the length of the second offset will be four times the length of the first; the third, nine times the first, and so on; the n^{th} offset will be n^2 times the length of the first.

Horizontal Sun Dial.

On a line NS, fig. 21, mark off a convenient length NO; from O set off a line towards L making an angle NOL equal to the latitude of the position (taken in diagram 53° 30′), and from N draw a perpendicular to OL, cutting the latter in point L. From N and with length NL mark off a point C in the line NS, and with C as centre and the same length NL as radius describe the quadrant NQ and join NQ with a straight line.

Divide the quadrant NQ into six equal parts (each corresponding to one hour) and from these divisions on the arc draw lines towards the centre C and mark the points at which these radial lines cross the line NQ.

From O draw a line perpendicular to NS and mark off OE equal to CQ and join NE.

From each division on line NQ draw a line paralle' to NS, marking where this line crosses the line NE in each case.

Then lines drawn from O through each of these points, marked off in NE, will be the dial lines corresponding to the hour divisions on the quadrant from which they were found respectively.

Having completed the divisions of one quadrant, these angles may be transferred to the other quadrants but in reverse order (VI–VII equals VI–V, VI–VIII equals VI–IIII, and so on).

An allowance must be made for the thickness of the style between the AM and PM portions of the dial, the AM dial being exactly similar to the PM dial but drawn with the XII lines the thickness of the style apart.

The style is constructed with the angle of inclination of the inclined face equal to the latitude of the site, its lower point at O and its upper towards the elevated pole, so that its

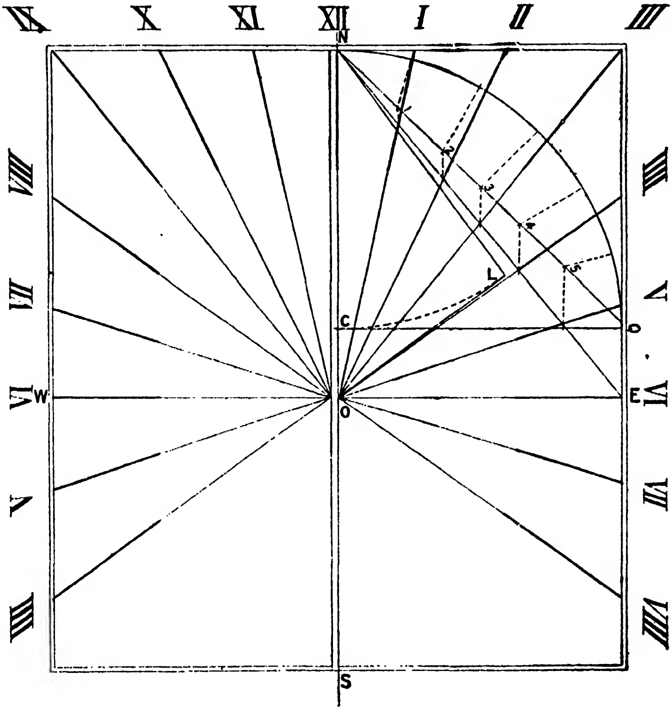


FIG. 21.

plane is in that of the meridian, the XII (noon) lines pointing towards the north in northern latitudes or towards the south in southern latitudes, and the VI–VI lines pointing east and west, so that the upper face of the style will be parallel to the Polar axis. The plane of the dial should be perfectly level.

(Methods of fixing the direction of the true north and south line will be found on page 261.)

The time indicated by the dial is sun time, and to get mean time the table of Equation of Time, page 261, should be applied as there directed. To arrive at standard time a constant correction must be applied according to the difference of longitude between the dial and the standard meridian (Greenwich in Great Britain) adopted in the country (subtracted if to the east or added if to the west).

The dial angles may be calculated if preferred by formula

$$\tan t = \sin l \tan h$$

where

t = angle at O from NO to the hour line on dial corresponding to h; l = latitude of position; h = hour angle.

SOURCES OF INFORMATION.

***Geodetic Surveying.*—The results of very precise observations are published annually in the Geodetic Report, included in the Report of the Survey of India.**

***United States.*—Articles relating to surveying problems will be found in recent and current issues of *Engineering News-Record* and *Civil Engineering* (U.S.A.).**

***Transition Curves.*—(See Section XVIII, 'Literature,' and XXXII, Part I).**

***Photography.*—(1) 'Engineering Applications of Aerial and Terrestrial Photogrammetry. By B. B. Tolley. A book, reviewed, *The Surveyor*, March 24, 1939.**

(2) 'Aerial Photography applied to Surveying.' By C. A. Hart. A book, reviewed, *Municipal Engineering*, February 6, 1941.

***Civil Engineers' Surveying.* (1) 'Modern Surveying for Civil Engineers.' A book, by H. F. Birchall. Reviewed, *The Engineer*, May 7, 1935.**

(2) 'Route Surveying.' By G. W. Pickels and C. C. Wiley. Second edition, 1939. A work assembling, in a single presentment, the information necessary with respect to purposes so much akin that both the operations and the computation are, to a large extent, common to all the purposes. Operations peculiar to, or more especially useful in, surveys for roads and for railways respectively, or in those for other classes of construction, are included in the scope of the book.

(3) 'A Treatise on Surveying.' Vol. 2. By R. E. Middleton, and the late O. Chadwick. Fifth edition, revised by M. T. M. Ormsby.

(4) 'Plane and Geodetic Surveying. Part I, Plane Surveying.' By the late David Clark. Fourth edition, 1946, revised and enlarged by J. Glendinning, London. Constable, price 30*s*.

(5) *Photographic Surveying.* 'Photographic Surveying in Relation to Road Engineering in Highly Developed Countries.' By B. F. J. Bradbeer and Professor C. A. Hart. Paper, *Inst. C.E.*, March 1948. Covers ground photography and air photography.

(6) Whitaker's Almanack.

SECTION IX

PART I

TIMBER—SEASONING—PRESERVING—STRENGTH PROPERTIES

(pp. 277–296.)

(Revised by Forest Products Research Laboratory.)

PART II

ROOFS—FLOORS—PARTITIONS—STAIRS—BRICKWORK—LONDON BUILDING ACTS—FOUNDATIONS—MASONRY

(pp. 297-334)

(Revised by A. B. Searle.)

SECTION IX

PART I

TIMBER — SEASONING — PRESERVING — STRENGTH PROPERTIES.

(Revised by Forest Products Research Laboratory.)

TIMBER.

In all specifications for building materials and construction, it is customary to use the terms 'Timbering,' 'Joiners' work,' and 'Cabinet work.' The first-named applies only to the wood used in constructional work. These terms are again subdivided into the following :—

Baulks.—Pieces of sawn or hewn timber of equal or approximately equal cross dimensions of greater size than 4 ins. × 4½ ins.

Planks. (softwood)—Pieces of square-sawn timber 2 ins. to 4 ins. in thickness by 11 ins. and over in width.

Planks (hardwood).—Pieces of square-sawn timber 2 ins. and over in thickness by 6 ins. and over in width.

Deals.—Pieces of square-sawn softwood timber 2 ins. to 4 ins. in thickness by 9 ins. to under 11 ins. in width. Thicknesses less than this are termed boards.

Battens.—Pieces of square-sawn softwood timber 2 ins. to 4 ins. in thickness by 5 ins. to 8 ins. in width, inclusive.

Boards.—Pieces of square-sawn softwood timber under 2 ins. in thickness by 4 ins. or over in width.

Scantlings.—Pieces of square-sawn softwood timber 2 ins. to 4 ins. in thickness × 2 ins. to 4½ ins. in width, inclusive.

Squares.—Pieces of strictly equal-sided sawn timber of any stated square dimension.

Timber Measures.

BAULKS AND TIMBER.—Sold at per load of 50 ft. cube. Deals, planks, battens, and scantlings at per Leningrad standard. (One standard = 165 ft. cube.)

PREPARED FLOOR BOARDS.—These are usually sold at per square. This is a standard measurement which equals 100 ft. superficial. Till recently it was the trade custom to supply a specific number of feet according to the width of the board, which should approximate to the square. These were as follows :—

4-in. boards	300 ft.	6-in. boards	200 ft.
4½ „	270 „	6½ „	190 „
5 „	240 „	7 „	180 „
5½ „	220 „		

This practice, which was always unsatisfactory owing to the inaccuracy of the figures, has now been discontinued.

LUMBER (see above) AND AUSTRALIAN WOOD.—Sold at per 1,000 ft. of 1-in. superficial measure for all thicknesses of 1 in. and over. Under 1 in. at per 1,000 ft. superficial of their thicknesses. Teak at per load of 50 cub. ft.

Descriptions of Timbers.*

(For specific quantities and weights per cub. ft., see Section V, Part II, pp. 123–124.)

ALDER, COMMON or BLACK (*Alnus glutinosa*).—Europe, including British Isles. A soft non-durable timber closely resembling birch. Light reddish-brown. Imported in the form of plywood; also used for turnery.

APITONG or BAGAC.—For description see GURJUN.

ASH, EUROPEAN (*Fraxinus excelsior*).—Europe, including British Isles. A straight-grained wood nearly white in colour, chiefly remarkable for its outstanding toughness and resistance to shock loads. A good bending timber. Used for motor vehicle framing, under-framing and bent work; aircraft construction; shafts and agricultural implements; handles of picks, shovels, etc.; sports goods.

ASH, AMERICAN WHITE (principally *Fraxinus americana*).—North America. Similar to English ash but generally considered inferior for aircraft construction, sports goods and other work where bending properties of a high order are required.

ASH, JAPANESE (*Fraxinus mandshurica*).—Japan. Similar to European and American white ash but not so strong. Used mainly for interior fittings, cabinet work and plywood.

BAGAC or APITONG.—For description see GURJUN.

BALSA (*Ochroma lagopus*).—Central and South America. The lightest timber in general use, varying from as little as 2½ lb. per cub. ft. to 20 lb. or more. The lighter grades are very soft and spongy. Nearly white in colour. Used for model aeroplanes, fairings in certain types of aeroplanes, floats, sound and heat insulation, etc.

BASSWOOD (principally *Tilia americana*).—North America. A soft, non-durable timber, nearly white to pale brown in colour. Used for turnery, cabinet work and especially pianoforte manufacture.

BEECH, EUROPEAN (*Fagus sylvatica*).—Europe, including British Isles. A straight-grained wood with a fine even texture. The natural colour is a light reddish-brown; the wood is often steamed, giving it a darker reddish colour. Comparable to oak in strength but not durable in exposed situations. A good bending timber. Used for a great variety of purposes, including furniture and cabinet making; automobile bodies; general turnery; bent work; parts of textile and other machinery; pianoforte manufacture.

BIRCH, EUROPEAN (*Betula pubescens* and *B. pendula*).—Europe, including British Isles. A fairly straight-grained, fine-textured wood, white to light brown in colour. Large quantities are imported in the form of plywood. It is also widely used for turnery.

BIRCH, CANADIAN YELLOW (principally *Betula lutea*).—North America. Somewhat heavier than European birch and darker in colour, light to dark reddish-brown. Extensively used for chair-making, furniture parts, framing for upholstered work, kitchen utensils, tool handles and general turnery; also for automobile bodies; flooring and general utility work.

BLACK BEAN (*Castanospermum australe*).—Australia. A handsome decorative timber, greyish-brown in colour. Used for cabinet work and panelling.

BLACKWOOD, AUSTRALIAN (*Acacia melanoxylon*).—Australia. A handsome reddish-brown timber marked with darker streaks. Used for interior decorative work.

BOXWOOD (*Buxus sempervirens*).—Europe (including British Isles) and Asia Minor. Exceedingly hard and fine-textured. Used largely for blocks for wood engraving, turnery and mathematical instruments. Other commercial boxwoods are West Indian boxwood (*Gossypiospermum praecox*), Knysna or Kamassi boxwood (*Gonioma kamassi*), and Cape boxwood (*Buxus macowani*), both from South Africa.

CAMPHORWOOD, BORNEO or KAPUR (various species of *Dryobalanops*).—Borneo and Malaya. A light reddish-brown constructional timber, closely resembling gurjun (*q.v.*) and used for similar purposes.

CAMPHOR, EAST AFRICAN (*Ocotea usambarensis*).—East Africa. A cabinet timber of pleasing appearance, darkening on exposure to a deep brown colour. Used for interior decoration work and fittings.

CEDAR, BORNEO.—See SERAYA.

CEDAR, CENTRAL AMERICAN (principally *Cedrela mexicana*).—Central America and West Indies. A soft, reddish, fragrant wood, somewhat resembling the lighter grades of true mahogany. Used for cigar and cigarette boxes and also for boat-building, cabinet work and joinery.

* More detailed descriptions of most of these timbers may be found in the official publications of the Forest Products Research Laboratory (Department of Scientific and Industrial Research) entitled 'Home Grown Timbers' and 'Empire Timbers,' obtainable from H.M. Stationery Office, York House, Kingsway, W.C. 2, or through any bookseller.

A more complete list of timbers used in Great Britain, their trade names, botanical names, and sources of supply, may be found in British Standard No. 881—1939 (Nomenclature of Hardwoods) and British Standard No. 589—1935 (Nomenclature of Softwoods), obtainable from the British Standards Institution, Publications Department, 28 Victoria Street, London, S.W. 1.

CEDAR, PORT ORFORD (*Chamaecyparis lawsoniana*).—North America. A high grade softwood with a strong cedar scent. Durable in exposed situations and resistant to acids. Used for interior fittings, furniture (especially for lining clothes chests) and for battery separators.

CEDAR, WESTERN RED (*Thuja plicata*).—North America. A reddish-brown, non-resinous, light-weight softwood noted for its outstanding durability under all conditions. Used for roofing shingles, weather-boarding, glass-house construction, and also for interior joinery work and cabinet making.

CHESTNUT, SWEET or SPANISH (*Castanea sativa*).—Europe, including British Isles. Resembles oak, from which it may be distinguished by the absence of the silver-grain figure. Lighter in weight and not so strong as oak. Durable. Used for many of the same purposes as oak. Timber from old trees is often spiral-grained and liable to ring-shake.

CHUGLAM, WHITE, and INDIAN SILVER-GREY WOOD (*Terminalia bialata*) are two varieties of timber furnished by the sapwood and heartwood respectively of the same species. White chuglam is a general utility hardwood. Indian silver greywood has a fine ornamental figure and is used for interior decorative work and panelling.

CRABWOOD or ANDIROBA MAHOGANY (*Carapa guianensis*).—Tropical South America. Closely related to true mahogany and resembling a plain mahogany in appearance and general properties. Somewhat heavier and darker in colour than average Honduras mahogany. Used as a substitute for mahogany.

DEAL, RED or YELLOW.—See REDWOOD, BALTIC.

DEAL, WHITE.—See WHITEWOOD, BALTIC.

DOUGLAS FIR.—See FIR, DOUGLAS.

EBONY (various species of *Diospyros*).—India, Ceylon, East Indies, Tropical Africa. The different kinds of ebony vary from brown, streaked with grey and black, to jet black. They are all heavy, hard and close-textured. Principally used for turnery, and also in the form of veneer for cabinet work.

ELM, COMMON ENGLISH (*Ulmus procera*).—British Isles. Elm is inclined to be cross-grained and has a corresponding tendency to warp. High temperature steaming treatment is effective in reducing this tendency, and timber so treated can be used successfully for furniture and interior fittings. A fairly strong, tough timber, somewhat inferior to oak in all strength properties. Durable under water and excellent for piles or dock and wharf construction. Other typical uses are for coffin boards; ends of packing cases and boxes; carts, wheelbarrows and wagons; furniture, chiefly chairs; barge and boat-building; sea groynes; agricultural buildings in the form of weather-boarding.

ELM, ROCK (*Ulmus thomasi*).—North America. A tough, dense wood, strong and durable; especially useful in resisting abrasion. Chiefly used in dock and wharf construction and as fenders, ships' belting, and in the construction of rowing boats and other small craft. Also for bent work as in motor body and railway carriage roofs.

ELM, WYCH (*Ulmus glabra*).—British Isles. Superior to other species of home-grown elm, being straighter in the grain and not so coarse-textured. Bends well and approaches ash in toughness. Used in chair-making, boat-building, for shafts and agricultural implements.

ENG or IN (*Dipterocarpus tuberculatus*).—Burma. A constructional timber very similar to gurjun (*q.v.*) but appreciably heavier and correspondingly harder and stronger.

FIR, DOUGLAS (*Pseudotsuga taxifolia*).—North America. Also known as OREGON PINE and COLUMBIAN PINE. A general utility softwood, usually straight-grained, moderately resinous. Slightly denser than Baltic redwood. Its more than average strength, combined with the large sizes in which it is available, renders it eminently suitable for heavy constructional work. The dense grades compare favourably with the best pitch pine. It is also used for interior fittings, joinery and a great number variety of similar uses.

FIR, SILVER (*Abies alba*).—Europe, including British Isles. Similar to European spruce and marketed with that timber as WHITEWOOD (*q.v.*).

GABOON or OKOUMÉ (*Aucoumea klaineana*).—Gaboon and Spanish Guinea. Also known as Gaboon mahogany. Not a true mahogany, but widely used as a mahogany substitute, principally in the form of plywood and lamin-board.

GREENHEART (*Ocotea rodiaei*).—British Guiana. A timber of outstanding strength and durability, especially in sea water, straight-grained and remarkably free from knots and other defects. Eminently suitable for marine work, piles, dock-gates and wharf construction. Also a first-class ship-building timber.

GURJUN, KERUING, APITONG or BAGAC and YANG (various species of *Dipterocarpus*).—Gurjun is the usual trade name for the timber of this group of species from Burma and the Andaman Islands. The corresponding species from Malaya and Borneo are known as keruing. Apitong or bagac from the Philippine Islands and yang from Siam belong to the same class. The timbers of this group are nearly indistinguishable for practical purposes. They are useful constructional timbers obtainable in large sizes. Used in railway wagon and carriage construction, in buildings as a substitute for oak and for parquet flooring.

HEMLOCK, WESTERN (*Tsuga heterophylla*).—British Columbia. A non-resinous softwood timber, typically straight-grained and of fairly even texture, superior in these respects to Eastern hemlock (*Tsuga canadensis*). Similar in strength properties to Douglas fir but not quite so strong. Not durable in exposed situations. Useful as a building material and for interior work generally. Also for boxes and crates, especially for foodstuffs.

HICKORY (various species of *Carya*).—North America. Exceedingly tough, hard, smooth and straight-grained. Principally used for spokes and rims of wheels, tool handles, machine parts, etc. White and red hickory are respectively the sapwood and heartwood of the same tree.

HORNBEAM (*Carpinus betulus*).—Europe, including British Isles. A hard, strong, tough timber, superior to oak in all strength properties, equal to ash in resistance to shock. Turns well and takes a very smooth finish. Used for cogs, wood screws, mallets, tool handles and other small articles, also for piano work and for flooring.

HORSE CHESTNUT (*Aesculus hippocastanum*).—Europe, including British Isles. Creamy-white with a fine, uniform texture. Not strong or durable. Used for turnery, dairy and kitchen utensils, etc.

IN.—See ENG.

IROKO (*Chlorophora excelsa*).—West and East Africa. A useful constructional timber, similar to home-grown oak in strength. Fairly durable and of good appearance, being light to dark brown in colour. Suitable for joinery and building work.

JARRAH (*Eucalyptus marginata*).—Western Australia. A heavy, hard, dark red timber with high mechanical properties and unusual durability. Obtainable in large dimensions and used for heavy structural work, in railway wagon building and for flooring.

KAPUR.—See CAMPHORWOOD, BORNEO.

KARRI (*Eucalyptus diversicolor*).—Western Australia. A high class constructional timber very similar to jarrah in appearance and properties but slightly stronger and not so durable. Suitable for structural work where strength, long lengths and large cross sections are required. For underground work or in damp situations preservative treatment is necessary.

KAURI (various species of *Agathis*).—New Zealand, Australia, etc. A valuable softwood timber with a fine, even, silky texture and a lustrous surface. Very strong for its weight; durable. The best variety is from New Zealand. Queensland kauri is lighter and softer. Other species occur in the islands of the Pacific. Kauri is obtainable in large sizes and is used for vat-making; also for deck planking and high-class joinery.

KERUING.—See GURJUN.

LARCH (*Larix decidua*).—Europe, including British Isles. The most valuable home-grown softwood on account of its strength and durability. Used for gates and fencing, mine timber, and out-door work generally.

LAUAN.—For description see SERAYA.

LAUREL, INDIAN (*Terminalia alata*, *T. crenulata*, and *T. coriacea*, formerly known as *T. tomentosa*).—India and Burma. A handsome walnut-brown timber figured with irregular streaks of darker colour. Used for high grade interior decorative work, such as panelling, doors, staircases, etc.

LIGNUM VITÆ (principally *Guaiacum officinale*).—West Indies. One of the hardest and heaviest timbers in commercial use. Heartwood dark greenish-brown to nearly black in colour, sharply defined from the yellowish sapwood. Used as bearings for propellor shafts for steamers on account of its natural lubricating property; also for mallets and turnery.

LIME, AMERICAN.—See BASSWOOD.

LIME, COMMON EUROPEAN (*Tilia vulgaris*).—Europe, including British Isles. A fine-textured, white or pale yellow wood, soft yet compact, cuts cleanly without splitting. Not durable. Excellent for carving; also used for turnery and parts of musical instruments including pianos.

MAGNOLIA (various species of *Magnolia*).—North America. A fairly soft, fine-textured, white wood, used in cabinet work, for mouldings and interior fittings.

MAHOGANY, AFRICAN (principally *Khaya ivorensis*).—Tropical Africa. Commercial consignments are commonly classified according to the port of shipment or district from which they are derived, *e.g.*, Lagos and Benin (Nigeria), Axim, Half-Assinie and Takoradi (Gold Coast), Grand Bassam (Ivory Coast), etc. Colour varies from a light pinkish-brown to a deep reddish shade. Grain usually interlocked, producing the characteristic stripe or roe figure of quarter-sawn stock. Texture somewhat coarser than Honduras mahogany and the wood is not so stable under varying conditions of atmospheric humidity. Obtainable in large sizes.

MAHOGANY, ANDIROBA.—See CRABWOOD.

MAHOGANY, CUBAN or SPANISH (*Swietenia mahagoni*).—West Indies. The original mahogany noted for its handsome appearance and remarkable stability. Denser and finer-textured than other commercial species of mahogany. Rarely used nowadays except for the finest furniture and interior decorative work, having been largely replaced by Honduras and African mahoganies.

MAHOGANY, GABOON.—See GABOON.

MAHOGANY, HONDURAS and CENTRAL AMERICAN (*Swietenia macrophylla*).—Central America. The same or a closely allied species supplies the bulk of the mahogany imported from other parts of Central and South America, including Mexico, Nicaragua, Peru and Brazil. Lighter and softer than Cuban mahogany. Colour varies from light yellowish-brown to a rick dark shade practically indistinguishable from Cuban mahogany. Grain tends to be interlocked but there is a good proportion of plain, straight-grained timber. Remarkably stable and does not shrink and swell as much as most woods. Used for furniture and interior decorative work; also for instruments of precision, aeroplane propellors, turnery, mouldings, printers' blocks, and superior joinery.

MAHOGANY, SAPELE (*Entandrophragma cylindricum*).—Tropical Africa. A decorative wood of the mahogany family, with a characteristic regular stripe or roe figure. Used almost exclusively in the form of veneer for interior decorative work and furniture.

MANSONIA (*Mansonia altissima*).—Tropical Africa. Resembles American black walnut and is used as a substitute for that timber.

MAPLE, ROCK or SUGAR (principally *Acer saccharum*).—North America. A light-coloured wood of fine, even texture; hard and strong. One of the best timbers for flooring and widely used for that purpose; also for interior decoration and for a variety of industrial uses where its hardness makes it particularly suitable.

MATAI (*Podocarpus spicatus*).—New Zealand. Technically a softwood, but its texture and general properties place it in the hardwood class for many practical purposes. Used for flooring.

MERANTI.—For description see SERAYA.

OAK, AMERICAN RED (*Quercus borealis* and other species).—North America. Differs from American white oak in being on the whole coarser in texture, less uniform in colour, not so strong or durable and generally inferior.

OAK, AMERICAN WHITE (*Quercus alba* and other species).—North America. Similar to European oak in appearance and general properties but subject to more variation in quality. Considered inferior to European oak for high-class work.

OAK, EUROPEAN (*Quercus robur* and *Q. petraea*).—Europe, including British Isles. The two species are not separated in practice. English oak is unexcelled by other varieties for constructional purposes demanding a combination of strength and durability. Various grades of Continental oak differ as regards their hardness, ease of working and general quality. Uses too well known to be enumerated.

OAK, JAPANESE (principally *Quercus mongolica* var. *grosseserrata*).—Japan. Is characterised by the open, porous texture of the wood and easy working properties. Obtainable in large sizes. Used principally for interior work and furniture.

OAK, SILKY (*Cardwellia sublimis*).—Australia. Not a true oak but has a somewhat similar appearance. Used for furniture and interior decorative work.

OAK, TASMANIAN (*Eucalyptus obliqua*, *E. regnans* and *E. gigantea*).—Australia. Not a true oak but has a passing resemblance to plain oak. Used principally for flooring as a substitute for true oak.

OBECHE or AFRICAN WHITEWOOD (*Triplochiton scleroxylon*).—West Africa. A soft, light, nearly white wood, comparable to poplar in strength. Works easily to a smooth finish and takes stains and paint well. Obtainable in large dimensions. Used principally in the furniture trade and for joinery.

OKOUMÉ.—See GABOON.

OLIVE, EAST AFRICAN (*Olea hochstetteri*).—East Africa. A highly decorative wood, light brown marked with irregular dark grey-brown veins or streaks of varying width. Used for flooring and ornamental purposes.

PADAUK, ANDAMAN (*Pterocarpus dalbergioides*).—Andaman Islands. A distinctively coloured wood of a rich handsome hue. Very strong, durable and hard wearing. Formerly employed for constructional work; now principally used (in the United Kingdom) for cabinet work and interior fittings and panelling in banks, offices, etc.

PADAUK, BURMA (*Pterocarpus macrocarpus*).—Burma. Rather harder and heavier than the better known Andaman padauk and used for similar purposes. Yellowish-red to brick-red.

PINE, CORSICAN (*Pinus nigra* var. *calabrica*).—Widely planted in the British Isles in recent years. Yields timber similar to Scots pine but with a wider sapwood zone and usually somewhat coarser in texture. Generally similar to the lower grades of imported deal or redwood and used for the same purposes.

PINE, JACK (*Pinus banksiana*).—Canada. Similar to Canadian red pine and Baltic redwood but inferior to the best grades of those timbers. Used for similar purposes.

PINE, PARANA (*Araucaria angustifolia*).—South America. A close-textured softwood with some resemblance to kauri. Heartwood often streaked with red.

PINE, PITCH (various species of *Pinus*).—Supplies from the Southern United States consist principally of longleaf pine (*Pinus palustris*), shortleaf pine (*P. echinata*), and loblolly pine (*P. taeda*), of which the first-named is considered the best. The three species are often indiscriminately mixed. The term Rosemary pine refers to coarse, fast-grown timber of inferior quality. Honduras pitch pine is *P. caribaea*. Pitch pine is strong, hard and heavy and is used principally for constructional work and to some extent for interior fittings.

PINE, RED (*Pinus resinosa*).—Canada. Similar to Baltic redwood and used for the same purposes.

PINE, SCOTS (*Pinus sylvestris*).—British Isles. The same species as imported redwood. Used for similar purposes.

PINE, SIBERIAN (*Pinus koraiensis* and *P. cembra* var. *sibirica*).—Siberia and Manchukuo. A softwood of fine texture, generally similar to Baltic redwood but the best grades are superior to corresponding grades of Baltic redwood. Used for general joinery work.

PINE, SUGAR (*Pinus lambertiana*) and PINE, WESTERN WHITE (*Pinus monticola*).—Western North America. Similar to the yellow or white pine of Eastern North America but slightly harder and stronger and obtainable in larger sizes. Used for high grade joinery, pattern-making, etc.

PINE, YELLOW or WHITE (*Pinus strobus*).—Eastern North America. A fine-textured softwood, straight-grained, easy to work, cutting cleanly in all directions of the grain. Shrinks and warps less than most softwoods. Used for superior joinery work of all kinds, pattern-making, etc.

POPLAR (various species of *Populus*).—Europe, including British Isles. Comparatively light and soft, straight-grained, rather woolly, not durable. Fairly tough and withstands rough usage without splintering. Used for the bottoms of carts and wagons, brake blocks, etc.

PYINKADO (*Xylia dolabriformis*).—Burma. A very hard, durable timber, used for heavy constructional work including marine structures.

REDWOOD, BALTIC, or RED or YELLOW DEAL (*Pinus sylvestris*).—Europe. The standard timber for housebuilding, carpentry and joinery, railway sleepers, scaffold and transmission poles, etc. Quality depends largely on growth rate and knottiness.

REDWOOD, CALIFORNIAN.—See SEQUOIA.

ROSEWOOD (various species of *Dalbergia*). Various tropical countries. Very hard and durable, usually brown in colour with darker brown or black markings. Used for turnery and ornamental work.

SEQUOIA or CALIFORNIAN REDWOOD (*Sequoia sempervirens*).—North America. A high grade softwood, red in colour, of light weight outstanding durability. Used for out-door constructional work and for interior fittings.

SERAYA, RED, with RED MERANTI and RED LAUAN (various species of *Shorea*).—Red seraya or Borneo red cedar is the usual trade name for the timber of this group of species from Borneo. The corresponding timbers from Malaya and the Philippine Islands are known as red meranti and red lauan respectively. They are largely used for interior constructional work, joinery, panelling, and as substitutes for mahogany in furniture. They are obtainable in large sizes.

SERAYA, WHITE, with YELLOW or WHITE MERANTI and WHITE LAUAN (various species of *Parashorea* and *Shorea*).—White seraya or Borneo white cedar is the usual trade name for timber of this group of species from Borneo. Yellow or white meranti and white lauan are the names for the corresponding timbers from Malaya and the Philippine Islands respectively. They are obtainable in large sizes and are used for constructional purposes and also for ships' decking, joinery, and furniture.

SPRUCE, CANADIAN (principally *Picea glauca*).—Eastern Canada. An almost white, straight-grained softwood, without appreciable odour or taste and only slightly resinous, closely resembling European spruce (Baltic whitewood). Not durable in exposed situations, used for general construction, boxes and packing cases, ladders, etc.

SPRUCE, SITKA (*Picea sitchensis*).—Western Canada and Western U.S.A. A high class light-weight straight-grained, softwood, non-resinous, odourless and tasteless. Remarkably strong and tough in proportion to its weight and with high elastic properties. Not durable in exposed situations. Used in aircraft construction and for masts, spars, oars and paddles.

SYCAMORE (*Acer pseudoplatanus*).—Europe, including British Isles. A fine-textured, lustrous, white or yellowish white wood, often finely figured. Used for rollers in the textile industries and for laundry work, mangles, etc.; also for furniture and decorative work.

TEAK (*Tectona grandis*).—Burma, India, Java, Siam, Indo-China. Combines the properties of durability, strength, moderate weight and hardness with a pleasing appearance. Used for a great variety of purposes including ship-building, railway carriage work and house-carpentry.

TEAK, RHODESIAN (*Baikiaea plurijuga*).—Rhodesia. A handsome reddish-brown wood finishing with a smooth, hard surface. Used principally for flooring. Not a true teak.

TURPENTINE (*Syncarpia laurifolia*).—Australia. A strong, heavy constructional timber used principally in marine engineering work.

WALNUT, AFRICAN (*Lovoa klaineana*).—West Africa. A handsome golden-brown wood of the mahogany family. Not a true walnut. Used for chair-making, cabinet work, panelling and high class joinery.

WALNUT, AMERICAN BLACK (*Juglans nigra*).—North America. Purplish or greyish brown. Used for much the same purposes as European walnut.

WALNUT, EUROPEAN (*Juglans regia*).—Europe, including British Isles. Greyish-brown, often finely figured. Used principally for furniture, cabinet work and interior decoration; also for gun and rifle stocks, air-screws and turnery.

Walnut, Queensland (*Endiandra palmerstonii*).—Australia. An ornamental wood resembling true walnut in appearance and used for similar purposes, principally in the form of veneer.

Whitewood, American, or Canary Whitewood, or Yellow Poplar (*Liriodendron tulipifera*).—North America. Pale yellow or greenish, soft, light, easily worked, takes a good polish and stains well. Used for interior joinery, mouldings, furniture parts, etc.

Whitewood, Baltic, or European Spruce (*Picea abies*).—Europe, including British Isles. Used in Scotland and the north of England for housebuilding, flooring and general joinery work, Baltic redwood being generally preferred for these purposes in the south. Slightly lighter in weight and not so durable as redwood.

Willow (various species of *Salix*).—British Isles. A light-weight, fairly soft, perishable wood. Used for cricket bats, artificial limbs, flooring, cart bottoms, crates, etc.

Yang.—For description see Gurjun.

Yew (*Taxus baccata*).—British Isles. Reddish-brown, very hard, and exceptionally durable. Barely used on account of difficulty in working, but sometimes employed for furniture, gateposts and fencing, and turnery.

Number of Cubic Feet in 1 ton of Timber.

Wood.	Cubic Feet per Ton.			
	Air-dry (15%)	Shipping dry (20%)	Green.	M.C. %.
Ash, European	51	49	43	46
Beech ,,	49	47	37	80
Redwood	64	62	34	140
Elm, Common	72	70	59	50
Mahogany, Honduras	66	64	50	58
Oak, European	49	47	33	85

Comparative Weight of Green and Seasoned Timber.

Wood.	Weight of a Cubic Foot in Lb.		
	Green.	M.C. %	Seasoned (15% M.C.)
Greenheart	83	50	65
Jarrah	68	50	53
Beech, European	60	80	46
Gurjun	62	73	46
Oak, European	67	85	46
Ash ,,	52	46	44
Yew	70	100	42
Iroko	63	—	41
Pine, Pitch	52	60	41
Teak	55	60	41
Meranti, Red	43	66	36
Elm, Common English	66	140	35
Mahogany, African	48	64	35
Fir, Douglas	37	35	33
Redwood	38	50	31
Whitewood	50	116	30
Sequoia (Californian Redwood)	55	111	29
Poplar, Black Italian	55	145	28
Pine, Sugar	51	130	27
Cedar, Western Red	30	52	24
Obeche	34	90	24

Durability of Timber.

Resistance of Fungal Attack.—Fungi are low forms of plant life, comparatively few of which attack wood. These are, however, responsible for the major part of the destruction of timber. Fungi grow by means of hair-like threads termed hyphae, which travel through the wood, ultimately reducing it to powder. Under certain conditions they may form fruiting bodies which liberate minute spores to be blown about by the wind and spread infection. Fungi require certain conditions for development. They will not attack dry wood (under 20 per cent. moisture content) nor will they attack wood which is completely saturated. Consequently, practically all timber, whatever species, is extremely durable if kept dry (*e.g.* furniture), or if kept submerged (*e.g.* pier foundations and submerged piles). The durability of timber can only be compared, therefore, when it is under conditions favourable to decay. Under such conditions the following approximate classification gives the resistance of various species to fungal attack. In all cases it applies to the heartwood, the sapwood generally being much less resistant to attack.

High Resistance or very Durable.	Moderate Resistance or Moderately Durable.	Low Resistance or Non-Durable.
Oak, European	Scots pine	Spruce
Teak	Redwood	Beech
Pitch pine (Longleaf)	Douglas fir	Sycamore
Yew	Pitch pine (Shortleaf)	Maple
Larch	Elm	Ash
Jarrah	Red pine (Canada)	Birch
Pyinkado		Western hemlock
Western Red cedar		Poplar
Sweet chestnut		Willow
Greenheart		
Sequoia		
Teak (Rhodesian)		
Turpentine		

Dry rot is the name given to the decay of timber in buildings caused by the attack of various fungi, but in the majority of cases it is due either to the fungus *Merulius lacrymans*, *i.e.* the true dry rot fungus, or to *Coniophora cerebella*, the cellar fungus, which is only capable of attacking wood that is definitely wet. Outbreaks of attack are nearly always due to excess moisture coming in contact with the wood, and the most efficient way to prevent dry rot is by proper design and construction. Unseasoned timber should be avoided and all timber should be protected from ground moisture by efficient damp-proof coursing. All under-floor spaces should be well ventilated. Where there is any risk of timber becoming damp, a preservative should be used.

For further details, see 'Dry Rot in Wood,' Third Edition, price 1*s.* net, published by H.M. Stationery Office.

Resistance to Insect Attack.—The Common Furniture beetle may attack most species of wood, but the damage to external structural timbers is not of economic importance. It is frequently found attacking rafters, joists, floor boards and other structural timbers in buildings, particularly in old outhouses, barns and sheds where it may cause considerable damage, and is of course the main source of damage to old furniture. The Death-watch beetle (*Xestobium rufovillosum*) will attack old hardwoods, such as oak, elm, walnut, chestnut, alder, and is mainly found in old oak roofing timbers. Softwoods are unattacked except where in direct contact with seasoned hardwoods. The *Lyctus* Powder-post beetle attacks the sapwood of freshly seasoned hardwoods, mainly oak, ash, walnut, elm and hickory, and never attacks softwood. It is the most serious economic cause of damage to such hardwoods during seasoning and storage. Termites are the most serious cause of insect damage to timber in tropical countries. They are not active in temperate climates, and do not exist in the British Isles. There are many different species of termites, varying in their powers of destructiveness. No timber is known to be absolutely immune from attack, although a few, such as teak, pyinkado and greenheart, are highly resistant. There are two classes of preservatives effective against termites—coal-tar creosote oil type and compounds containing arsenic. In both cases impregnation treatment is essential, brush treatment being of little value.

Resistance to Marine Borers.—There are two main organisms which attack timber in sea-water round Great Britain, the Teredo or Shipworm (*Teredo navalis*), and the Limnoria or Gribble.

The Teredo is a long worm-shaped bivalve mollusc which bores into the timber, forming tunnels up to ½-in. diameter and 24 ins. long. These tunnels are lined with a calcelareous deposit, and may be so numerous as to completely honeycomb the timber without very obvious sign of attack on the external faces. On the other hand, Limnoria only form tunnels about

½ in. deep, but these are so close together that the remaining wood is easily eroded by wave action, exposing a fresh surface to attack, and the action becomes continuous, resulting in the well-known 'waisted' appearance of piles at water level. A similar borer called Chelura is often found associated with Limnoria.

Very few timbers are immune from attack by marine borers, but the following are generally reputed to be highly resistant: pyinkado, greenheart, jarrah, teak, turpentine, opepe, ironbark.

Experiments carried out by the Sea Structures Committee of the Institution of Civil Engineers tend to show that the most satisfactory method of protecting timber in sea water is pressure creosoting, and that the efficacy of such protection is dependent on the depth of penetration of the creosote. Places of poor penetration and untreated surfaces exposed by cutting or drilling after treatment are nearly always attacked by Limnoria.

Resistance to Fire.—All woods will decompose when exposed to heat of sufficient intensity and duration, and most woods will commence to char rapidly at about 270° C. Some species are more resistant to the passage of fire than others, and generally speaking hardwoods are more resistant than softwoods, and resistance increases with the density of the wood. Teak and pyinkado are highly resistant timbers. The resistance to fire can be considerably increased by impregnation with various chemicals, and in fact with proper treatment timber can be rendered non-inflammable in that it will not flame or glow, but merely char, and will not, therefore, assist in the propagation of fire. Impregnation of the timber is necessary for such results, and surface treatment such as brushing or dipping has little value. The usual chemicals used for fireproofing timber are: aqueous solutions of ammonium phosphate, sulphate and chloride, borax and boric acid.

Some protection against the initial outbreak of fire is afforded by 'Fire-resistant paints.' Generally these consist of aqueous solutions of sodium silicate with an inert filler such as kaolin, asbestine, lithopone, mica, whiting, etc. An exception to this type is the class of fire-retardant coatings consisting mainly of calcium sulphate plaster. Fire-retardant paints are not suitable for timber exposed to the weather, and with the exception of some proprietary makes they are not compatible with ordinary oil paint.

Wood Preservatives.

Many chemicals have been suggested at one time or another for the purpose of preserving timber, but those which have found general application are remarkably few.

There are three main classes of wood preservatives, which can conveniently be described as the 'oil,' 'water soluble' and 'solvent' types.

Practically all commercial impregnation treatments are carried out with the oil or water soluble types, as the high cost of the solvent precludes the use of the third type for treatments involving high absorptions of liquid.

Coal tar creosote is the most important preservative of the oil type, having been in use for over 100 years without having been seriously challenged by any other preservative for the preservation of structural and other timbers used under the most severe and exposed conditions. Coal tar creosote for preservation purposes is usually purchased to the British Standard Specification No. 144, 'Creosote for the Preservation of Timber,' (revised and extended in 1936). In addition to the specifications, standard methods of test are also described. Three types, A, A_2 and B, are described representing the normal types of creosote produced in this country, and no superiority of any one type over another is implied. Naturally some users have preference for one or the other, but any creosote conforming to the specification can be regarded as a satisfactory preservative.

Type A oils are the heavy oils, the specification allowing a specific gravity range of 1·010 to 1·065 at 38° C. when compared with water at 20° C. As regards the distillation range not more than 78 gms. should distil over when 100 gms. are distilled under prescribed conditions up to 315° C.*

Type A_2 oils are lower in specific gravity, the range being from 0·995 to 1·065 at 38° compared with water at 20° C. The distillate up to 315° C. is allowed to reach 85 gms. When low temperature creosote is specially ordered, the lower limit for the specific gravity is given as 0·935.

Type B is the designation given to creosotes produced in Scotland, and the general requirements of the specification are the same as those for Type A_2. An extended specific gravity range is allowed for creosotes produced from Blast Furnace Tar, the range being 0·935 to 1·065 at 35° C. compared with water at 20° C. In all cases no upper limit for the tar acid content is specified.

Other substances used for the preservation of timber which come under the heading of 'oil type' preservatives include coal tar, wood tar creosote and petroleum oil.

Coal tar from a preservative point of view is by no means as efficient as the creosote derived from it. It is much less toxic to wood-destroying agencies, and being very viscous does not penetrate the wood deeply. When used alone it should be applied hot. Coal tar is sometimes mixed with ordinary creosote, and provided the mixture is not too viscous it is a satisfactory preservative for pressure treatments, although the surface of the treated wood is much blacker and dirtier than when creosote alone is used.

* Revised 1941.

Wood tar creosotes, though satisfactory preservatives, are not used to any large extent in this country, mainly because they are not produced in any quantity nor are they prepared to any standard quality.

Petroleum oils by themselves are not satisfactory wood preservatives, as they have but little toxicity. Crude petroleum or petroleum fuel oils are, however, used in admixture with creosote in the U.S.A. for the treatment of railway sleepers with satisfactory results. These oils, when used for preservative purposes, should be regarded rather as diluents than as preservatives.

Water soluble salts.—These have, in certain cases, advantages over the oil type. They are easier to transport (since the solid or concentrated form can be used) and the majority involve very little fire risk. The treated wood, after drying, can generally be satisfactorily painted with ordinary oil paint. They are also for the most part odourless. On the other hand, being water soluble they are more readily washed out of the wood, if this is used in contact with the ground, and are therefore not so suitable as the oil type for use under severe conditions of exposure. Recent developments in this type of preservative have indicated the possibility of fixing to a fairly high degree some of the salts in the wood. Potassium dichromate has generally been the chemical used for fixing purposes.

The common water soluble salts which have found application for the preservation of timber are zinc chloride, sodium fluoride, copper sulphate, mercuric chloride and arsenic compounds.

Solvent preservatives consist of toxic substances dissolved in volatile solvents, and in quite a number of these the toxic material is insoluble in water. The solvent is used as a vehicle for carrying the toxic material into the wood, where it remains after evaporation of the solvent. Being much more expensive than the water soluble type, these preservatives are usually applied by brush or other surface treatment.

Methods of Applying Wood Preservatives.

The efficient preservation of timber is dependent as much on the method of application as on the preservative itself. It is not always practicable, nor is it necessary in the majority of cases, to obtain complete impregnation of the timber, but for permanent protection it is essential to obtain sufficient penetration of the preservative to ensure that at no time will unimpregnated wood be exposed by splitting, mechanical abrasion or cutting, and to provide a sufficient reserve of antiseptic in the timber to allow for losses that take place owing to leaching, evaporation, etc. An impregnation treatment is essential in all cases where long life is required in timber exposed to conditions favouring decay (*i.e.* in all timber in contact with the ground, water or saturated air). For timber not in contact with the ground and exposed to ordinary atmospheric conditions, the environment is not so favourable to the growth of decay-producing fungi, and a surface treatment is often sufficient.

Surface Treatments.

Probably the best known and most widely used method of applying wood preservatives is a brush treatment. Except for the sapwood of the more absorbent timbers, such application results in little more than skin deep penetration. A certain amount of protection is, however, afforded to the timber, but such protection is very slight compared with that afforded by impregnation treatments, and in the case of timber kept continually under conditions favourable to decay, it is negligible. When applying an oil paint by means of a brush, it is usually applied sparingly and worked well into the timber. The same method should not be adopted with a toxic preservative. The liquid should be applied liberally and swilled over the surface, the timber being allowed to absorb as much as possible. Brushwork is not necessary except to ensure that the preservative penetrates all cracks and crevices. A long-handled tar-brush is the best type for the application of creosote.

Better penetration is effected in most cases if the preservative is applied hot, particularly in the case of preservatives of the oil type. Heating, however, is not essential, and most preservatives sold for surface application are reasonably penetrating at normal temperatures. Preservatives should not be applied while the surface of the timber is wet, and brush treating is best carried out in the summer or autumn, when the timber is in a dryer and more absorbent condition. The higher temperatures obtaining in summer also aid penetration and obviate any necessity of heating the preservative.

The covering power of a preservative depends on its viscosity and on the species and condition of the timber treated. In the case of non-absorbent timbers, such as Douglas fir or oak, a gallon of liquid will cover a larger area than in the case of absorbent timbers, such as Scots pine or beech, and a machined surface will absorb less than a sawn one.. It is of course not desirable to obtain maximum covering power with a toxic preservative. An approximate figure for one of the oil type applied at normal temperatures on machined redwood (*Pinus sylvestris*) is 400 square feet to the gallon. This is for the first coat, the covering power for a second coat applied within a few days being about twice as great. For sawn surfaces the covering power is approximately half that for machined surfaces.

Spraying is a convenient method of surface application, which probably results in a more effective covering of the timber than can be obtained by brush treatment in that the preservative is more likely to penetrate cracks and its application is more liberal. In all other respects spraying is similar to brush application.

Dipping the timber in a bath of the preservative is better than either brushing or spraying, and results in the effective coating of all surfaces of the timber.

Steeping.

By prolonging the duration of dipping, or steeping the timber, the absorption and penetration are increased, but at a very slow rate.

The penetration obtained by steeping varies considerably with the species of timber used. With resistant timber, such as the heartwood of oak, larch or Douglas fir, steeping for as long as 200 days only results in slight penetration, whereas with permeable timber, such as beech and the sapwood of Scots pine, practically complete penetration of 2–3 in. material can be obtained in this period.

An absorption of 10–12 lbs. per cub. ft. can be considered as giving sufficient protection even under conditions favourable to decay. With many species this can be accomplished in using creosote by steeping for periods varying from 1 to 20 weeks. The actual time required will depend mainly on the species and dimensions of the timber and without extensive trial no definite times of treatment can be laid down.

As a rough practical guide for steeping the more permeable species in creosote, one week per inch of thickness should give good protection. This should be regarded as only very approximate, and if steeping is practised to any appreciable extent, the time of treatment to give the desired absorption should be ascertained by trial for each individual case.

Open Tank Treatment.

In order to obtain effective impregnation by steeping, the timber must be allowed to remain in the liquid for long periods often extending into weeks. Similar, if not better, results can be obtained in 24 hours by what is known as the open tank hot and cold process. Although the apparatus used consists of a tank open to the atmosphere, the impregnation is effected by means of pressure just as if a force pump had been used, the only difference being that it is the pressure of the atmosphere which forces the liquid into the wood. This pressure is brought into effect by means of variations in temperature. Wood in an air-dry condition contains minute air spaces totalling say 40–80 per cent. of its volume. If this piece of wood is immersed in a liquid which is then heated, the air in the wood expands, and a certain amount is driven out and can be seen escaping as bubbles through the liquid. On cooling, the remaining air contracts, tending to form a partial vacuum, and the liquid is gradually drawn into the timber. In other words, during cooling the atmospheric pressure is greater than the air pressure inside the minute spaces in the timber. *Thus absorption takes place during the cooling period.* The greater the difference between the maximum and minimum temperatures the greater the difference in the pressure of the air inside and outside the timber, and consequently the greater the absorption. Immersion of timber in either hot or cold liquid maintained at constant temperature does not induce this pressure difference and, consequently, absorption is not by any means so great.

This principle can be applied to impregnate timber with preservative in a very simple manner. The method of treatment is essentially the same for all species, but the times and temperatures may be varied to obtain the required absorptions for the particular species and dimensions of the timber treated. Actual experience will determine which treatment is the most suitable. The general method is as follows:—

The timber is immersed in the cold creosote, which is then gradually heated up to 180°–200° F. and maintained at this temperature for about 1–2 hours, depending on the size and species of timber being treated. The creosote is then allowed to cool, care being taken that the timber is kept completely submerged during the cooling period, and if necessary, more preservative added to maintain the level. The timber is then removed and allowed to drain, after which it is ready for use. The total time taken depends on the size and species being treated, but it is usually found convenient to do one run every 24 hours. Heating is started in the morning and lasts 3–4 hours, after which steam is turned off or the fire drawn, and the whole left to cool until the following morning. Thus the operator need only be in attendance for several hours each day. It may sometimes be found possible to carry two runs during the 24 hours, but this depends on the rate of heating and cooling of the tank and the species of timber under treatment, and can only be decided by actual trial.

For the treatment of fence posts, efficient results can be obtained by what is known as the 'butt treatment.' In this case the lower end only of the post is given an open tank treatment, and the upper end is either brushed with, or dipped in, the creosote. Thus the part of the post most liable to attack by fungus, *i.e.*, at and below ground level, receives the heavier treatment. All the apparatus required consists of an iron drum which can be heated by a fire underneath.

Pressure Treatment.

The most effective method of impregnating timber with preservatives is by means of pressure, and this process is always used when the quantity of timber to be treated justifies the capital outlay of a pressure plant. It is much quicker than the open tank process, the treatment is more easily controlled and better impregnation is obtained with the more resistant species. Practically all telegraph poles and railway sleepers used in Great Britain are impregnated under pressure.

The process consists essentially of impregnating timber in closed cylinders with a preservative under pressure. The cycle of pressure differences both below and above atmospheric and the duration of the pressure periods may be and are varied considerably in order to obtain the desired results, but pressure treatments are generally classified under the following types:—

(1) Bethell or Full Cell Process.
(2) Rueping or Empty Cell Process.
(3) Boulton or Boiling under Vacuum Process.

The Bethell or Full Cell Process is the oldest process in use, having been introduced about 1838, and it is probably the most widely used pressure process at the present day. The timber is enclosed in a cylinder and subjected to a vacuum of 15 to 25 ins. of mercury for a period varying from ½ to 2 hours. The cylinder is then flooded with preservative and pressures of 100 to 200 lb. per sq. in. applied for a period of ½ to 5 hours, depending on the species of timber being treated and the results desired. The cylinder is then emptied of preservative and a final vacuum applied.

The Rueping or Empty Cell Process was introduced about 1904. It differs from the Full Cell Process in that pressure of the air in the cylinder containing the timber is increased and not diminished. The usual oil-pressure treatment is then applied and on release of the pressure the compressed air in the cell spaces expels the excess creosote, leaving a coating only on the cell walls. Thus a deep impregnation is obtained without a heavy absorption of creosote.

The Boulton or Boiling under Vacuum Process is really a seasoning process and is applied to green timber. The timber is immersed in creosote at a high temperature and a vacuum applied for the purpose of removing moisture from the wood. After 12 to 24 hours of this treatment an ordinary full cell or empty cell pressure treatment is applied.

The Full Cell Process is the most suitable for general purposes and for timber exposed to bad conditions and which is required to have as long a life as possible. The Empty Cell Process is only suitable for timbers which are very easily treated and where conditions of exposure are not too severe. It is not suitable for instance for marine timbers. The Boiling under Vacuum Process is suitable for timber which is difficult to impregnate and which is used in large sizes. Its particular application is for marine piling of Douglas fir.

Different species of timber vary considerably in their resistance to the penetration of liquids. Larch and oak heartwood are practically impervious, whilst beech and the sapwood of most species can be completely impregnated. Douglas fir and spruce are fairly resistant to impregnation, but the resistance of spruce seems to vary considerably in timber from different localities. Most of the non-durable timbers, such as sycamore, birch, and the sapwood of Scots and Corsican pines, are very easily impregnated either by means of pressure or by the open tank hot and cold process, and it is with these species that the economy of a preservative treatment is most pronounced.

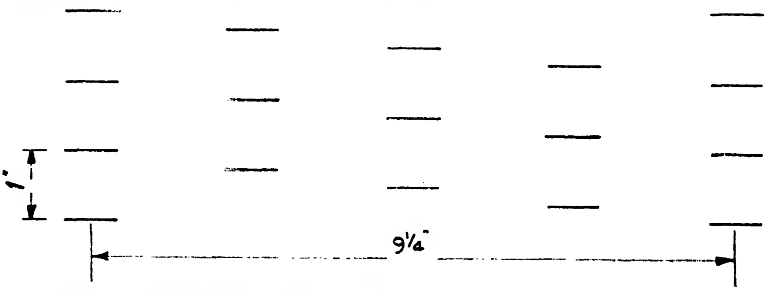
'Pressure Creosoting of Timber' is now the subject of British Standard Specification No. 913—1940.

Incising.

Some timbers, such as Douglas fir, are difficult to treat even under pressure, as not only do they absorb but small amounts of preservative, but the absorption during treatment is very irregular. Several methods have been suggested for improving the treatment of such woods, most of which entail perforating the longitudinal surfaces of the timber.

The method most commonly employed consists of making narrow incisions, usually about ¾ in. deep, over all longitudinal surfaces of the timber by passing it under a spring loaded roller from which project the incising knives. The knives are fairly thin, and are designed to part the fibres of the wood, rather than sever them, so that the strength is not seriously impaired by the process. The spacing of the incisions varies somewhat, but an average arrangement suitable for Douglas fir is shown in the diagram, fig. 1.

Such an arrangement allows the creosote to penetrate the ¾ in. depth of the incisions and then spread in each direction so that the wood is eventually uniformly penetrated to the depth of the incisions.



DEPTH OF INCISIONS 3/4" FIG. 1.

Diffusion Methods.

These methods of treatment are applied to timbers in green or damp condition, and their main use is for timbers already erected which cannot otherwise be treated. Several methods of application have been suggested, but they all rely on the diffusion of a water soluble preservative into the wood. One method consists of injecting the preservative as a paste into the wood through a hollow incising tool. A similar method is to bore holes in the timber, which are then filled with the preservative paste. Another but slightly different method consists of holding a quantity of the preservative in contact with the wood by means of a waterproofed bandage.

Seasoning of Timber.

Why Timber must be Seasoned.

Freshly felled timber contains a large quantity of moisture. In ash, for example, one-third of the total weight is water, and in the sapwood of some coniferous trees, only one-third of the weight is timber. If kept as logs, most timbers dry very slowly indeed, but if converted to scantlings, planks or boards and exposed to air in this form, they lose their moisture more freely, and sooner or later most, but not all, of the original moisture of the tree evaporates. This process is accompanied by shrinkage of the wood, hence if unseasoned timber is employed, the work for which it is used will lack stability. Floors and framing laid on such material will sink, joints in structures will become loose, and in pattern-making and decorative work, shrinkage will cause loss of shape and size, and the wood will probably split and crack.

Additional advantages of seasoning are that wood becomes lighter and stronger, more durable, better able to take preservatives, paints and polishes, and its machining qualities improve.

Shrinkage of Timber.

The difficulties caused by the shrinkage of timber during seasoning are intensified by the fact that the movement is not the same in all directions. The greatest shrinkage occurs in a direction tangential to the growth rings of the tree. In the radial direction of the tree the shrinkage is roughly only half as much, while along the grain, that is the longitudinal direction of the tree, the movement is relatively very small, say only 2 to 3 per cent. of that tangential to the rings. The magnitude of the average shrinkage of timber can be seen from the chart on p. 290. The differences in directional shrinkage are the main cause of distortion which occurs in drying. The square cross-section of a piece of wood cut from a log tends to become diamond shaped in drying if the growth rings are not roughly parallel to the faces of the square. A board or plank cut other than diametrically from the log tends to cup or curve across its width in drying. A board cut along a radius or diameter of the log, that is a quarter-sawn board, remains flat in drying. Such a board tends to shrink in width least in drying and to move least in subsequent service and is thus most suitable for use where dimensional variation is undesirable as in large panels, etc.

Moisture Content.

The only satisfactory way of expressing the degree of seasoning or dryness of timber is by moisture content. Terms such as ' air-seasoned ' or ' kiln-seasoned ' may bear some conventional meaning, but they are too loose to use as specifications. The phrases ' thoroughly seasoned,' ' well seasoned ' and ' bone dry ' are strongly to be deprecated.

Moisture content is the amount of water in a piece of timber expressed as a percentage of the weight of dry wood substance. It is generally determined as follows :—

(1) Cut a full cross-section from the board, ½ in. long in the direction of the grain and at a distance of 9 in. from the end.

(2) Weight this sample as quickly as possible. This is the Initial Weight W.

(3) Dry in a ventilated oven at 100° C. (212° F.) until constancy of weight is attained. This is the Dry Weight D.

(4) Moisture content $= \frac{W - D}{D} \times 100$ per cent.

Where it is inconvenient to cut the timber (as in large baulks, sleepers, etc.) a sample can be obtained with an auger or bit, but it is advisable to collect the chips in a bottle to guard against loss of moisture before they can be weighed.

Electrically operated machines are on the market that indicate moisture contents from about 6 per cent. to 25 per cent. If used with intelligence and a knowledge of their limitations, they are sufficiently accurate for commercial work.

Moisture Content of Timber for Various Purposes.

The degree of seasoning desirable in timber depends on the use to which the timber will be put, and as a practical issue, upon the amount of shrinkage permissible in the finished article. For marine piling, and generally where timber is to be in contact with water, there is obviously no point in drying before use, unless it has first to be creosoted, and even then, only a sufficient thickness of seasoned wood is necessary for the required penetration to be obtained.

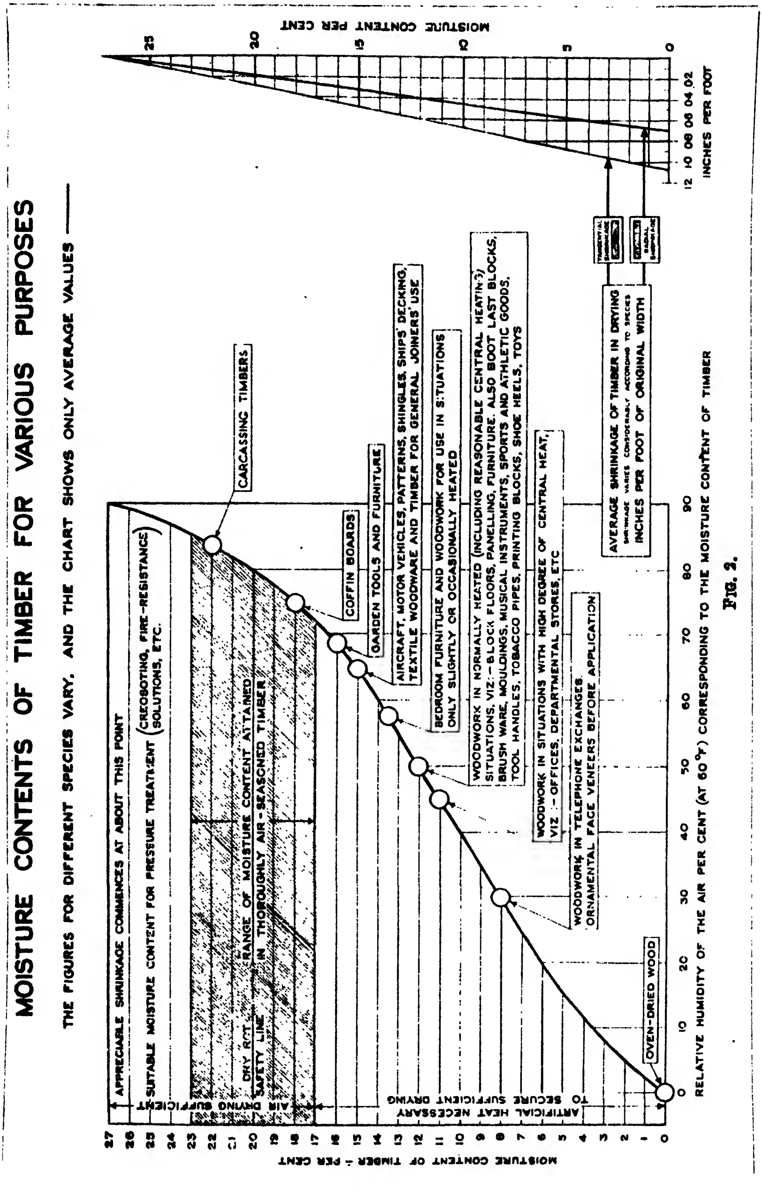

MOISTURE CONTENTS OF TIMBER FOR VARIOUS PURPOSES
THE FIGURES FOR DIFFERENT SPECIES VARY, AND THE CHART SHOWS ONLY AVERAGE VALUES
APPRECIABLE SHRINKAGE COMMENCES AT ABOUT THIS POINT
SUITABLE MOISTURE CONTENT FOR PRESSURE TREATMENT (CREOSOTING, FIRE-RESISTANCE SOLUTIONS, ETC.
DRY ROT SAFETY LINE
RANGE OF MOISTURE CONTENT ATTAINED IN THOROUGHLY AIR-SEASONED TIMBER
AIR DRYING SUFFICIENT
ARTIFICIAL HEAT NECESSARY TO SECURE SUFFICIENT DRYING
CARCASSING TIMBERS
COFFIN BOARDS
GARDEN TOOLS AND FURNITURE
AIRCRAFT, MOTOR VEHICLES, PATTERNS, SHINGLES, SHIPS' DECKING, TEXTILE WOODWARE AND TIMBER FOR GENERAL JOINERS' USE
BEDROOM FURNITURE AND WOODWORK FOR USE IN SITUATIONS ONLY SLIGHTLY OR OCCASIONALLY HEATED
WOODWORK IN NORMALLY HEATED (INCLUDING REASONABLE CENTRAL HEATING) SITUATIONS, VIZ:—BLOCK FLOORS, PANELLING, FURNITURE. ALSO BOOT LAST BLOCKS, BRUSH WARE, MOULDINGS, MUSICAL INSTRUMENTS, SPORTS AND ATHLETIC GOODS, TOOL HANDLES, TOBACCO PIPES, PRINTING BLOCKS, SHOE HEELS, TOYS
WOODWORK IN SITUATIONS WITH HIGH DEGREE OF CENTRAL HEAT, VIZ:—OFFICES, DEPARTMENTAL STORES, ETC
WOODWORK IN TELEPHONE EXCHANGES ORNAMENTAL FACE VENEERS BEFORE APPLICATION
OVEN-DRIED WOOD
AVERAGE SHRINKAGE OF TIMBER IN DRYING
SHRINKAGE VARIES CONSIDERABLY ACCORDING TO SPECIES
INCHES PER FOOT OF ORIGINAL WIDTH
TANGENTIAL SHRINKAGE
RADIAL SHRINKAGE
MOISTURE CONTENT PER CENT
INCHES PER FOOT
MOISTURE CONTENT OF TIMBER - PER CENT
RELATIVE HUMIDITY OF THE AIR PER CENT (AT 60°F) CORRESPONDING TO THE MOISTURE CONTENT OF TIMBER

Fig. 2.

The diagram on p. 290 shows the moisture contents that obtain in various environments, and for important work, the timber should first be dried to the appropriate figure. The diagram also gives data from which specifications may be framed. For general joiners' uses, as an example, a moisture content of 15 per cent. is satisfactory, but for furniture and decorative woodwork, 12 per cent. is a more suitable figure. Shrinkage consequent upon moisture change can be estimated from the scale on the right.

An important point in builder's work is immunity from dry rot and other wood-destroying fungi. Unless the moisture content in the timber exceeds 20 per cent. these will not attack the wood, and if the joists, etc., are seasoned to this figure or less prior to installation in adequately ventilated buildings, safety is ensured.

Methods of Seasoning Timber.

Air Seasoning.—Round timber for telegraph poles, pit props, etc., should have the bark removed to accelerate the seasoning, and be stacked in some open form such as self-crossing, with spaces between adjoining pieces in each row.

Timber sawn into boards, planks, etc., is piled in the open or in a ventilated shed so that air can circulate freely over its surfaces.

The stacks should be raised about 12 ins. off the ground by level foundations, and for rapid drying the pile should not exceed 6 ft. in width; length and height, consistent with convenience and safety, are immaterial. Between individual piles a space of 9 in. to a foot will ensure adequate air movement, and it has been found that special orientation of the yard with respect to the prevailing wind is unnecessary.

Within the stack the boards should be laid with slight gaps between them, and the rows should be separated by piling sticks about 1 in. square. Certain refractory timbers such as oak will benefit if the sticks are only ½ in. thick, but there is little advantage in increasing the thickness beyond 1 in. even with free-drying softwoods. The distance between the sticks depends on the timber, and as a guide, these may be taken as 9 to 18 ins. with ½ in. timber; 1 ft. to 2 ft. with 1 in.; 1 ft. 6 ins. to 2 ft. 6 ins. with 2-in. stocks and 2 to 4 ft. with thicker material.

If the timber is stacked in the open, it is advisable to fit a good roof to prevent rain penetrating to the middle of the pile. It is very necessary that the ground beneath the stacks be kept clear of weeds and odd scraps of wood. These latter may decay and may infect the timber with wood-destroying fungus.

The moisture content attained by timber in air-drying in the open varies with the season, 20 per cent. being a fair average with upper and lower limits of about 23 per cent. and 17 per cent. Whilst the drying times depend on the time of year at which the timber is piled, the table gives approximate data for freshly-cut 2-in. material of a few common species under favourable conditions.

Species.	Time (Months).	Species.	Time (Months).
Oak	18	Ash	6
Beech	9	Elm	6
Pitch pine	6	Scots pine	2

Kiln Seasoning.—The relative merits of air seasoning and kiln drying are frequently contrasted, and there are many misconceptions regarding both methods. It can be stated definitely, however, that seasoning and drying are synonymous terms for all practical purposes and the manner in which moisture leaves the wood is the same in the two methods, the difference being merely one of speed and temperature. There virtually is no way of distinguishing between air-dried stock and that which has been seasoned in a kiln under correct conditions and proper control, assuming the degree of seasoning, that is the moisture content of the timber, is the same.

There is no strict line of demarcation between the methods, for the two are frequently combined. Thus, freshly-converted timber can be dried directly in a kiln to say 12 per cent. moisture content, or it can be partly air-seasoned first and then finished off in a kiln or in a warm store, which is virtually a low temperature kiln.

One point must always be remembered; air-seasoned timber is insufficiently dry for interior uses, and if shrinkage is to be avoided, it must be given further treatment.

Essentials of a Timber Drying Kiln.

The three essentials of a kiln are: *Heat, Humidification* and *Circulation*, the terms applying to the air, which is the medium with which most dryers operate.

Heat is needed to evaporate the moisture from the timber, and the use of high temperatures increases the rate of movement to the surface from the interior.

Humidification of the air is necessary to effect a balance between evaporation at the surface of the wood and the internal transfusion of moisture to that surface.

Circulation is required to move air continuously over the surfaces of the boards. If it remained stagnant in contact with the undried timber, the air would soon become too humid and drying of the timber would cease.

In most kilns, a large proportion of the air after passing once through the timber is recirculated, but some fresh air is constantly introduced for ventilating the kiln, otherwise the air as a whole would become too humid. The volume introduced for ventilation is, however, small.

Temperature and humidity should be under the complete control of the operator, and the air movement should be adequate and uniform.

Types of Timber Drying Kiln.

Progressive Kilns.—The progressive kiln usually consists of a tunnel from 60 to 300 ft. or more in length along which the timber, which is piled on trucks, is moved by stages. Below the trucks in a basement, steam heated coils run for about two thirds the length of the kiln, and there are generally steam sprays for humidification purposes.

Heated air rises from the coils at one end of the kiln, passes in a more or less horizontal direction along the kiln through the timber stacks, and falls back into the basement at the opposite end. It then returns along this basement and is again heated for recirculation. As the air passes through the piled timber, it becomes cooler due to the heat given up in evaporating moisture and more humid owing to the moisture taken from the wood.

At one end of the kiln therefore, the air is hot and relatively dry and in passing from one end to the other it gradually becomes more humid and cooler. The timber is put in the kiln at the wet or cold end, and is moved by stages to the other, by which time, if the conditions are suitable, it will have dried. The air movement can be accelerated by fans and the drying rate thus speeded up.

Progressive kilns depend for their efficient working on a continuous supply of timber of the same species and thickness, and will not work satisfactorily if only partly full. Generally speaking, production conditions are on too small a scale in England to lend themselves to the efficient use of progressive kilns.

Compartment Kilns.—In the compartment kiln, the load is not moved until drying is completed, and the air temperatures and humidities are varied to suit the current moisture content of the stock. Such dryers are under more exact control than progressive kilns, and whilst it is not good practice to mix species and dimensions in any one charge, these kilns can be used to dry a wide variation of timbers and sizes.

Kilns are usually built of brick, but concrete, timber, asbestos sheeting or non-corrosive metals have been successfully employed. The walls must be reasonably heat- and damp-proof.

Compartment kilns are classified according to the means by which circulation is produced. In natural draught dryers, air movement is induced by convection currents. The type is not efficient for handling very wet timber, but in some of its forms, is suitable for completing the seasoning of partly-dried stock.

Forced circulation kilns are fitted with fans for moving the air, and are designated by the type and disposition of these parts. The external fan pattern has a centrifugal blower that forces the air into the stacks through a duct and exhausts it for re-circulation through a second trunk. In the internal fan kiln, propellers are used to keep the air in motion. The fans are carried either on one shaft running lengthwise or on several spindles placed across the kiln, and the air is deflected into the stack by baffles or grids. The general practice is to use steam-heated coils for heating and steam sprays to control humidity, and in the case of external fan kilns, these are usually in unit form embodied in the fan casing or in the ducting.

Internal fan kilns can be arranged with the fans, heating coils and humidifying sprays in a basement below the timber or they can be placed above the piles. The latter is generally preferable, since the timber can be piled on a solid floor and the expense of excavating a basement is obviated. The return of condensate from the heating pipes is facilitated by the overhead construction.

The essential of a fan kiln is that the air blown into the stack should be uniform in rate of flow, temperature and humidity along the whole length and subject to this proviso, the methods employed are unimportant.

Kiln Operation.

Air conditions in a kiln should be regulated according to the species being dried, the thickness of the stock and the moisture content of the timber. An operation schedule based on time alone is unsound and may lead to disastrous results. Such a method can sometimes be evolved after long experience with a particular class of timber, but where the loads vary in species and moisture content, it is strongly to be deprecated.

Successful kiln operation depends on a sound knowledge of the principles of drying, and should never be regarded as a rule-of-thumb process.

KILN DRYING TIMES.

The times given in the table are for drying 2-in. timber from the freshly felled state to a uniform moisture content of 12 per cent. in a moderately efficient fan kiln, the timber being then in a fit condition for immediate use for any purposes including resawing or moulding. Where the needs are not so exacting, or where uniformity of moisture content and stress-free state are unimportant the times can be reduced appreciably.

Species.	Time (weeks).	Species.	Time (weeks).
Oak	10–12	Ash	4–5
Elm	4–5	Pitch pine . . .	3–4
Beech . . .	5–6	Scots pine . . .	1–2

STEAMING OF TIMBER.

The steaming of fresh timber is commonly practised in the belief that it accelerates seasoning. In practical terms, this is erroneous, although the process is frequently of value in sterilising the wood and in producing a more pleasing colour.

Timber attacked by wood-destroying beetles and fungi can be sterilised by a suitable short kiln treatment. This kills all insects, grubs and eggs or spores, but does not necessarily render it immune from re-infestation.

STORAGE OF LOGS AND TIMBER.

The storage of timber in log form is common practice, and in broad terms is fairly safe. Certain non-durable woods such as beech, ash, poplar, etc., are liable to fungus attack, whilst in valuable logs, end-splitting may seriously reduce their value. The use of a water-proof end coating will reduce splitting, and if the paint has toxic properties, the danger from fungus is lessened. Both dangers are eliminated by keeping the logs under water. Where water storage is impracticable, the timber should be placed on skids and not allowed to lie on the bare ground.

On no account should wet sawn timber be closely stacked for any appreciable time, for such conditions may lead to staining and to the growth of wood-destroying fungi.

Boards that have been air-seasoned (*i.e.* to a moisture content of about 20 per cent.) may safely be bulk-piled, but the ingress of rain must be prevented. Material dried to 15 per cent. or less will absorb moisture if stacked in the open air, and should for preference be stored in a warmed shed. Where this is impracticable, the absorption rate can be greatly reduced by bulk piling and covering the stack with tarpaulins.

Strength Properties of Timbers.

In addition to wide natural variation the strength properties of timber are sometimes considerably influenced by its moisture content, by the conditions under which it is tested and by knots, cross grain and other features, as found in any particular piece. In consequence, for the derivation of basic data for comparing the strength properties of various species of timber the conditions of test are standardised, the timber is tested at a specific moisture content, and, owing to the impossibility of matching defects, the tests are made on clear specimens of timber, *i.e.* pieces cut straight with the grain and selected free from knots, splits, etc. The standard test pieces are cut from planed sticks 2 ins. square in cross section, and the tests are, briefly, as follows:—

STATIC BENDING.

The test piece is 30 ins. long and is supported over a span of 28 ins. and centrally loaded, the rate of increase of deflection at the centre of the span being 0·105 in. per minute. Readings of load and deflection are continued past maximum load, until either the load has fallen to 200 lb. or a deflection of 6 ins. has been reached. The quantities computed are equivalent fibre stress at maximum load (sometimes referred to as Modulus of Rupture), modulus of elasticity and work expended in bending (*a*) to the point of maximum load (*b*) over the whole range of deflection. The work expended in bending is a measure of the resilience of the timber.

IMPACT BENDING.

The 30-in. specimen on the same span of 28 ins. is broken by a weight of 50 lb. falling, in vertical guides, on to the specimen at the centre of the span, from heights starting with 2 ins. and increasing 1 in. at a time to 12 ins. then by 2 ins. at a time until failure occurs. The maximum drop of the 50 lb. weight required to break the specimen is a measure of the strength of the timber to resist suddenly applied loads.

STRENGTH PROPERTIES OF SMALL CLEAR SPECIMENS OF TIMBER.

Standard Name		Shrinkage Green to Air-dry		Static Bending				Impact Bending	Compression Parallel to Grain	Hardness		Shear Parallel to Grain		Cleavage	
		Radial	Tangential	Equivalent Fibre Stress at Maximum Load	Apparent Modulus of Elasticity	Work in Bending: To Maximum Load	Work in Bending: Total	Maximum Drop	Maximum Crushing Strength	Side	End	Plane of Failure Radial	Plane of Failure Tangential	Plane of Failure Radial	Plane of Failure Tangential
Softwoods		%	%	lb per sq in	1000 lb per sq. in.	Inch lb per cu in.	Inch lb per cu. in.	Inches	lb. per sq. in	lb	lb	lb. per sq. in.	lb. per sq. in.	lb. per inch	lb. per inch
Cedar, Western Red (Canadian)	GREEN	1·1	2 2	5300	1050	4 9	8·2	16	2770	270	430	670	720	140	140
	AIRDRY			7600	1210	5·3	8·6	17	4850	320	680	740	740	130	170
Douglas Fir (Canadian) . .	GREEN	2 5	4 0	8100	1770	7 1	20 5	26	3850	520	610	910	940	220	220
	AIRDRY			14800	2260	11 8	28·2	34	8480	810	980	1470	1500	230	280
Hemlock, Western (Canadian)	GREEN	2 6	4 1	6900	1470	6 6	16·7	22	3560	460	560	710	790	190	210
	AIRDRY			11100	1690	9·3	17·6	26	6390	580	940	940	960	190	210
Pine, Longleaf Pitch . .	GREEN	2 6	3 9	8700	1600	8 9	32·4	35	4300	590	550	1040		210	
	AIRDRY			14700	1990	11 8	21 9	34	8440	870	920	1500		270	
Pine, Scots	GREEN	2 3	4 8	5980	1240	5·6	25 4	26	3020	440	450	750	750	210	220
	AIRDRY			11970	1590	11 2	19·0	31	6640	680	800	1430	1380	260	330
Redwood (European) . .	GREEN	2 1	4 4	6550	1240	—	—	26	3140	430	450	800	790	190	180
	AIRDRY			11160	1420	—	—	24	6360	550	740	1360	1280	230	250
Spruce, Canadian . . .	GREEN	2 0	4 1	5500	1230	6 9	16 5	21	2640	310	380	740	780	160	170
	AIRDRY			9370	1460	7 9	13·8	22	5500	450	650	1070	1110	230	240
Spruce, European (Imported) .	GREEN	2 1	4 6	5190	1060	—	—		3050	360	390	680	700	—	—
	AIRDRY			10980	1670	—	—		6570	440	510	1240	1090	—	—
Spruce, Sitka (Canadian) . .	GREEN	2 2	3 7	5400	1370	4 8	16 8	20	2550	320	400	610	660	150	180
	AIRDRY			10900	1750	10 4	22 8	26	5600	550	780	1090	1000	200	240
Hardwoods															
American Whitewood . .	GREEN	2·2	3 9	5400	1090	5·4	8 9	18	2420	340	390	740		220	
	AIRDRY			9200	1500	6 8	11 2	20	5290	450	560	1100		280	
Ash (English) . . .	GREEN	2 9	7 6	8930	1500	17·7	45 2	50	3750	920	1020	1150	1220	360	460
	AIRDRY			15130	1360	19·2	41·8	43	6990	1490	1760	1910	1960	620	660
Ash, American . . .	GREEN	4·7	7 5	9500	1430	14·2	34·6	36	4100	910	980	1320		340	
	AIRDRY			14700	1710	15 5	29 1	37	7240	1260	1670	1930		460	
Ash (European) . . .	GREEN	—	—	9420	1360	—	—	31	—	1100	—	—	—	430	
Balsa	AIRDRY	Heartwood		2790	470	—	—	—	1680	—	—	—	—	—	—
	AIRDRY	Sapwood		5310	710	—	—	—	3750	—	—	—	—	—	—
Beech (Home-Grown) . .	GREEN	2·8	7 6	5930	1520	13 2	26·3	36	3860	960	1080	1080	1340	400	580
	AIRDRY			16210	1950	17 4	27 4	45	7870	1440	1820	1840	2230	430	620
Birrh (Canadian Yellow) . .	GREEN	3 7	4 4	8100	1530	20·6	49 5	51	3310	880	960	1140	1290	350	430
	AIRDRY			17250	2270	20 9	44 6	61	8880	1430	1750	1970	2300	540	670
Elm	GREEN	2 6	4 9	5530	810	7 6	13·9	26	2360	760	800	1000	1050	330	380
	AIRDRY			9310	1090	8·8	14·6	23	4740	820	1040	1600	1630	350	420
Elm, Rock (Canadian) . .	GREEN	3 1	5 0	9900	1400	25 0	75 6	89	4380	1430	1370	1580	1600	520	550
	AIRDRY			16550	1800	29 4	66 8	71	8000	1710	1870	2100	2210	530	520
Elm, Wych . . .	GREEN	2 7	7 0	9450	1450	13 8	25·1	32	4250	890	970	1090	1040	400	460
	AIRDRY			14510	1640	17 8	32 4	43	6070	1010	1350	1680	1620	510	610
Greenheart	GREEN	2 7	3 7	19290	2970	12 7	35 3	46	10500	2110	2160	1230	1420	420	580
	AIRDRY			30530	3400			62	14940	2650	—	2690	2980	370	600
Gurjun (Burma) . . .	GREEN	2 1	5 0	10040	1760	7 8	17·8	30	5060	940	960	930	1100		
	AIRDRY			15870	2260	14 8	28 0	37	8330	1230	1380	1400	1720		
Hickory	GREEN	4 1	5 7	11100	1570	26 1	74·6	88	4480			1280			
	AIRDRY			19200	2220	22 6	67·5	77 (a)	8940			1740			
Karri	GREEN	4·8	9 5	11270	2320	9·5	24·8	45	5850	1470	1460	1190	1460	380	530
	AIRDRY			20600	2970	20·8	39 6	—	10510	1890	2080	1950	2590	500	730
Lignum Vitae . . .	AIRDRY	—	—	—	—	—	—	—	—	4580	—	—	—	—	—
Mahogany, African . .	GREEN	1 7	2 5	7600	1180	8·5	—	26	3670	510	570	1270		380	380
	AIRDRY			10700	1480	7 8	—	22	5680	790	1080	1340		380	380
Mahogany, Central American	GREEN	1·9	2 6	9200	1290	—	—	27	4540	650	750	1310		—	—
	AIRDRY			11480	1370	9·9	12 3	23	6170	700	980	1630		320	
Maple, Rock	GREEN	2 7	4 9	10100	1680	18 4	42·6	54	4570	1170	1330	1490	1750	480	600
	AIRDRY			17200	2200	19 6	44·1	55	8100	1730	2080	2230	2690	510	760
Oak (English) . . .	GREEN	2 6	7 5	8140	1290	9·4	20·4	34	3850	1050	1120	1080	1260	390	470
	AIRDRY			13340	1560	11·1	19·1	33	7210	1230	1460	1560	1960	360	500
Poplar, Black Italian . .	GREEN	1 9	6 2	5710	1060	8 8	24·0	24	2700	460	480	690	820	210	270
	AIRDRY			9890	1340	9·9	16·4	22	5230	500	740	1010	1250	260	330
Seratah, White . . .	GREEN	1 4	4 5	8480	1610	9·9	20·1	27	4410	640	630	—	—	250	290
	AIRDRY			11800	1790	12·5	20·4	29	7190	660	620	—	—	280	290
Teak, Burma	GREEN	1 2	2·2	11430	1670	9·3	27·1	35	5870	1030	910	1030	1170		
	AIRDRY			14300	1830	10·7	24·8	26	8320	1130	1070	1250	1460		
Turpentine	GREEN	6 2	14 6	10560	1640	8 0	18 9	34	5760	1370	1320	1310	1390	420	440
	AIRDRY			22340	2410	19 2	20·8	41	11500	2900	3320	2150	3280	700	980

COMPRESSION PARALLEL TO GRAIN.

The specimen is 8 ins. long and is crushed between parallel plates. The maximum load is recorded and the maximum crushing strength computed therefrom.

SHEAR.

The specimen is 2 ins. long (*i.e.* a 2-in. cube) and is loaded so as to produce shear along the grain, tests being made with the plane of shear parallel to the growth rings and at right angles to them.

HARDNESS.

Measurement is made of the load required to indent the timber by means of a hemispherically headed plunger 0·44 in. diameter to a depth equal to the radius of the head.

CLEAVAGE.

The specimen is 3¾ ins. long with a notch ⅜ in. deep across one end. Hook grips inserted on the opposite sides of the notch enable the specimen to be subjected to a tensile force at right angles to the grain which is increased until the specimen splits. The splitting load is recorded.

For full details regarding these tests see 'Project I. Mechanical and Physical Properties of Timbers. Tests of Small Clear Specimens' by C. J. Chaplin, published by H.M. Stationery Office.

Above approximately 26 per cent. moisture content the strength of wood is uninfluenced by moisture changes. Below this value most strength properties vary inversely with the moisture content, the relationship being approximately logarithmic. The practice is, therefore, to record the strength of timber under two standards conditions: (i) as near as possible freshly felled (moisture content between 50 and 100 per cent., depending on the species) and (ii) at 12 per cent. The former relates in practice to all timber exposed to the weather and the latter to timber in a centrally heated building. From these data the strength at other moisture contents may be calculated.

The accompanying table gives the average values of strength and shrinkage of a few important timbers. Many other species have been tested by various authorities under the same system of standardised tests, the results of most of these tests will be found in the following publications:—

'A Handbook of Home-Grown Timbers," H.M. Stationery Office, London.

'A Handbook of Empire Timbers.' H.M. Stationery Office, London, 1939. Price 3*s.* 6*d.*

'Strength and Related Properties of Woods grown in the United States.' Technical Bulletin, No. 479. U.S. Dept. of Agriculture, Washington, D.C. Price 25 cents.

'Mechanical Properties of Canadian Woods.' Forest Service Bulletin No. 82, Dept. of the Interior, Ottawa, Canada.

'The Physical and Mechanical Properties of Woods grown in India.' The Indian Forest Records, vol. xvii, part 10. Govt. of India, Delhi. Price 7*s.*

It is emphasised that these figures are the ultimate strengths and other properties of small clear specimens, and as such are not to be confused with safe working stresses for which see p. 296.

THE EFFECT OF SEASONING ON THE STRENGTH OF WOOD.

The influence of seasoning on the strength properties of wood is shown by the difference between the values of each property at the two moisture contents in the table of mechanical and physical properties. The effect is most pronounced in the case of the crushing strength parallel to the grain, which is increased by approximately 100 per cent. in drying from 26 to 12 per cent. moisture content. Other properties are increased by less amounts and in certain cases, notably the strength in impact bending, the property is sometimes slightly reduced by seasoning. This latter irregular behaviour is because ability to withstand impact loads depends not only on strength but also on ability to absorb the energy of the blow by deflection. Timber at low moisture contents is stiffer and therefore not capable of yielding so far without breaking. These progressive changes in strength have been found to continue to moisture contents as low as 5 per cent. and probably also continue to zero moisture (oven-dry).

Providing the timber is dried to the same moisture content and as long as the drying schedule employed in the kiln is suitable for the timber, these changes in strength are the same whether drying takes place in the open air or under controlled conditions. The employment of an ununsuitable seasoning schedule involves the risk of impairing the resilience or toughness of the timber, and also, with certain species, of developing honeycombing, case hardening or collapse.

SAFE WORKING STRESSES.

As the strength of structural timber is largely influenced by knots and other defects it is necessary to define the extent to which these may be permitted before safe working stresses can be evaluated. It follows that the derivation of safe working stresses is greatly facilitated if timber is marketed in definite grades based on defects (as distinct from blemishes). Such is the position in America where the practice has been adopted of grading timber according to stated schedules of defects and evaluating safe working stresses for each grade. British Standard Specifications (B.S. 940, Parts I and II) have been published detailing stress grades for the species of timber commonly used for structural purposes in this country. So far, however, timber graded to these specifications has not been available through normal trade channels.

The Strength of Poles.

Poles in the line are subjected to overturning moments arising from (1) the load at the top due to the pull of the wires, or the wire pressure due to snow on the wires, and (2) wind pressure causing a load distributed along the length of the pole and on sleet-covered wires. Theoretically the maximum bending moment occurs either at the ground line, or, if the taper on the poles is sufficiently acute, at the section at which the diameter is 1½ times the diameter of the top. In practice poles more often fail a little way above the ground line, which suggests that the stresses set up differ slightly from the theoretical.

In testing poles for their ultimate strength the theoretical stress distribution of service conditions is obtained approximately by supporting the pole at each end and applying a single concentrated load at the point corresponding to the ground line. As the load on a pole is greatest at the wettest time of the year, when, consequently, the pole in, and just above, the ground is saturated with water, it is necessary, in order that test results shall indicate the strength of the pole under the worst conditions, that the poles should be soaked in water prior to test.

The following results have been obtained from tests of creosoted and uncreosoted poles carried out under the above conditions at the Forest Products Research Laboratory :—

Species.	Ultimate Stress in Bending, Lb. per Sq. In.	Weight, Lb. per Cub. Ft. at 30% Moisture Content.
1. Redwood (Scots pine, *Pinus sylvestris*) . .	7,980	37
1. ,, Home grown (,, ,,) . .	7,750	37
1. Home grown Corsican pine (*Pinus laricio*) .	8,150	37
1. ,, ,, Larch (*Larix deciduc*) . . .	10,000	39
2. Australian Ironbark (*Eucalyptus paniculata*) .	16,600	7[illegible]
2. ,, Tallowwood (*Eucalyptus microcorys*) .	15,300	65
2. ,, Blackbutt (*Eucalyptus pilularis*) .	12,970	54

1, Creosoted. 2, Uncreosoted.

SECTION IX

PART II

ROOFS — FLOORS — PARTITIONS — STAIRS — BRICKWORK — LONDON BUILDING ACTS — FOUNDATIONS — MASONRY

ROOFS.

The Ministry of Health's Model Byelaws, Series IV, Buildings (1938), Clauses 78 to 79, relating to roofs are compulsory in many localities. Obtainable from H.M. Stationery Office, London, price 1*s.* 6*d.* net.

ROOFING MATERIALS.

The chief roofing materials are :—

(I) *Slates.*—The following terms are used as in slaters' work :—

Head : the upper or edge top of slate. *Back* : the upper or exposed surface of slate when laid. *Bed* : the lower or under surface of slate. *Tail* : the lower or bottom edge of slate. *Margin* : the part of each course of slates exposed to view (*a–b*, fig. 1). *Gauge*: the distance battens are apart from centre to centre ; required to determine the position of battens to receive the nails by which the slates are secured. *Bond.*—When a joint of two adjacent slates is immediately in the centre of the slate, the tail end of which rests upon them, the slates are said to bond. *Lap* : the distance the tail of one slate overlaps the head of second course below, when slates are nailed near the centre, or the distance the tail of slate overlaps the nail-hole of second course below when slates are nailed near the head. The lap, in practice, ranges from 2½ inches to 4 inches, and averages as 3 inches. *Holeing* : the piercing of slates to receive nails. *Sorting.*—When roof is to be covered by slates of different lengths, they are regulated to proper dimensions, so that the largest slates may be nailed near the eaves, and the smallest at the ridge. *Eaves* : the lower part of slating hanging over a wall.

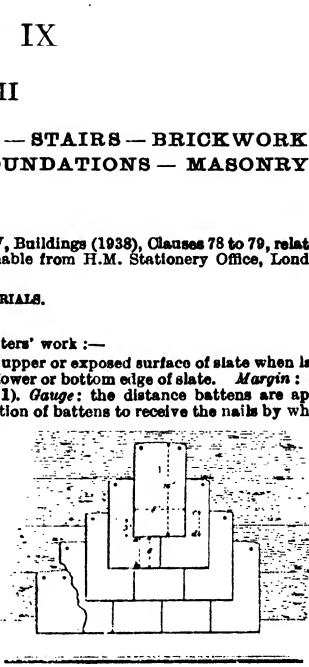


FIG. 1.

See British Standard Specification No. 680.

SIZES OF SLATES.

The sizes of slates in common use are as follows :—' Duchess ' slates, 24 inches by 12 inches, for moderate flat roofs of ⅙ pitch ; ' Countess ' slates, 20 inches by 10 inches, for roofs of ¼ pitch ; and ' Ladies,' 16 inches by 8 inches, for roofs of ⅓ pitch, the rule being, the steeper the pitch, the smaller the slate, and the more exposed the building, the greater the pitch of the roof.

Slates are laid on *boards* (fig. 1) or *battens* (fig. 2), the latter being the cheapest. Boards, ¾ inch thick for Countess slates, and rafters not more than 18 inches in the clear, 1 inch or thicker, otherwise. Battens, 2 inches by ¾ inch for Countess, with rafters 12 inches interval ; 3 inches by 1 inch for Duchess. Slates are often *holed* 1 inch or 1½ inch from the head, as in fig. 1, but are better centre-holed, i.e. just clear of head of slate below, as in fig. 2.

If a *good slate* is stood half its depth in water for several hours the moisture should not rise to the top.

Composition nails are recommended for all good work; they are stiffer than copper or zinc, present a brassy appearance, and are cheaper than copper nails. If iron nails are used, they should be galvanised, or boiled in linseed oil. Two to each slate, 1⅜ inch long for small slates, 1½ inch long for Countess, and 1¾ for Duchess slates.

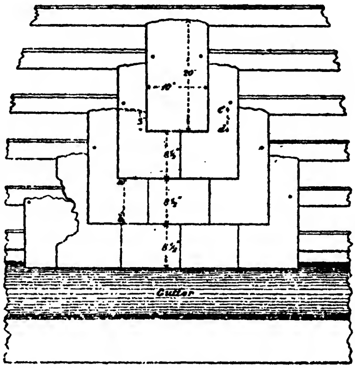


FIG. 2.

Zinc and Copper Clips.—**When a slate that is nailed to boarding gets broken and has to be repaired, narrow clips, about ½ inch wide, are pushed between the slates to catch the head of slate below. The new slate is then inserted, and the end of clip turned as a hook over the tail of the slate. When slates are nailed to battens, and the interior is accessible, they may be repaired by means of a piece of copper wire which is pointed at one end and driven into the batten or boarding at the head of under slate, passed between the side-joints of slates immediately above it, and lapped over the tail of the next slate but one above it. A man under the battens will secure a piece of copper wire to the battens, and then push the wire between the slates for the man above to clip over the tail of the new slate.**

SIZE AND WEIGHT OF SLATES.

A thousand slates means 1,200. A square is 100 super feet.

Name.	Size.	Gauge for 3-in. Lap Nailed in Centre.	No. of Squares covered by 1,000 of 1,200.	Weight per 1,000 of 1,200.	Weight* per Square.	No. to Cover One Square.	Nails required per Square.	
							Iron.	Copper.
	Ins.	Ins.		Cwts.	Cwts.		No.	Lbs.
Smalls . . .	12 × 8	4½	2·8	17½	6¼	430	860	5
Doubles . . .	13 × 6	5	2·5	18	6	480	960	6
Ladies . . .	14 × 12	5½	5	31	6¼	240	480	3¼
" . . .	16 × 8	6½	4·5	25	5·5	300	600	3½
Viscountesses . .	18 × 10	7½	6	36	6	200	400	2¾
Countesses . .	20 × 10	8½	7	40	5·7	171	342	4
Marchionesses . .	22 × 12	9½	9·4	55	6	130	260	3½
Duchesses . . .	24 × 12	10½	10	63	6	125	250	3
Imperials . . .	33 × 24	13½	25	..	..	48	96	3
Rags	36 × 24	16½	30	..	..	40	80	3½
Queens . . .	36 × 24	16¼	30	..	..	40	80	2

*** The weights are for first quality Penrhyn or Bangor; second quality weigh about one-third more. As the sizes of slates of the same name sometimes differ slightly, the size required should be given as well as the name, when of importance. This Table allows about $\frac{1}{20}$th for waste.**

See Brit. Stand. Specific. for No. 680—1936, Welsh slates; No. 690—1940 for Asbestos slates.

(ii) *Asbestos-cement* in the form of sheets (often known as 'slates') of rectangular, honeycomb, and diamond shape.

These slates are made rectangular, honeycomb, and diamond in pattern. They may be obtained in five rich colours, red, blue, grey, russet-brown, and brindled.

The original weather-resisting qualities of some Asbestos Cement Slates are guaranteed for twenty years.

These slates can be cut to fit hips, valleys, verges, round chimneys, and roof lights, etc.

Rectangular sizes, 24 ins. × 12 ins. and 16 ins. × 8 ins. Diamond and honeycomb sizes, 15¾ ins. × 15¾ ins. and 11¾ ins. × 11¾ ins.

The lap to be given is determined by the pitch of the roof, position and building, and prevailing weather conditions.

British Standard Specification No. 690—1940 relates to asbestos-cement slates and also to unreinforced flat sheets and corrugated sheets.

(iii) *Roofing Tiles.*—Roofing tiles may be *plain* (i.e. rectangular and almost flat), or they may be wavy, as *pantiles*, or they may have grooves or other structures which enable them to *interlock* with one another. Interlocking tiles are much heavier than plain tiles, but form a more weather-proof roof. (See also pp. 367 and 369.)

The unit of measure for roofing tiles (when fixed) is a 'square' having an area of 100 sq. ft. Thus, if 4 in. is to be exposed on a *plain tile*, 600 such tiles will form a 'square,' but in flat roofing, where the overlap is much less, only 210 tiles will be required for a square. 1 square weighs about 15 cwt. 100 plain tile laths 5 ft. long = 1 bundle.

For *pantiles*, 180 to a 10-in. gauge, 164 to an 11-in. gauge, and 150 to a 12-in. gauge. The lap of the tile over the one below it will be just the difference between its length and the length of the gauge. 1 square weighs about 8 cwt. 12 pantile laths 10 ft. long = 1 bundle. Fir tiling laths are 1¼ in. by 1 in.; and oak tiling laths, 1¼ in. by ½ in. 1 square of tiling requires 1 bundle of laths, 150 nails, 1 peck of tile-pins and 3 hods of mortar. (See p. 367.)

Wall and Floor Tiles are usually glazed; they are not very suitable for external use.

See also British Standard Specification No. 1281.

Roofing tiles are also made of concrete (see p. 350).

(iv) *Corrugated Iron Sheets.*—See pp. 236 and 301.

(v) *Zinc.*—Roofs are not covered with zinc, but with galvanised iron, *i.e.* zinc-coated iron.

The gauges of zinc recommended for roofing purposes are Nos. 13, 14, 15, and 16, the weights per sq. ft. of each gauge being respectively 16 ozs. 15 drs., 18 ozs. 12 drs., 21 ozs. 12 drs., and 24 ozs. 12 drs. The sheets usually measure 8 ft. × 3 ft. See also p. 86.

(vi) *Lead.*—Sheets of lead are used for some flat roofs (see p. 132).

(vii) *Wood.*—Wooden roofs are preferably covered with felt or mastic which are easily renewed (see also Table on p. 301).

(viii) *Felt.*—Usually impregnated with tar, asphalt, bitumen, or other impervious material (see Mastic). The felt is laid over matchboard, sometimes with a prepared paper between. (See British Standard Specification No. 747.)

(ix) *Mastic.*—The mastic compositions generally used for roofing are a form of asphalt, *i.e.* mineral grains (grit) embedded in a plastic binder, such as bitumen. The material may be 'natural' or 'synthetic.'

The mastic may be applied direct to concrete roofs, but for wooden roofs it is applied to a layer of felt impregnated but not coated with bitumen. Expanded metal or wire netting is sometimes laid over the felt. A paper underlay is often advantageous.

Mastic asphalts tend to creep in warm weather so some form of 'key' to hold them in position is desirable.

See British Standard Specifications Nos. 988 and 1162.

Heat-Insulating Value of Roofing Materials.*

A National Physical Laboratory test of roofing by measuring the heat lost from a heated room leads to the remarkable conclusion that the rate of emission of heat by radiation from the covering surface has more effect on the inside temperature than the rate of conduction of heat through the material. This result is of importance in the roofing of large factories of

* W. M. Thornton, D.Sc., D.Eng., Armstrong College, Newcastle-on-Tyne; *Engineering*, October 19, 1917.

a semi-permanent nature, where the temperature depends more upon the covering than in the case of buildings having a closed air space under the roof.

Old galvanised iron allows more heat to pass than glass, a remarkable result. Slates are good, but in proportion heavy. Deal boarding covered with asphalted felt, though thick, is not so heavy as Welsh slate, and is one of the best insulators. In this case its poor conductivity is evidently not negligible, and in all thick roofing materials it is probably of importance. (See also British Standard Specifications Nos. 1304 and 1334.)

Heat Transmitted through Roofing Material Exposed to Strong Radiation.

Material.	Rise of Temperature in Testing Chamber Deg. C. per Minute.	British Thermal Units per 10 sq. ft. per Hour.	Thickness in Inches.	Weight in Pounds per Square Foot.
1. Bright galvanised iron sheet	0·268	111	0·04	1·6
2. Galvanised iron, blackened below	0·40	108	0·04	1·6
3. Galvanised iron, blackened above	0·93	385	0·04	1·6
4. Galvanised iron, blackened above and below	1·40	581	0·04	1·6
5. Galvanised corrugated iron after one month's exposure to the weather	0·75	310	0·033	1·28
6. Do. one year	1·02	422	0·033	1·28
7. (6) painted black above	1·13	472	0·033	1·28
8. Roofing glass, serrated	1·10	453	0·22	2·25
9. Welsh slate	0·81	337	0·17	2·9
10. Westmoreland slate	0·60	248	0·25	4·8
11. ¾-in. T.G. deal covered with asphalted felt	0·30	124	1·0	2·6
12. Corrugated fibrocement after one month in use	0·78	325	0·2	1·8
13. Do. after one year in use	0·80	334	0·2	1·8
14. Do. painted dead black	0·82	341	0·2	1·8
15. Do. 'aluminium-finished' outside	0·50	207	0·2	1·8
16. Do. laid on top of thin asphalted felt	0·51	211	0 25	2·0

A suggestion was made in a National Physical Laboratory Report that air spaces in roofing materials might be advantageous. A satisfactory means of doing this is to lay corrugated asbestos-cement plates on the thinnest asphalted felt or other light sheet material supported by the roof frames. The effect of this is shown by the table to be equal to that of an aluminium surface, and where heat insulation and lightness are of the first importance a strong and efficient roofing can be made in this way at a reasonable cost.

Load on Roofs.

Description of Covering.	Per Square of 100 Superficial Feet.	Minimum Slope.	Description of Covering.	Per Square of 100 Superficial Feet.
	Lbs.	Degrees.		Lbs.
Lead covering weighs	784	4	Timber framing for slated or tiled roofs	560 to 672
Zinc	168	4		
Corrugated iron	336	4		
Slates	840 to 1,008	25½ to 30	Additional load for pressure of wind	4,032
Tiles	896 to 1,680	26½ to 30		
Boarding ¾ inch thick	290	25	Ditto for Hurricanes, say	8,960
Boarding 1½ inch thick	580	25		

The angle of Gothic roofs should not exceed 60°.

Corrugated-Iron Roofing.

B. Wire Gauge.	Size of Sheets.	Weight per Square.	Square Feet per Ton.	B. Wire Gauge.	Size of Sheets.	Weight per Square.	Square Feet per Ton.
No.	Feet.	Lbs.		No.	Feet.	Lbs.	
16	6 × 2 to 8 × 3	350	800	22	6 × 2 to 7 × 2½	175	1,550
18	6 × 2 to 8 × 3	258	1,000	24	6 × 2 to 7 × 2½	136	1,880
20	6 × 2 to 8 × 3	204	1,250	26	6 × 2 to 7 × 2½	118	2,170

One-tenth of the weight to be added for lappage. Sheets should overlap about 6 inches, and be double riveted at joints. 3 lbs. of rivets required per square of roofing. Purlins should be 6 feet apart. Curved roofs may be made up to 20 feet span without framing; tie-rods 12 feet apart.

Angles of Roofs.

Proportion of Rise to Span.	Angle.	Slope.	Proportion of Rise to Span.	Angle.	Slope.
	° ′			° ′	
⅙	18 25	3 to 1	⅔	53 0	¾ to 1
¼	26 35	2 to 1	¾	56 20	⅔ to 1
⅓	33 42	1½ to 1	1	63 30	½ to 1
½	45 0	1 to 1			

Dimensions of Parts of Wooden Roofs.

Span.	Section.						
	Principal.	Tie Beam.	King Posts.	Queen Posts.	Small Queens.	Straining Beam.	Struts.
Feet.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
20	4 × 4	9 × 4	4 × 4	..	..	..	3 × 3
25	5 × 4	10 × 5	5 × 5	..	..	..	5 × 3
30	6 × 4	11 × 6	6 × 6	..	..	..	6 × 3
35	5 × 4	11 × 4	..	4 × 4	..	7 × 4	4 × 2
45	6 × 5	13 × 6	..	6 × 6	..	7 × 6	5 × 3
50	8 × 6	13 × 8	..	8 × 8	8 × 4	9 × 6	5 × 3
55	8 × 7	14 × 9	..	9 × 8	9 × 4	10 × 6	5½ × 3
60	8 × 8	15 × 10	..	10 × 8	10 × 4	11 × 6	6 × 3

Weight of Components of Roofs.

	Per Square Foot.
Lead covering, including laps, but not boarding or rolls	5½ lbs. to 8½ lbs.
Zinc ,, ,, 14 to 16 zinc gauge	1¼ ,, ,, 1¾ ,,
Corrugated iron, galvanised, 16 W.G.	3½ ,,
,, ,, ,, 18 W.G.	2½ ,,
,, ,, ,, 20 W.G.	2 ,,
Sheet iron, 16 W.G.	2½ ,,
,, ,, 20 W.G.	1½ ,,
Slating laid with a 3-inch lap, including nails but not battens or iron laths:	
Doubles, 13 inches × 6 inches, at 18 cwt. per 1,200	8½ ,,
Ladies, 16 inches × 8 inches, at 31·5 cwt. per 1,200	8¼ ,,
Countesses, 20 inches × 10 inches, at 50 cwt. per 1,200	8 ,,
Duchesses, 24 inches × 12 inches, at 77 cwt. per 1,200	8¼ ,,

Per Square Foot.

Tiles *plain (clay) $10\frac{1}{2}$ inches × $6\frac{1}{2}$ inches, hand-made :
laid to $3\frac{3}{4}$-inch gauge . . . . 13·22 to 17·38 lbs.
„ 4 „ „ . . . . 12·32 to 16·46 „
„ machine-made :
laid to $3\frac{3}{4}$-inch guage . . . . 11·76 to 16·24 „
„ 4 „ „ . . . . 10·96 to 16·12 „
„ *Marseilles (interlocking) . . . . 6·5 „
„ *pan- . . . . 8·5 „
„ *Poole . . . . 8·9 „
„ *Roman, single . . . . 5·5–6·5 „
„ * „ double . . . . 6·5 „
„ *Spanish, over . . . . 5·1 „
„ „ under . . . . 6·2 „
„ *Somerset (interlocking) . . . . 7·0 „

Slate battens, $3\frac{1}{2}$ inches × 1 inch :
For doubles . . . . 2 „
For countesses . . . . $1\frac{1}{4}$ „
Slate boarding, $\frac{3}{4}$ inch thick . . . . $2\frac{1}{2}$ „
„ „ 1 inch thick . . . . $3\frac{1}{4}$ „
„ „ $1\frac{1}{4}$ inch thick . . . . $4\frac{1}{4}$ „
Wrought-iron laths, angle irons :
For duchess slates . . . . 2 „
For countess slates . . . . $1\frac{3}{4}$ „
Cast-iron plates, $\frac{3}{8}$ inch thick . . . . 15 „
Thatch, including battens . . . . $6\frac{1}{2}$ „
Felt, asphalted . . . . 3 „
Rok . . . . $\frac{1}{3}$ lb. to $\frac{1}{2}$ lb.
Ruberoid . . . . $\frac{1}{2}$ lb. to $\frac{3}{4}$ lb.
Lath-and-plaster ceiling . . . . 8 lbs.
Ceiling joists . . . . 3 „
Common rafters . . . . 3 „
Collar beams . . . . 2 „

Framing for wooden roofs, including purlins and ridge boards, but exclusive of tie beams :
20 feet span, king post, rise one-fourth span . . . . 2 lbs. per foot super.
30 „ „ „ „ . . . . 2 „ „ „
40 „ queen post „ „ . . . . 2 „ „ „
50 „ „ „ „ . . . . 3 „ „ „
60 „ „ „ „ . . . . 4 „ „ „

Tie beams : 20 feet span, king post . . . . 11 lbs. per foot run of tie.
30 „ „ . . . . 20 „ „ „
40 „ queen post . . . . 18 „ „ „
50 „ „ . . . . 20 „ „ „
60 „ „ . . . . 30 „ „ „

Angles of Roofs for Different Coverings.

Kind of Covering.	Inclination to the Horizon.	Height of Roof in Parts of Span.	Kind of Covering.	Inclination to the Horizon.	Height of Roof in Parts of Span.
	° ′			° ′	
Copper . . .	3 50	$\frac{1}{30}$	Thin slabs of stone or flags . . .	29 41	$\frac{2}{7}$
Lead . . .	3 50	$\frac{1}{30}$	Pantiles . . .	24 0	$\frac{2}{9}$
Zinc	4 0	$\frac{1}{25}$	Thatch of straw, &c.	45 0	$\frac{1}{2}$
Slates, large . .	22 0	$\frac{1}{5}$	Plain tiles . .	45 0	$\frac{1}{2}$
„ ordinary .	26 33	$\frac{1}{4}$	Ruberoid . .	3 50	$\frac{1}{30}$
Asphalted felt . .	3 50	$\frac{1}{30}$			

* Figures published by the British Standards Institution (see B.S.S. No. 648—1935) as typical of average tiles. but not legally binding.

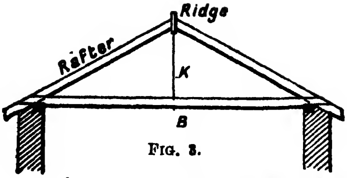


FIG. 3.

Single-span or Couple Roof.

May be used when walls can take the thrust without spreading; when they cannot, insert 4″ × 1″ binders as B (fig. 3), or ⅝′ iron ties, at about 10′ intervals, supported by a light iron king rod (K).

SCANTLINGS FOR COUPLE ROOFS.
Pitch up to 30°, with countess slates on 1″ boards.

Span.	Rafter.	Ridge Board.	*Ceiling Joist.	Span.	Rafter.	Ridge Board.	*Ceiling Joist.
Feet.	Inches.	Inches.	Inches.	Feet.	Inches.	Inches.	Inches.
8	3 × 2	7 × 1½	4 × 2	14	4½ × 2	7 × 1½	7 × 2
10	3½ × 2	7 × 1½	5 × 2	16	5 × 2	8 × 1½	8 × 2
12	4 × 2	7 × 1½	6 × 2	18	5½ × 2	8 × 1½	9 × 2

Collar-beam Roof.

Ceiling carried on under side of rafter and collar; collar best placed half-way up rafter; used for roofs up to 18′ span, under the same conditions as to wind, walls, and ties as given for couple roofs (fig. 4).

The rafters are nailed to ridge-piece and bird's-mouthed over wall-plate, 1′ apart in the clear, or from centre to centre on very exposed sites.

Ridge piece 7″ × 1½″.

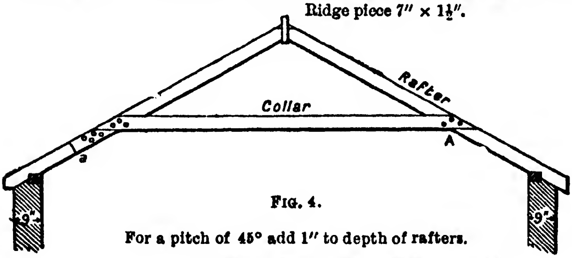


FIG. 4.

For a pitch of 45° add 1″ to depth of rafters.

SCANTLINGS FOR COLLAR-BEAM ROOFS.
Pitch up to 30°. If 45°, add 1″ to depth of rafters.

Span in Feet.	Rafters. For Countess Slates on ¾″ Boards, 1′ clear apart, or centre to centre if very exposed Site.				Collars.	
	† Thrust taken by Walls or Ties, Collars ½ way up.		‡ Tied by Collars ¼ way up to prevent Walls spreading.		From ¼ to ½ way up.	At any Height.
	No Ceiling.	Ceiled to Collars.	No Ceiling.	Ceiled to Collars.	No Ceiling.	Ceiled to Collars.
	Inches.	Inches.	Inches.	Inches.	Inches.	2″ wide × ½″ added to ½″ for every foot run of clear length of underside of collar.
8	1½ × 2½	1¾ × 3	2¼ × 3½	2¼ × 3¾	1¾ × 2½	
10	1¾ × 2½	1¾ × 3	2¼ × 4	2¼ × 4½	2 × 2½	
12	1¾ × 2¾	1¾ × 3¼	2¼ × 4½	2¼ × 5	2 × 2¾	
14	1¾ × 3	1¾ × 3½	2½ × 5	2½ × 5½	2 × 3	
16	2 × 3¼	2 × 3¾	2½ × 5½	2½ × 6	2 × 3½	
18	2 × 3¾	2 × 4¼	2½ × 6	2½ × 6½	2 × 4	

* When used they act as ties, B (fig. 3), but without K.
† Halve these collars on to rafters without cutting into latter.
‡ These rafters allow of the tension joint. If the collar is required halfway up, about one-fourth must be added to both breadth and depth of rafters, and ¼ in. to depth of collars. But with unstable walls, ties are far cheaper, and may be at long intervals if secured to wall-plates of sufficient width to take the thrust between the ties.

King-Post Roof.

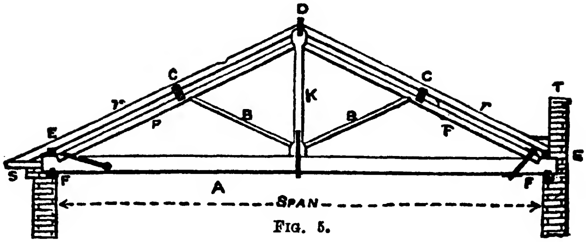


FIG. 5.

Used for spans from 20' to 30' (fig. 5).

A, tie-beam; B, braces or struts; C, purlins; D, ridge-board; E, pole-plate; F, wall-plate; K, king post; P, principal rafter; r, small or common rafter; S, eaves boarding; T, parapet.

SCANTLINGS FOR KING-POST ROOFS.

Span.	Tie Beam. A.	King Post. K.	Principal Rafters. P.	Braces. B.	Purlins. C.	Common Rafters. r.	Ridge. D.	Pole-plate. E.
Feet.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
20	9½ × 4	4 × 3	4 × 4	3½ × 2	8 × 4¾	3½ × 2	6 × 2	4 × 4
22	9½ × 5	5 × 3	5 × 3	3¾ × 2¼	8¼ × 5	3¾ × 2	6 × 2	4 × 4
24	10½ × 5	5 × 3½	5 × 3½	4 × 2½	8½ × 5	4 × 2	7 × 2	4 × 4
26	11½ × 5	5 × 4	5 × 4¼	4¼ × 2½	8¾ × 5	4¼ × 2	8 × 2	4 × 4
28	11½ × 6	6 × 4	6 × 3½	4½ × 2¾	8¾ × 5¼	4½ × 2	8 × 2	4 × 4
30	12½ × 6	6 × 4½	6 × 4	4¾ × 3	9 × 5½	4¾ × 2	8 × 2	4 × 4

Trusses, 10′ apart; angle of roof, 26° 33′; covering, slate; wood, Baltic pine.

Wall-plates, F, 4″ × 4″ to 5″ × 4″; if the trusses are over openings the wall-plates should be increased.

Queen-Post Roof.

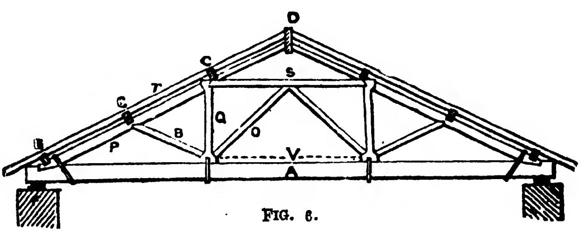


FIG. 6.

Used for spans over 30 and up to 46 feet (fig. 6). The braces, O, are generally omitted, and straight straining-sill placed at V.

SCANTLINGS FOR QUEEN-POST ROOFS.

Span.	Tie Beam. A.	Queen Posts. Q.	Principal Rafters. P.	Straining Beam. S.	Braces. B.	Purlins. C.	Common Rafters. r.	Ridge. D.	Pole-plate. E.
Feet.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
32	10 × 4½	4½ × 4	5 × 4½	6¾ × 4½	3¾ × 2¼	8 × 4¾	3½ × 2	6 × 2	4 × 4
34	10 × 5	5 × 3½	5 × 5	6¾ × 5	4 × 2½	8¼ × 5	3¾ × 2	6 × 2	4 × 4
36	10½ × 5	5 × 4	5 × 5¾	7 × 5	4¼ × 2½	8½ × 5	4 × 2	6 × 2	4 × 4
38	10 × 6	6 × 3¾	6 × 6	7¼ × 6	4½ × 2½	8½ × 5	4 × 2	6 × 2	4 × 4
40	11 × 6	6 × 4	6 × 6	8 × 6	4½ × 2½	8¾ × 5	4¼ × 2	7 × 2	4 × 4
42	11½ × 6	6 × 4½	6¼ × 6	8¼ × 6	4½ × 2¾	8¾ × 5¼	4½ × 2	7 × 2	4 × 4
44	12 × 6	6 × 5	6½ × 6	8½ × 6	4½ × 3	9 × 5	4¾ × 2	8 × 2	4 × 4
46	12½ × 6	6 × 5½	7 × 6	9 × 6	4¾ × 3	9 × 5½	5 × 2	8 × 2	4 × 4

Trusses, 10 apart ; pitch of roof. 26° 33′, or ¼ ; countess slates : timber, Baltic pine.
Wall-plates, F, 5″ × 4″ to 6″ × 4″ ; more if trusses are over openings.

Common Rafters.

12″ from centre to centre, or apart if not very exposed site.

Bearing in Feet.	Breadth in Inches.				
	1¾	2	2¼	2½	3
	Depth in Inches.	Depth in Inches.	Depth in Inches.	Depth in Inches.	Depth in Inches.
5	3	2⅞	2¾		
6	3¼	3½	3¼	3⅛	
8	4¾	4⅝	4⅜	4¼	4
10	6	5¾	5½	5⅜	5
12	7	6¾	6¼	6⅜	6
14	8¼	8	7¼	7½	7
16	9½	9¼	8¾	8½	8
18	10½	10¼	10	9½	9
20	11¾	11½	11	10½	10

Thin and deep rafters most economical; should not be thinner than 1¾″ to 2″, or may split when boards are nailed to them. For stone or tile covering should be one-third stronger.

Pressure of Wind and Weight of Snow on Roofs.

The *Wind Pressure* is usually assumed to be equal to 40 lbs. per foot in cases where the inclination of the roof is 60° ; for less angles than this the pressure is taken as follows :—

Inclination . . .	5°	10°	15°	20°	25°	30°	35°	40°	45°	50°	55°	60°
Pressure in lbs. per sq. ft.	5·1	9·6	14·2	18·4	22·6	26·5	30·1	33·3	36·0	38·0	39·4	40·0

The *Weight of Snow* in Great Britain varies from 5 to 11 lbs. per cubic foot. It is not likely to accumulate on roofs to a greater depth than 6 inches ; 5 lbs. per superficial foot of horizontal surface covered seems a quite sufficient allowance to make.

In Canada and the United States snow has been found in some instances to weigh as much as 30 lbs. per cubic foot some hours after it has fallen, and an allowance of at least 12 lbs. per superficial foot of horizontal surface covered should be allowed for it.

FLOORS

Wooden Floors.

Wooden floors are of three chief types, (i) single-jointed floors, (ii) double floors, and (iii) framed floors. They are described in architectural text-books. (See also British Standards Nos. 1018 and 1297.)

Jointless Floors.

Various bitumastic and other materials are used for *composition floors* or jointless floors, but the chief material used for this purpose is a magnesium oxychloride cement heavily loaded with a fibrous or other organic filler such as wood-fibre or sawdust. This can be finished with a denser, hard wearing surface, to produce a warm, comfortable, and durable floor. As the material is laid in a plastic condition it can be used in large uninterrupted areas and worked as a curved skirting to walls—a matter of some importance in hospitals and similar buildings. An excess of absorptive filler must not be used or the floor may 'sweat.' Magnesium oxychloride floors need protection by oiling or waxing to protect them against humidity and against decomposition by atmospheric carbon dioxide. They should be polished, but not washed or scrubbed (See p. 307).

Fire-resisting Floors.

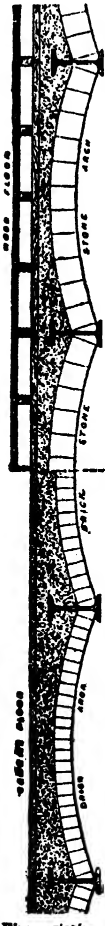


FIG. 7.

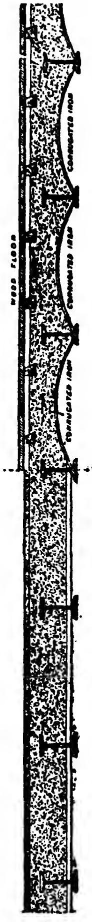


FIG. 8.

FIG. 9.

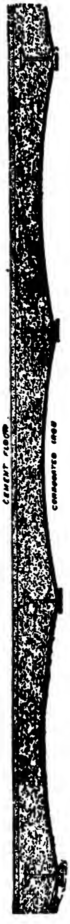


FIG. 10.

Fire-resisting floors are made of :—

(i) Reinforced concrete throughout.
(ii) Reinforced concrete between iron girders or rolled joists (figs. 8, 9).
(iii) Arches of corrugated iron or steel supported by rolled joists or girders (fig. 10).
(iv) Arches of brickwork or stone surmounted by concrete (fig. 7).
(v) Hollow blocks of burned clay (terra-cotta) between rolled joists, surmounted by just sufficient concrete to produce a smooth level surface without apparent joints. This type of

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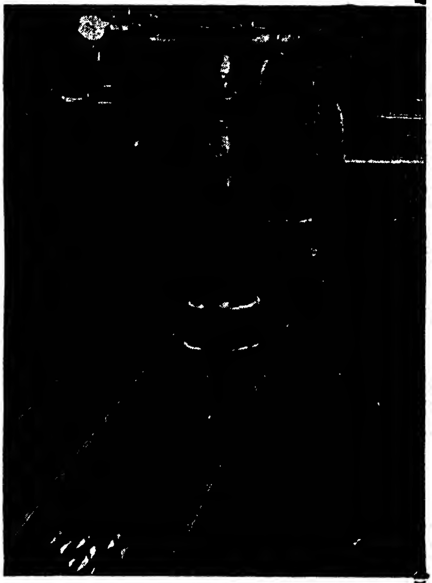

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floor is the most modern: it is the most resistant to heat and is relatively inexpensive. Many different shapes of hollow blocks are used.

See also 'Fire Resistant Construction,' ***Building Research Board Report No.*** **8 (1927), published by His Majesty's Stationery Office, York House, Kingsway, London, W.C. 2; price** 1*s*. 6*d*. net, and British Standard Specification No. 476.

The Ministry of Health's Model Byelaws, Series IV, Buildings, 1938, Clause 77, relating to floors is compulsory in some localities. Obtainable from H.M. Stationery Office, price 1*s*. 6*d*. net.

Factory Floors.

Floors used in some industrial buildings are particularly liable to corrosion in addition to ordinary wear and tear. Such floors should not undergo changes in volume or form shrinkage-cracks (produced in drying); for the sake of comfort, the floor should not be too hard, or too cold, slippery or noisy and should also look well and be easy to clean.

To resist very severe abrasion, paving brick, dense concrete, or heavy asphalt, reinforced with steel or iron grids are often used. Dust is highly objectionable in certain trades, for example, in painting shops and in food-manufacture. Concrete floors are liable to be dusty unless suitably treated. Wear is greatly reduced if factory trucks are fitted with rubber tyres with ball bearings and some means of steering; they also economise human effort, and are quieter in service.

In cheese factories, dairies, etc., where the destructive agents are mainly organic acids and fats, and abrasion is caused by the rolling of heavy churns, the best construction appears to be a hard, acid-resistant granitic asphalt-mastic, with cast-iron grids laid to form paths for churns; the provision of rubber mats at unloading points reduces wear, and prolongs the life of the churns.

In factories handling animal and vegetable fats or sugar, jams or fruit juices, quarry-tiles with a thin, neat Portland cement joint, or 'blue bricks' set in and jointed with aluminous cement mortar, is advisable. Timber is not likely to be attacked chemically, but will absorb fats, etc., and be liable to become very slippery and unhygienic.

In bacon, hide-tanning and other curing factories, the washing water contains salt which makes it necessary to protect a reinforced concrete sub-floor by an impermeable surface, such as asphalt.

The following British Standards may be helpful: Mastic asphalt Nos. 1076 and 1410. Magnesia compounds No 776. Pitch Mastic Nos. 1093 and 1177. Coloured Mastic No. 1375.

Sound-proof Floors.

The transmission of sound through floors can be minimised and in some instances prevented by providing a floating floor isolated from the walls, by the use of an isolating material of non-homogenous nature between the floor-covering and the floor proper, by massive and rigid construction of the floor itself, and by the use of a suspended ceiling. The suspended ceiling can be dispensed with in ordinary buildings if the floor-surface floats upon a resilient layer and has sufficient mass to absorb the energy of impact vibrations.

In converted flats and other locations where existing floors are to be made sound-proof a good result may usually be obtained by packing the space between the ceiling and the floor lightly yet fully with slag-wool. If the floor-boards are taken up in order to do this, strips of rubber or hair felt should be laid on the joists before refastening the boards and joists. Prior to using slag-wool, precautions should be taken to ensure that the ceiling will carry the additional weight which may be 80–90 lbs. per sq. yd.

See also 'Pugging' on p. 314.

Fire-resistant construction in floors normally provides adequate insulation against air-borne sound, but satisfactory insulation of the noise of footsteps and similar impact sounds can only be secured by the installation of a floor surface weighing some 20 lbs. per sq. ft. and floating upon a resilient medium. Normally this involves an increase of several shillings per square yard.

See also *Sound-proof Partitions* (p. 309) and *Sound Insulation* (p. 311).

Safe Loads on Floors.—**see p. 318 and 437.**

Partitions.

Partitions for internal use may be made of:—

(i) Deal boards—usually 2 ins. × 1 in. or 2 ins. square—placed about 2 ft. apart and covered with cheap canvas or calico, stretched tightly before nailing. The canvas may afterwards be covered with wall-paper applied in the usual manner. Such partitions are easily damaged, but are quite satisfactory for many purposes.

(ii) Slabs of highly porous concrete, sorel cement and sawdust, terra-cotta, plaster, or anhydrite, each slab being of a size convenient to handle, and fitted with a tongue and groove at opposite ends and sides so as to make a thin strong joint. (See British Standard No. 492.)

Lightness may be secured by mixing kieselguhr, diatomite, slag-wool, or sawdust with the clay of which terra-cotta slabs are made or with the plaster or sorel cement for other kinds of slabs. Alternatively, the plaster may be mixed with some substance which produces a foam or lather when wetted and so introduces a large volume of air-cells into the wet plaster.

Plaster-board is the most suitable form; *plaster-base* is a cheaper product which is satisfactory for rougher work. (See British Standard Specification No. 1230.)

The slabs may be simple (*i.e.* 'all in one piece') or they may be built up of two thin slabs of plaster, concrete, or terra-cotta with a different highly porous material, such as spun-glass.

(iii) Wooden panels, though these are usually expensive and are not sufficiently sound-proof.

(iv) Wooden laths nailed on to timber are cheap, but not sound-proof. The laths are usually covered with plaster in the same manner as ceilings. This kind of partition is stronger than (i) and is extensively used for bedrooms. In recent years, slab partitions (ii) have been much more popular as, though the slabs are more expensive than timber, the total cost of the partition is less and less skill is required in fixing.

(v) Multicellular glass has been reported upon favourably.

One of the lightest partitions on the market consists of very thin sheets of wood about an inch apart, with the intervening space filled with compressed wood fibre or wood-wool. It is extremely light and easy to fix, but very inflammable.

It is usually essential to avoid unnecessary weight in partitions, particularly when they are used in buildings not intended for them.

Where additional resistance to sound is required, two partitions about 2 ins. apart, with the intervening space packed with some light fibrous material, is usually satisfactory, but may form a home for mice.

When partitions are made of wood covered with canvas or other textile material they may be constructed as shown in figs. 11, 12 or 13.

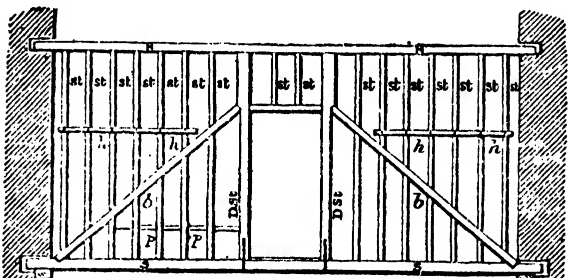


FIG. 11.

Partitions (fig. 11) should be framed so as to be supported only by the walls. *b* braces, H head, S sil, *st* studs, D*st* door-studs.

Dimensions of principal timbers, *b*, H, D*st*:—

4 ins.	× 3 ins.	for a bearing	not	exceeding	20 feet.
5 ,,	× 3½ ,,	,,	,,	,,	30 ,,
6 ,,	× 4 ,,	,,	,,	,,	40 ,,

The filling-in pieces, 'studs,' or 'quarterings,' need not be thicker than 2 ins., or just sufficient to nail the laths to. They are tennoned to the top and bottom plates, but butted and nailed on to the braces. If they exceed 3 ft. or 4 ft. in length they should be strengthened by short struts or horizontal pieces, as at *p*, *p*, called 'nogging pieces,' or, what is better, by a continued rail, *h h*, notched across the uprights and nailed to each. In a brick-nogged partition the nogging pieces are essential. They are much thinner than in a lath-and-plaster partition, being about ¾ in. thick and 2 ins. wide, placed at every fourth course of the brickwork in height.

The studs or quarterings for a lath-and-plaster partition should be spaced at from 12 ins. to 18 ins. from centre to centre; in a brick-nogged partition they should be 18 ins., 2 ft. 3 ins., or 3 ft. apart. If a partition is not well framed, or is made with unseasoned timber, the plastering will crack. Weight of a square (100 feet super) of partioning may be taken at 1,480 to 2,000 lbs.

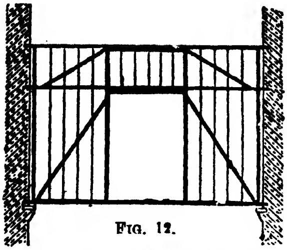


FIG. 12.

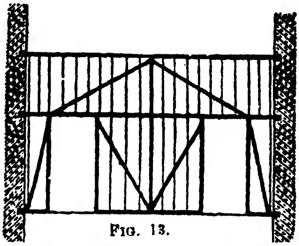


FIG. 13.

Figs. 12 and 13 show arrangements for partitions having doorways; in both cases the main portion of the partition is suspended from the truss forming the upper part of it.

Concrete Partition Slabs.

British Standard Specification No. 492—1933 requires that pre-cast solid concrete partition slabs shall be made of a specified aggregate (clinker, coke-breeze, pumice, or granulated blast-furnace slag) and Portland cement. The slabs must be matured for at least four weeks. The slabs shall be 17⅞ or 23¼ (±⅛) ins. long, 8⅞ or 11⅞ (±⅛) ins. high, and 2, 2½, 3, or 4 ($\pm\frac{3}{32}$) ins. thick, with bedding surfaces at right angles to the face. The edges forming the vertical sides shall be provided with tongues and grooves of specified radii. The slabs shall conform to the following:

For hollow concrete partition slabs, see British Standard Specification No. 728—1937.

Transverse test of six saturated slabs each supported on rods 15 ins. apart to correspond to a modulus of rupture of 100 lb. per sq. in.

Shrinkage on Drying not to exceed the following:

Type of Aggregate.	Linear shrinkage. Per cent.	Type of Aggregate.	Linear shrinkage. Per cent.
Clinker	0·08	Pumice	[illegible]
Coke-breeze	0·09	Granulated blast-furnace sla[illegible]	[illegible]

Moisture Movement or linear expansion after three days soaking in water [illegible]

Type of Aggregate.	Linear expansion. Per cent.	Type of Aggregate.	Linear expansion. Per cent.
Clinker	0·08	Pumice	[illegible]08
Coke-breeze	0·09	Granulated blast-furnac[illegible]	0·08

Loss of Ignition of slabs made with clinker and crushed to pass Stand[illegible] Sieve No. 72 not to exceed 28 per cent.

Density of the dried slabs not to exceed:

Type of Aggregate.	Average weight. Lbs. per cub. ft.	Type of Aggregate.	Average weight. Lbs. per cub. ft.
Clinker	90	Pumice	70
Coke-breeze	70	Granulated blast-furnace slag	90

A method for testing the suitability of clinker for use as aggregate is included in an Appendix to the Specification.

For Precast Concrete *Hollow* Partition slabs, see British Standard Specification No. 728—1937.

Sound-Proof Partitions.

The most effective sound-proofness of walls is obtained by a hard reflecting surface on the outside of the wall, by a non-homogeneous structure containing inert air-cells, by an air gap to prevent continuity, by a layer of insulating material, by a sound-absorbent surface facing the other room, and by massive and rigid construction: this is too costly for general use.

Other suitable sound-insulating materials are:

Rigid Acoustical Materials: boards of compressed straw and reeds, synthetic wood and other fibre boards, wood-wool boards made up with cement or plaster, and porous slabs of terra-cotta, stone or concrete.

Flexible Acoustical Materials: acoustic felts, eel grass blankets, slag-wool and vegetable fibre quiltings, glass silks, and sponge rubber.

Plastic Materials: sprayed asbestos, and acoustic plasters.

Tiling: porous tiling, perforated tiles with absorbent interiors, cork slabs, and fibre-board tiling.

Filling Materials: slag-wool and eel-grass, kapok, asbestos fibre, granulated cork, exfoliated mica, etc.

No partition is perfectly sound-proof, but sufficiently good results for most purposes may be obtained by using partitions of type (ii) on p. 307.

Partitions between rooms and flats and offices are commonly made of breeze two ins. thick and plastered both sides, but a minimum thickness of three ins. is required for adequate sound insulation, which thus entails a possible increase in cost. 'Party' walls between flats and offices are commonly made of two 2-in. thicknesses of breeze plastered both sides, but a more satisfactory sound insulation is obtained by the use of two 4½-in. brick partitions plastered on their outer faces or by some equivalent construction, but it is much more expensive.

Large cavities or plain spaces between a pair of walls do not produce good sound-proof partitions. The space should be filled with some porous or fibrous material so that a very large number of small separate air-cells is formed. It is important that as many as possible of the air-cells should be separate, as connecting cells are not nearly so sound-proof as are separate ones.

The smaller the cells the better, provided the 'walls' between them are sufficiently thin.

The larger the number of cells in a given volume (or weight) of partition the better will be the power of suppressing sound.

It is also important to use fibrous packing between a partition and the walls or floor with which it comes in contact.

'Solid to solid' fastenings destroy much of the value of an otherwise sound-proof partition. Hence, nails and screws should have rubber or other sound-insulating buffers.

Sound Insulation.

When a chamber is to be constructed so that no sound produced within it can be heard outside, or no externally produced sound can be heard in the chamber, it is seldom sufficient to make the walls, floor and ceiling merely sound-proof in the sense in which this word is applied to partitions. It is usually necessary to surround the chamber completely by a larger one with a space at least 4 ins. wide between the two and to make the walls, ceiling and floor of this external chamber also of sound-proof materials.

It is particularly important to avoid direct (hard) contact between solid materials, so fibrous or highly porous buffers must be used to prevent this. For some purposes rubber or asbestos cubes may be used as buffers, though they are neither porous nor highly fibrous. Metal-work should be avoided as much as possible as it is a good conductor of sound, and so is concrete unless it is sub-di[illegible]d with porous rubber, bituminous, or other sound-deadening materials between each sectio[illegible]

Vent[illegible] often a very difficult problem and much ingenuity is usually needed if adequate ventilat[illegible]e secured. Usually some form of forced draught, such as that created by an electri[illegible]tial.

A[illegible] principles just mentioned, there are no general rules for insulating buildings so tha[illegible] enter or leave them as this subject is still in an early stage of development.

See also S[illegible] *of Partitions* (p. 309), and *Sound-proof Floors* (p. 307).

Thermal Conductivity and Heat Transference.

The ready passage of heat through materials is sometimes of great importance, e.g. the plates of boilers, the tubes of recuperators and the sides of retorts and crucibles. The ability to transfer or conduct heat is a function of the material itself (metals being the best conductors and air and other gases the worst), of its density, and of its temperature. The denser the material the more readily will it transmit heat, whilst the more porous a material the lower is its conductivity.

Most inorganic building materials have a greater thermal conductivity when hot than when cold, but magnesian bricks are an exception. Carborundum has a much greater conductivity than fireclay, terra-cotta, silica or stone.

The following Table shows the thermal conductivity in gram-calories transmitted in one second through a plate of one square centimetre area and 1 cm. thick, the difference in temperature between the two faces being 1° C. (these figures can be converted into the British unit, viz. :— B.Th.U. per sec. per lin. cube for a difference of 1° F. by multiplying by 8·672).

Material.	Thermal Conductivity.
Stoneware	0·0045
Calais sand	0·0026
Carborundum	0·0145
Fused quartz	0·0039
Fireclay brick	0·0030
Silica brick	0·0020
Kieselguhr	0·0018
Slag-wool	0·0006
Calcined magnesia	0·0034
Porcelain	0·0043
Asbestos	0·0024
Graphite	0·0250

A valuable series of Reports on the transference of heat through walls of various kinds has been made by the Building Research Board, Watford, and published by H.M. Stationery Office. Kingsway, London, W.C. 2.
See also the Table on p. 300.
A valuable paper on Heat-conductivity by R. H. Heilman appeared in *Trans. Amer. Inst. Chem. Eng.*, 1935, 31, 165.
The *prevention* of heat-transference is effected by the use of *Heat Insulating Materials* (p. 375).

JOINTING BEAMS

Fishing.

To resist Tension.—For lengthening ties a plain fished joint is most economical for labour and material. The two ends of the beams are butted together, and secured on each side by means of bolts, as shown by fig. 14, which shows the fishing of a 6-in. × 4-in. beam with an iron plate on one side and a wooden plate on the other, the bolts being 1 in. in diameter with $1\frac{1}{2}$-in. heads, and the washers $2\frac{1}{2}$ ins. × $\frac{1}{4}$ in.

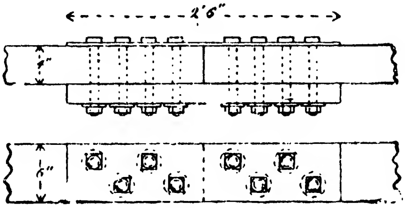


FIG. 14.

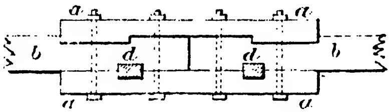


FIG. 15.

Another method is shown by fig. 15.
In this method, in order to relieve the strain on the bolts, the beam is indented or tabled as shown on upper side, or joggles or keys, *d*, *d*, are inserted. In either case the effective area is reduced by the indents or keys. The joint sectional area of the fish pieces, *a*, *a*, should equal that of the beam, *b*. The joint sectional area of bolts should be at least one-fifth that of timber minus bolt-holes. Bolts are better square than round. Large washers should be under bolt-heads and nuts; for oak the washers should be $2\frac{1}{2}$ times diameter of bolt, and for fir $3\frac{1}{2}$ times.

Scarfing.

To resist Compression (fig. 16).—The bearing surfaces, *e*, *e*, should always be perpendicular to compressing force. Fig. 17 is a more elaborate form.

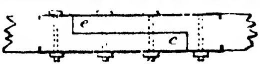


FIG. 16.

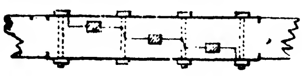


FIG. 17.

The lengths of scarfs should bear the following proportions to the depth :—

Wood.	Without Bolts.	With Bolts.	Bolts and Indents Combined.
Hard wood (oak, ash, elm)	6	3	2
Fir and straight-grained woods	12	5	4

The sum of the depths of the indents should equal 1½ depth of beam.

If assisted by fish-plates, it may be equal to the strength of a simple fished joint, which is as strong as the tie itself, less one rivet hole; and no advantage is gained by making the joint complicated, but it should be formed similar to that shown by fig. 16.

To resist Tension.—Fig. 18. is often used. It will hold without bolts or straps. At *c* a key or joggle of hard wood is inserted, wedged in so as to tighten the joint moderately; depth of key one-third depth of beam. This joint has only one-third the strength of solid timber tie; it may be increased by bolting an iron fish-plate on each side, as in fig. 19.

FIG. 18.

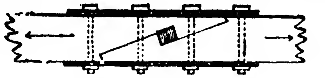


FIG. 19

To resist Compression and Tension.—Fig. 20 is suitable. The wedges at *c* are only required when bolts are to be added, in which case they are used to bring the parts together before the bolts are inserted, in order to prevent violent strains on the latter.

FIG. 20

To resist Cross Strains.—Fig. 21 is the best. The joint at *a*, being in compression, should be kept square; the lower, having to resist tension, may be oblique. There should be as great a thickness as possible at *b c*. To resist cross-strain, the scarf is strongest when made vertical, *i.e.*, the pressure in the figure being at right angles to the paper.

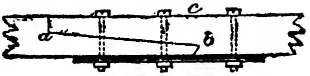


FIG. 21.

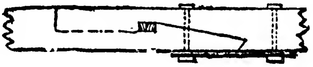


FIG. 22.

To resist Cross Strain and Tension.—The resistance to tension is given by indenting the beam and inserting wedges, as shown in fig. 22.

T Joint.

A very effective method of securing two pieces of wood, A and B, at right angles to each other, is to insert in a mortice hole in B a piece of aluminium *a* which has been drilled with a hole into which an ordinary wood screw *s* is forcibly screwed. The aluminium being comparatively soft enables the wood screw to readily work its way through the hole, whilst it forms a nut which enables the screw to be very tightly screwed up as shown in fig. 23.

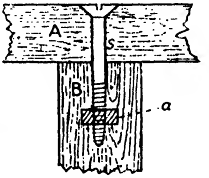


FIG. 23.

Building-up.

For building-up or joggling beams the method shown by fig. 24 may be adopted. This figure shows a portion of a 9 × 18-inch beam, 22 feet long, formed of two 9 × 9-inch beams secured

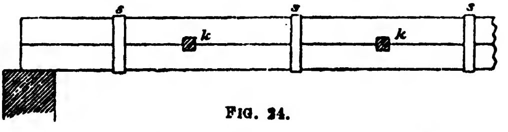


FIG. 24.

by wrought-iron straps, *s*, *s*, *s*, each 2 × ¼ inches; keys, *k*, each 2 × 2 inches, prevent the two pieces of which the beam is formed from sliding on one another.

Fig. 25 shows a similar beam with the two halves indented the one into the other so as to prevent sliding.

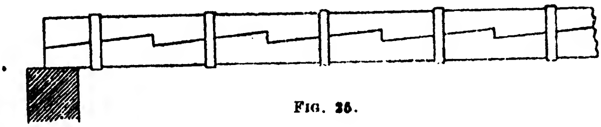


FIG. 25.

Mortise and Tenon.

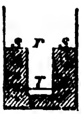


FIG. 26.

Mortising Upright into Horizontal.—The tenon, T (fig. 26), is usually one-third breadth of vertical. The pin-hole, if any, is generally placed at one-third length of tenon from shoulder, *s s*, and its diameter=one-fourth thickness of tenon. The mortise is made a little deeper, so that the shoulder may bear well; *r* is the *root*; C, C are the *abutment cheeks*.

Mortising Horizontal Beams.—The tenon is a projection from the end of the cross-beam, fitting closely into a mortise cut in the side of the main beam. It should be at the middle of its depth, so that the centre of the mortise may be at the neutral axis. Practically its lower edge is generally *on* the neutral axis.

Shouldered or Tusk Tenon.

To keep a cross-beam steady in its position a tenon requires length; to bear its share of load it requires depth; but a tenon long and deep would weaken the main beam too much. To avoid this a shouldered tenon is used. G (fig. 27) is the girder of a floor, B the binding joist, *a* the shoulder, *b* the tenon. The shoulder penetrates the girder about one-sixth depth of joist. Depth of tenon about one-sixth depth of joist; length about twice its depth, or it may be carried right through a thin girder and pinned, as in the case of a trimmer running into a trimming joist. Care should be taken that both tenon and shoulder bear exactly. Tredgold says the tenon should be one-third depth of joist above its lower edge. It should be, however, in the neutral axis.

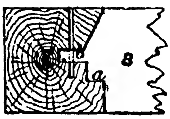


FIG. 27.

Post and Beam Joints.—To frame a horizontal beam into the side of post the 'shouldered tenon' is used, but the long tenon should be on edge or have its narrowest dimensions horizontal.

When the beam rests on the top of the post, a small tenon in centre of post may fit into mortise on the under side of the beam. Or the post may fit into a shallow rectangular notch in beam, the notch being divided into two parts by a *bridle* left uncut in the middle of the notch. about one-fifth breadth of beam.

Wedges.

When the end of the tenon is seen, it is kept tight by driving wedges of hard wood dipped in glue or white-lead between tenons and sides of mortise. When end of tenon is not seen, small wedges, called *fox wedges* (fig. 28), are inserted in saw cuts in end of tenon; when tenon is driven home, they enter and expand it. The mortise is dovetailed or made slightly wider at bottom This joint, when properly made, has the advantage that the post or style shrinks towards the shoulder and not away from it, and for this reason is sometimes employed for first-class work.

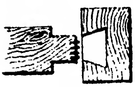


FIG. 28

Joint of Straining-Beam.

In oblique joints under compression the joint should be either perpendicular to one piece, as at *a* (fig. 29), or bisect the angle, as at *b*; the latter is the best in the case given in the figure. With a thicker strut at *b* it would be as shown by the dotted line.

Joints in carpentry should have white-lead applied with a brush over the surface forming the joint

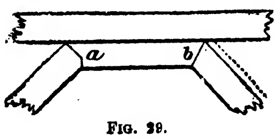


FIG. 29.

Strutting.

Joists laid with boards on top, and lath and plaster underneath, are not found to be sufficient to prevent bending and vibration under pressure. To remedy this, which is injurious to walls and to timber, strutting is placed in straight, continuous rows, at intervals not more than 4 feet apart, which reduces to a great extent the defect.

The following are three methods of strutting :—

Herring-bone Strutting.—This consists of pieces about 2 inches square fixed as shown in fig. 30.

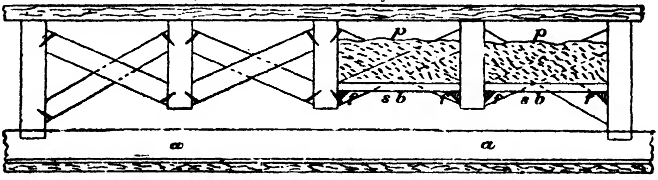


FIG. 30.

It is usual, instead of boring holes for nails, to make a saw cut at ends, and through this channel to drive the nail. By doing this any danger of splitting is avoided. This method has the recommendation that it acts where its leverage to prevent bending is greatest, viz., at the edges of joists, and is used largely in dwelling-houses.

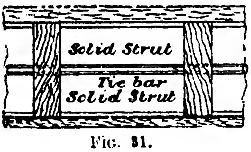


FIG. 31.

Solid Strutting.—1 or 1¼-inch boards, nearly the same depth as joists, are cut in tightly, and nailed between joists, as in fig. 31. This method is largely used for floors of factories and heavy buildings; and if cut so that the entire shoulder butts against the joists, is undoubtedly the firmest method; but unless care is exercised, the shoulders are apt to bite on the centre only. Sometimes long bolts, either of round bar or gaspipe, are passed through all the joists, and the strutting squeezed together with the joists and held tightly.

Key Strutting.—In this method struts are tenoned at their ends, and pass through mortises cut in the joists, and are secured with wooden keys. The method is expensive, weakens the joist, and altogether is a defective and unsatisfactory arrangement.

Pugging.

To prevent sound, sound-boarding, *s b* (fig. 30), is supported by fillets, *f*, *f*, cut diagonally out of 2-inch × 1¼-inch stuff nailed to bridging joists half-way between floor-boards and ceiling joist, *a*, *a*: over these pugging, *p*, *p*, of coarse plaster about 1½ inch thick is put. Dry moss or a mixture of lime mortar, earth, and smith's ashes, or chopped straw, is sometimes used instead of plaster. Strips of felt are sometimes placed between the joists and boarding to deaden sound.

See also p. 307.

Stairs.

Stairs (fig. 32) are described according to their form in plan, viz. :—

1. *Straight*—*a*.
2. *Dog-legged*—*b*; when the flights rise alternately in opposite directions with no interval between the flights.
3. *Open newel*—*c*; when the flights are ranged round an open well-hole, with a newel post at each angle.

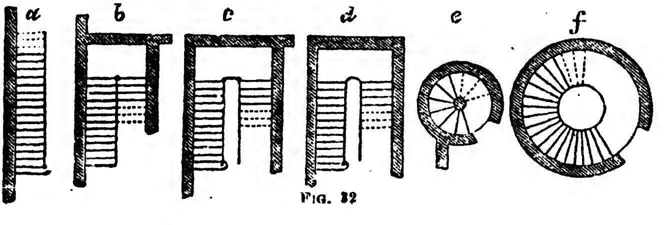


FIG. 32

4. *Geometrical—d*; similar to the last but with no newel posts: the handrail running uninterruptedly from bottom to top. The angles of the well-hole are generally rounded.

5. *Circular newel—e.*

6. *Circular geometrical—f*; when without a central newel.

Width—Stairs should be 3' 6" wide to allow two persons to pass easily.

***Tread and Rise.*—Tread should be from 9 ins. to 12 ins., rise from 6 ins. to 7½ ins. Some architects and builders increase the tread as the rise diminishes, but this is not satisfactory as there is no true mathematical relation between the tread and rise. The tread should be sufficiently large to accommodate average adult feet and the rise should not exceed 8 ins. for general use. Narrow treads and high rises are dangerous—particularly to elderly people and children—and should be avoided. If a very steep stairway with narrow treads is required it is better and much safer to make it in the form of a flat-runged ladder.**

***Safety treads* are increasingly used on public stairways. Their chief characteristic is a rough surface which will reduce slipping. This roughness may be obtained by using a cover of metal with a series of narrow grooves having sharp edges, but a still better method is to use a fine concrete containing a suitable proportion of small sharp particles of a very hard aggregate, such as carborundum; this grips the soles of the shoes sufficiently to prevent slipping, yet does not appreciably retard movement.**

The tread should be level throughout, strips fastened on to the head and projecting slightly above it are a cause of frequent accidents.

***Flights.*—It is dangerous to have a stair between two floors more than 9' apart in one continuous flight. It is desirable to limit height of one flight to about 7', and to avoid having two flights in one direction. *Winders* should be avoided if possible.**

Handrail.—Upper surface should be from 2' 7" to 2' 9" above the surface of the tread at the nosings, and on landings from 3' to 3' 3". Or, taking the height at landings as 3' subtract ½ the rise for height of rail from nosings of steps.

Materials.—In lofty dwellings it is desirable that stairs should be of stone or concrete, to be fireproof, at any rate in the lower storeys.

Stone Stairs.—Steps are supported by a wall at one or both sides. When at one side only they are called hanging steps, and should be built at least 9" into the wall. The bottom step is bedded on the floor, and the lower front edge of each other step rests on the upper back edge of the step below. The best form for the joint is shown in fig. 33, where the face *ab* is at right angles to the general inclination of the stair, and about 1" wide. Spandrel steps should have the sides which are built into the wall left square.

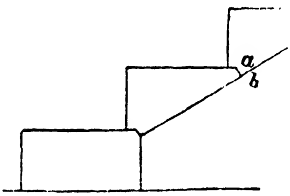


FIG. 33.

Repairing Stone Steps when Worn.—One plan is to fill up the hollows with Portland cement concrete made with fine sharp washed gravel about the size of a pea, 1 part gravel, 1 part cement. The worn part should be chiselled down, so as to give at least ¾" thickness of concrete in every part. The edge of concrete steps should always be rounded. Another method is to insert and cement slabs of artificial stone instead of concrete. **The surface of the whole tread must be flat when finished as any appreciable projection will probably cause accidents.**

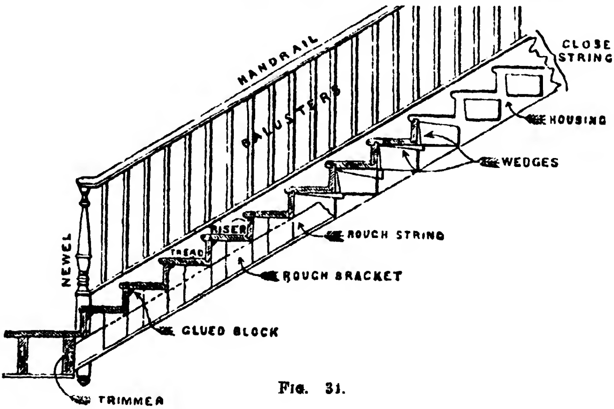


FIG. 34.

Wooden Stairs —Fig. 34.—Treads should be of hard wood, 1¼ in. thick, for steps 4 ins. long, increasing ⅛ in. for every additional 6 ins. of length.

The treads and risers are supported by a wall string and an outer string, the former 1¼ in. thick, the latter 2 ins. Treads and risers are housed into the wall string, and secured by wedges. The outer ends may be either housed into a close string, or may rest on and be mitred to the notches of an open string.

When stairs are more than 3 ft. wide, treads should be supported by rough strings or carriages fixed underneath and framed into the trimming joists of the landings. Scantlings of rough strings are calculated as bridging joists. Rough brackets are nailed to the side of the rough string to support the treads.

Joints between treads and risers are grooved and tongued, feathered or rebated. The last is the best joint.

BRICKWORK.

For *Regulations* concerning Brickwork, see Ministry of Health Byelaws, Series IV, Buildings (1938), Clauses 8–12 and 25–53 (H.M. Stationery Office, London, price 1*s*. 6*d*. net); see also p. 318.
For *Sizes of Bricks* see p. 365.

Brickwork is usually quoted in terms of a wall 1½ bricks or 14 ins. thick; such quotations are known as *reduced brickwork*. Thus, a wall of 320 sq. ft. area and 28 ins. thick would be quoted as 640 feet super of reduced brickwork. Such calculations only give roughly accurate results.

The thickness of walls is regulated by the size of the bricks, and walls are 4½ ins. or half-brick thick, 9 ins. or 1 brick thick, 14 ins. or 1½ bricks thick.

1 rod of brickwork = 272 superficial ft., 1½ bricks thick; = 11½ cubic yards = 296 cub. ft. = 235 ft. of bricks and 71 of mortar; = 4,500 bricks; = 15 tons average weight (see p. 317).

1 rod of brickwork requires about { 1 cubic yard of stone lime, 3½ cubic yards of sand; or 1½ yards of chalk lime, 3 yards of drift; or 1 cubic yard of cement, 3½ cubic yards of sand.

1 cubic yard of brickwork requires about { 6½ cubic feet of sand. 2½ " " lime.

1 builder's cart load of bricks = 500 bricks. 1,000 stock bricks, closely stacked = 55 ft. cube, 1,000 old bricks, cleaned and loosely stacked = 70 ft. cube. 1 ft. of reduced brickwork = 16 bricks. 1 ft. of superficial facing = 7 bricks. 1 ft. superficial of gauged arches = 10 bricks. 1 yard superficial in bricknogging, laid flat = 45 bricks and ¾ cubic foot of mortar. 1 yard superficial in bricknogging, laid on edge = 30 bricks and ½ cubic foot of mortar.

1 yard of paving = 36 stock bricks laid flat, = 52 bricks on edge. = 82 paving bricks on edge = 125 Dutch clinkers laid flat, = 140 Dutch clinkers on edge, = 136 Dutch clinkers laid herring-bone.

For crushing strength of bricks see p. 365, and for strength of brickwork see pp. 180, 365.

NUMBER OF PAVING BRICKS AND TILES.

(*Required per Square Yard.*)

The number of Paving Bricks or Flooring Tiles required per square yard (if all the joints are the same thickness) may be calculated from the formula *

$$N = \frac{1296}{L.W}$$

where N = number of bricks or tiles required per sq. yd.
L = length (in inches) of one brick or tile plus the thickness of one joint.
W = width (in inches) of one brick or tile plus the thickness of one joint.

When the tiles are square, L = W. It is often necessary to use some 'half tiles' to fit the desired space; this fact must be borne in mind when ordering.

ESTIMATING BRICKWORK.

A standard rod of brickwork is a wall 16½ ft. long, 16½ ft. wide and 1½ bricks (approx. 13½ in. thick.

The number of bricks in a rod of brickwork of any thickness depends on the size of the bricks and the thickness of the joints; it may be calculated from the formula †

---

* In this formula, no allowance is made for waste so that in practice a rather larger number of bricks or tiles should be ordered.

† No allowance is made for 'closers' or for waste, so that in practice a rather larger number of bricks should be ordered. An allowance of 3 per cent. for waste is usually ample but much depends on the thickness of the bricks, the conditions of transport and the manner in which they are handled.

$$N = \frac{39{,}204T}{H.L}$$

When N = Number of bricks.
L = Length (in inches) of one brick plus thickness of one joint.
H = Height (in inches) of one brick plus the thickness of one joint.
T = Thickness (in inches) of the wall divided by the width (in inches) of one brick.
In a standard rod, T = 3.

When standard bricks (British Standard Specification No. 657/1941) are used and all the joints are of equal thickness the number of bricks in a standard rod of brickwork is shown in the following table:

NUMBER OF BRICKS IN STANDARD ROD.

Height of brick (in.)	2	2	2	2	2⅝	2⅝	2⅝	2⅝	2⅞	2⅞	2⅞	2⅞
Thickness of joint (in.)	¼	⅜	½	⅝	¼	⅜	½	⅝	¼	⅜	½	⅝
Height of 4 courses (in.)	9	9½	10	10½	11½	12	12½	13	12½	13	13½	14
Number of bricks* (no waste)	5810	5439	5085	4780	4545	4300	4070	3860	4180	3965	3770	3585
Number of bricks (allowing 3 per cent. waste)[1]	5990	5600	5245	4930	4685	4430	4195	3980	4310	4090	3885	3695

[1] These figures are to the nearest five bricks.

Bricks absorb one-fifth of their weight of water, but the proportion varies greatly. Engineering bricks should be quite impervious to water.

A bricklayer's hod will hold 20 bricks, ⅝ cubic foot of mortar, or nearly ½ bushel.

In a day of 10 hours a bricklayer can lay about 1,500 to 1,600 bricks where the joints are left rough, about 1,000 when both faces are worked fair, or 500 when carefully pointed and faced with picked bricks.

For ordinary purposes, assume roughly that a bricklayer and labourer can, if they choose, turn out 2 cubic yards per diem, containing 800 bricks.

Bonds for Brickwork.

(Revised by E. A. W. Phillips, M.I.C.E., M C.I.)

ENGLISH THIN-WALL BOND (fig. 35) is the simplest for all ordinary work where the walls do not exceed three brick-lengths in thickness. The heading and stretching courses always alternate in this class of bond. No stretchers occur except those seen on faces of walls. No bricks in the same course should break joint with each other. All vertical joints in any course should be covered (by one-quarter brick laps at least) by the courses above and below, and this whether offsets occur or not, or other differences in the thicknesses of the walls.

FLEMISH BOND (fig. 36) shows headers and stretchers alternately in each course on the faces of walls. Is quite as strong as *English Thin-wall Bond*, since the quarter laps are in the same manner equally well maintained. *Flemish Bond* gives a better appearance for 9-inch walls, as a smoother face can be shown on both sides. In *double Flemish Bond*, both back and front of walls show Flemish Bond; in *single Flemish Bond*, one face shows Flemish, the other English bond, as above.

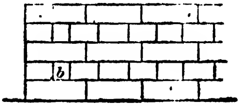


FIG. 35. English Bond.

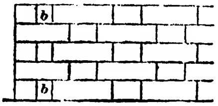


Fig. 36. Flemish Bond.

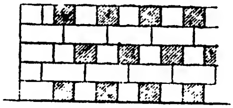


FIG. 37. English Thick-wall Bond.

ENGLISH THICK-WALL BOND (fig. 37). In this bond, every brick in a stretcher course should be a stretcher, and every brick in a header course should be a header. The bricks not only break joint, by half their lengths, with the bricks in the same description of course (i.e., stretcher or header) next above or below them, but also break joint, by half their lengths, in the same course. This is the strongest form of bond known, and should invariably be used in heavy walling, of 3½ bricks thickness and over. The quoin difficulty is best overcome by having a few specially moulded bricks, one brick-length in length, and ¾ of a brick-length in width, and of the usual thickness. The half-bricks on faces in fig. 37 are shaded.

* In this formula, no allowance is made for waste so that in practice a rather larger number of bricks or tiles should be ordered.

Garden-wall Bond is the *English Thick-wall Bond* adapted to thin one-brick walling, several courses of stretchers occurring to one course of headers. The same thing may be done with *chimneys*, where the walls are 9 ins. thick.

Diagonal Bond and **Herring Bone Bond** are only useful for paving. *English Thick-wall Bond* is stronger than either.

Header Bond consists entirely of headers, and is used for sharp curved or quick-sweep walls.

Stretcher Bond, entirely of stretchers, is used in ½-brick walls of all kinds.

Other bonds, Roman bond, and the Sussex bonds, are described in ' Indian Engineering. See also papers in the Minutes of Proceedings of Civil Engineers, vol. clxxi.

Closers, *b* (figs. 35, 36), are necessary in every alternate course to close the work on to any vertical line, or to start the bond from it; they run through the thickness of the wall.

Queen Closers, as *b*, are ½-headers formed by cutting a brick or a half-brick, called a *bat*, ongitudinally in half; ¾-stretchers, or ¾-*bats*, can be used as closers, but not so economically.

King Closers are bricks cut to a splay, so as to obtain as strong a bond as possible in closing on recessed openings, as for doors and windows.

Hoop-iron Bond, for walls, is generally formed of hoop iron, from 1 in. to 1½ in. wide and 18 B.W.G. ($\frac{1}{20}$ in.) to 16 B.W.G. ($\frac{1}{16}$ in.) thick; it should be clean bright and bedded in cement, for if allowed to rust it swells and injures the wall; it holds better if the edges are jagged. Useful in erecting works on bad foundations, but must not be relied on to give permanent strength unless laid as carefully as reinforcement.

LONDON BUILDING ACT (AMENDMENT) ACT, 1935.

(Abstract from the By-laws.)

For definitions of terms, see the ' London County Council By-laws for the Construction and Conversion of Buildings ' (obtainable from P. S. King & Son, Ltd., 14 Great Smith Street, Victoria Street, Westminster, S.W. 1).

Superimposed Loads.

The minimum superimposed load on each floor and on the roof shall be estimated as equivalent to the following dead loads :—

Class No.	Type of Building or Floor.	Slabs : lb. per sq. ft. of Floor Area.	Beams : lb. per sq. ft. of Floor Area.
1.	Rooms for residential purposes, corridors, stairs and landings in residences or flats	50	40
2.	Offices, floors above entrance floor .	80	50
3.	Offices, entrance floor and floors below, also garages for private cars of not more than 2½ tons net weight	80	80
4.	Corridors, stairs and landings other than Class I	100 or above.*	100 or above.*
5.	Workshops, factories and garages for cars other than those in Class 3	150 or above.*	120 or above.*
6.	Warehouses, book stores, stationery stores and the like.	200 or above.*	200 or above.*
7.	Any other purpose	*	*
	Roofs.	Slabs : lb. per sq. ft. of covered area.	Beams : lb. per sq. ft. of covered area.
8.	Flat roofs and roofs at an angle with the horizontal of not more than 20° .	50	30

On steeper roofs, a minimum superimposed load (including the wind load) of 15 lb. per sq. ft. of surface acting normal to the surface on the windward side and 10 lb. per sq. ft. of surface acting separately and not simultaneously outward on the leeward side. In estimating the vertical

* To be to the satisfaction of the L.C.C. surveyor.

superimposed roof load on all other parts of the construction a vertical superimposed load of 10 lb. per sq. ft. of covered area to be substituted.

In all cases of floors where the positions of partitions are not definitely located in the design, a uniformly distributed load to allow for them should be added to the dead floor load. For all floors of offices the minimum total allowance for internal partitions to be 20 lb. per sq. ft. of floor area.

Slabs and Beams.

Slabs and beams should be capable of carrying the following superimposed loads in any position on an otherwise unloaded floor :—

Class of Floor.	Minimum superimposed Load.	
	Slabs.	Beams.*
Floors of Class 1 (p. 318)	¼ ton uniformly distributed per foot width.	1 ton uniformly distributed.
All floors of Classes 2 to 6 (except garage floors of Class 5)	½ ton uniformly distributed per foot width.	2 tons uniformly distributed.
Garage floors of Class 5	1·5 × maximum possible combination of wheel loads, but each wheel load not less than 1 ton.	

Buildings of more than two storeys in height, in which loads and stresses are transmitted through each storey to the foundations, the superimposed loads of the roof and topmost storey shall be calculated in full in accordance with the table on p. 318, but for the lower storeys a reduction of the superimposed loads is permissible for all floors with a superimposed load of less than 100 lb. per sq. ft. namely :—

Next storey below topmost storey	10 per cent. reduction of its superimposed load
Next storey below	20 " " " "
Next storey below	30 " " " "
Next storey below	40 " " " "
Each succeeding storey below	50 " " " "

Where a superimposed load may move, proper provision to the satisfaction of the surveyor must be made.

Where floors are constructed for superimposed loads exceeding 100 lb. per sq. ft. a notice (in terms stated in the By-laws) must be shown in each storey having such a floor.

Wind Pressure.

All buildings must be able to resist a wind pressure in each horizontal direction of not less than 15 lb. per sq. ft. on the upper two-thirds of its surface up to the ridge of the roof and a further pressure in each horizontal direction of not less than 10 lb. per sq. ft. on all projections above the general roof level or ridge.

If the height of a building is less than twice the width of the base, the surveyor may not require the wind pressure to be calculated on the building as a whole.

Rules for the Walls of Buildings not Public and not of the Warehouse Class.

(*London Building Act (Amendment) Act*, 1935, *and By-laws* 39–62.)

According to this Act the external and party walls of dwelling-houses must be made throughout the different storeys of the thickness shown in the following Table, arranged according to the heights and lengths of the walls, and arranged for walls up to 120 feet in height, and supposed to be built of bricks not less than 8½ inches long or of stone or other blocks of hard and incombustible substance, the beds or courses being horizontal, the heights of the storeys being subject to the conditions given after the Table.

* Beams and ribs not more than 2½ ft. between centres shall be calculated for slab loads. Non-load bearing beams, such as those used solely as ties, are exempt.

TABLE I.

Height.	Length.	Thickness.
1. Up to 12 ft.	Unlimited.	8½ ins. throughout.
2. 12–25 ft.	Up to 30 ft.	8½ ins. throughout.
„	Over 30 ft.	13 ins. in lowest storey; 8½ ins. for remainder.
3. 25–30 ft.	Up to 20 ft.	8½ ins. throughout.
„	20–30 ft.	13 ins. for lowest storey; 8½ ins. for remainder.
„	Over 30 ft.	13 ins. for lowest two storeys; 8½ ins. for remainder.
4. 30–40 ft.	Up to 35 ft.	13½ ins. throughout except top storey which shall be 8½ ins
„	Over 35 ft.	17½ ins. for lowest storey, 8½ ins. in top storey and remainder 13½ ins.
5. 40–50 ft.	Up to 35 ft.	Ditto.
„	35–45 ft.	17½ ins. for lowest two storeys, remainder 13 ins.
„	Over 45 ft.	21½ ins. for lowest storey, 17½ ins. for the next storey and 13½ ins. for remainder.
6. 50–60 ft.	Up to 45 ft.	17½ ins. for lowest storey; 13½ ins. for remainder.
„	Over 45 ft.	21½ ins. for lowest storey; 17½ ins. for next two storeys, and 13 ins. for remainder.
7. 60–70 ft.	Up to 45 ft.	21½ ins. for lowest storey, 17½ ins. for next two storeys, and 13 ins. for remainder.
„	Over 45 ft.	Increased in thickness by 4½ ins. except in top two stories.*
8. 70–80 ft.	Up to 45 ft.	21½ ins. for lowest storey, 17½ ins. for next three storeys, and 13 ins. for remainder.
„	Over 45 ft.	Increased in thickness by 4½ ins. except in top two storeys.*
9. 80–90 ft.	Up to 45 ft.	26 ins. for lowest storey, 21½ ins. for next storey, 17½ ins. for next three storeys, and 13 ins. for remainder.
„	Over 45 ft.	Increased in thickness by 4½ ins. except in two top storeys.*
10. 90–100 ft.	Up to 45 ft.	26 ins. for lowest storey, 21 ins. for next two storeys, 17½ ins. for next three storeys and 13 ins. for remainder.
„	Over 45 ft.	Increased in thickness by 4½ ins. except in two top storeys.*
11. 100–120 ft.	Up to 45 ft.	30 ins. for lowest storey, 26 ins. for next two storeys, 21½ ins. for next two storeys, 17½ ins. for next three storeys, and 13 ins. for remainder.
„	Over 45 ft.	Increased in thickness by 4½ ins. except in top two storeys.*

NOTE.—If the thickness of the wall, calculated by the above Table, is less than one-sixteenth of the storey height, the thickness of the wall shall be increased to one-sixteenth of the storey height and the thickness of the wall below such storey height shall be increased to a like extent.

The Ministry of Health's Byelaws, Series IV, Buildings, 1938 (H.M. Stationery Office, London, price 1*s*. 6*d*. net) are compulsory in many localities.

Rules for the Walls of Buildings of the Warehouse Class.

(London Building Act (Amendment) Act, 1935, and By-laws 39–62.)

The external and party walls of buildings of the warehouse class must at the base be made of not less thickness than that shown in the following Table, arranged for walls up to 120 feet in height, and supposed to be built of bricks not less than 8½ inches, or of stone or other blocks of hard and incombustible substance, the beds or courses being horizontal.

In all walls exceeding 30 ft. in height the uppermost 16 ft. shall be less than 13 ins. thick, and the intermediate part between this 16 ft. and the base shall not be thinner than if the wall were to be built solid throughout the space between straight lines drawn on each side of the wall and joining the thickness at the base to the thickness at 16 ft. below the top or, where hereinafter specified, not less than 4½ ins. in excess of such thickness.

* Subject to By-law 53 respecting distribution in piers.

TABLE II.

Height.	Length.	Thickness.
1. Up to 25 ft.	Unlimited.	13 ins. at base.
2. 25–30 ft.	Up to 45 ft.	13 ins. at base.
"	Over 45 ft.	17½ ins. at base.
3. 30–40 ft.	Up to 35 ft.	13 ins. at base.
"	35–45 ft.	17½ ins. at base.
"	Over 45 ft.	21½ ins. at base.
4. 40–50 ft.	Up to 30 ft.	17½ ins. at base.
"	30–45 ft.	21½ ins. at base.
"	Over 45 ft.	26 ins. at base.
5. 50–60 ft.	Up to 45 ft.	21½ ins. at base.
"	Over 45 ft.	26 ins. at base.
6. 60–70 ft.	Up to 45 ft.	21½ ins. at base.
"	Over 45 ft.	Increased in thickness from base up to 16 ft. from top by 4½ ins.*
7. 70–80 ft.	Up to 45 ft.	21½ ins. at base.
"	Over 45 ft.	Increased in thickness from base up to 16 ft. from top by 4½ ins.*
8. 80–90 ft.	Up to 45 ft.	26 ins. at base.
"	Over 45 ft.	Increased in thickness from base to 16 ft. from top by 4½ ins.*
9. 90–100 ft.	Up to 45 ft.	26 ins. at base.
"	Over 45 ft.	Incteased in thickness up to 16 ft. from top by 4½ ins.*
10. 100–120 ft.	Up to 45 ft.	31 ins. at base.
"	Over 45 ft.	Increased from base up to 16 ft. from top by 4½ ins.*

NOTE.—If in any storey the thickness of an external or party wall as calculated from the above Table is less than one-fourteenth of the storey height, the thickness of the wall shall be increased to that figure and the thickness of the wall below such storey shall be increased to a like extent.

Rules for Permissible Stresses in Walls and Piers (where thicknesses are not determined as shown on pp. 320, 321).

(*London Building Act (Amendment) Act*, 1935, *and By-laws* 59–61.)

If in a storey, part of a wall is borne by a pier or a pier is borne by part of a wall, such pier together with the part of the wall of the storey directly above or below it shall be deemed to be a pier extending throughout such storey. Where a pier and a wall are in structural combination horizontally and the pier projects from the wall, if such projection from one face of the wall does not exceed one quarter of the wall thickness or if the sum of two projections from two faces does not exceed one-third the thickness of the wall such combination shall be deemed to be a wall. If such projections exceed the amounts mentioned above, the combination shall be deemed to be a pier, whose thickness or width shall be measured from the face of the projection on one side of the wall to the face of the projection on the other side.

The *slenderness ratio* of a wall or pier is the ratio of the effective height to the horizontal dimension lying in the direction of the lateral support.

The *effective height* of a wall or pier is :

Nature of Wall or Pier.	*Effective Height.*
(*a*) Wall without lateral support at the top.	1½ times the actual height.
(*b*) Wall with lateral support at the top.	¾ of the actual height.
(*c*) Pier without lateral support at the top.	Twice the actual height.
(*d*) Pier with lateral support at the top.	The actual height.

Any storey-height of a wall or pier not having a slenderness ratio exceeding 6, built of bricks or blocks, the total compressive stress due to the vertical load, horizontal pressure and any other forces shall not exceed the maximum pressure shown in Table III in respect of the bricks or blocks or the proportions of the mortar, whichever is the weaker. If the storey-height is built of concrete Table IV shows the maximum permissible compressive stress.

* See footnote to Table I, p. 320.

Where the slenderness ratio is 12, the total compressive stress shall not exceed 40 per cent. of those shown in Tables III or IV.

Where the slenderness ratio is between 6 and 12 the maximum permissible pressure shall be proportionate.

TABLE III.

MAXIMUM PERMISSIBLE PRESSURES ON WALLS OF BRICKS OR BLOCKS.

Class of Bricks or Blocks.	Compressive Strength of Bricks or Blocks. Lb. per sq. in.	Proportion of Mixture of Mortar (in Volumes). Cement	Lime.	Sand.	Maximum permissible Pressure on Wall. Tons per sq. ft.
Special	Over 10,000	1	—	2	$\frac{1}{500}$ of compressive strength of bricks + 10 but not exceeding 40
First	10,000	1	—	2½	30
Second	7,500	1	—	2½	23
Third	5,000	1	—	3	16
Fourth	4,000	1	—	3	13½
Fifth	3,000	1	—	4	11
Fifth	3,000	1	1	6	10
Sixth	1,500	1	—	4	8
Sixth	1,500	1	1	6	7
Sixth	1,500	1	2	9	6
Sixth	1,500	1	3	12	5½
Sixth	1,500	1	4	15	5
Sixth	1,500	1	5	18	4½
Sixth	1,500	—	1	3	4

TABLE IV.

MAXIMUM PERMISSIBLE PRESSURES ON WALLS OF PLAIN CONCRETE.

Concrete.	Cubic feet Aggregate per 112 lb. of Cement. Fine Aggregate	Coarse Aggregate.	Minimum Compressive Strength of Concrete 28 days after mixing.	Maximum permissible Pressure.
I.	1¼	2½	2,925 lb./sq. in.	40 tons/sq. ft.
II.	1⅝	3¾	2,500	35 „
III.	2½	5	2,250 „	30 „
IV.	7½		1,480 „	20 „
V.	10		1,110 „	15 „
VI.	12½		740 „	10 „
VII.	15		370 „	5 „

No solid wall of bricks, blocks or plain concrete shall have a slenderness ratio exceeding 12, but special provisions are made for cavity walls and some partition walls.

Where a wall or pier supports a beam or column and the stresses resulting from such local loading are distributed through adjacent material not so stressed, the compressive stress in the wall subject to local loading may exceed that shown in Tables III or IV by not more than 20 per cent.

Rules for the Use of Structural Steel.

(*London Building Act* (*Amendment*) *Act*, 1935, *and By-laws* 63-91.)*

The structural steel used shall comply with British Standard Specifications No. 4—1932, 4A—1934, or 6—1934, as the case may be.

Loading supported, collected or transmitted by structural steel shall be distributed upon the earth by concrete in conformity with the By-laws.

* For fuller details the Act and By-Laws should be studied. They are obtainable from P. S. King & Son, 14 Great Smith Street, Victoria Street, Westminster, S.W. 1.

The following rules only apply to structural steel which transmits the loads and stresses through each storey to the foundations:

(a) A steel column or beam in an external wall or in a recess in a party-wall shall be completely encased in brickwork, terra-cotta, concrete, stone or other incombustible material laid in Portland cement at least 4 ins. thick except on the underside and the edges of flanges, plates and angles where not less than 2 ins. will suffice.

(b) Other steel columns or beams shall be completely encased in similar material laid in Portland cement and to the satisfaction of the district surveyor, the casing being at least 2 ins. thick except on the upper surface of the upper flange and on projecting cleats, projecting rivet-heads and the like where a thickness of 1 in. may be permitted. This rule does not apply to buildings of one storey less than 25 ft. high.

(c) Except in buildings of one storey in which sufficient gussets and rivets are provided to transmit the whole load to the foundations, the foot of any steel column not of solid round section shall, after riveting, be properly machined over the whole area of the foot and shall have affixed thereto either—

(i) A base plate in effective contact with the whole area of the machined foot—the gusset plates, angles, cleats and stiffeners (if any) in combination with the bearing area of the foot and the base plate being sufficient to distribute the load properly.

(ii) A slab or bloom base-plate, in effective contact with the whole area of the machined column end. When the load under the slab is uniformly distributed the minimum thickness (in inches) of a rectangular slab shall be :—

$$\sqrt{\frac{3w}{4f}\,\frac{(B-b)}{D}} \text{ or } \sqrt{\frac{3w}{4f}\,\frac{(D-d)}{B}}$$

whichever is the greater

In this formula

w is the total axial load in tons.
B is the length (in inches) of the slab, measured at right angles to the web of the column.
b is the width (in inches) of the pillar, measured at right angles to the web of the column.
D is the length (in inches) of the slab measured parallel to the web of the column.
d is the width (in inches) of the pillar measured parallel to the web of the column.
f is the working stress in the steel taken as 9 tons per sq. in.

When the load under the slab is not uniformly distributed or where the slab is not rectangular the specified limits of stress shall not be exceeded.

(d) Except in buildings of one storey, both the ends of each length of steel column (other than a column of solid round section) shall, after riveting up complete with all gussets and end angle cleats, be properly machined over the whole area of the ends. All joints shall be close-butted and all caps and joint seating plates shall be in effective contact with the whole area of the column end.

(e) The bearing stress in any steel packing or beam between the ends of a superimposed column and a column beneath shall not exceed the permissible stress in the superimposed column and the width across such steel shall not be less than the corresponding width of the superimposed column.

(f) All joints in steel columns shall be as near as practicable to floor level. Where bending actions produce tensile stresses, the joints in columns shall be properly spliced to resist such bending. Column joints in which the resultant stress of all loads and bending moments is wholly compressive shall be sufficiently spliced to retain the members accurately in place and the length of each splice plate on each side of the joint shall be at least equal to the maximum breadth of the column flange or at least 12 ins. (whichever is the greater).

(g) Solid round structural steel columns shall have properly machined shouldered ends and shall be provided with caps and bases with bearing surfaces properly machined after being shrunk or screwed on. When the load under the cap is uniformly distributed the minimum thickness (in inches) of a rectangular cap or base shall be

$$\sqrt{\frac{3W}{4f}\,\frac{D}{B-d}}$$

where W is the axial load in tons.
B is the length of the shorter side of cap or base (in inches).
D is the length of the longer side of cap or base (in inches).
d is the diameter of the reduced end of the column (in inches).
f is the working stress in the steel, taken as 9 tons per sq. in.

No cap or base shall be less than $1\cdot5\ (d + 3)$ ins. in length B or in diameter.

Where the load is not uniformly distributed or the cap or base is not rectangular, the specified limits of stress shall not be exceeded.

(h) *Welding* may only be used if carried out under conditions prescribed by the Council in that case.

(i) Bolts and nuts shall conform to British Standard Specification No. 28—1932 (for black bolts) or 190—1924 (for turned bolts) except as regards the length of the threaded portion. Bolts

shall be provided with washers so that the thread is clear of the hole and the shanks shall project at least one full thread beyond the nuts and shall be secured so as to prevent the nuts from becoming loose. Washers shall be tapered, where necessary, to give the heads and nuts a true bearing.

(j) The diameter of a rivet shall be taken as that of the hole. The distance of a rivet hole or bolt-hole to the edge of the member shall not be less than the diameter of the rivet or bolt. Rivets shall not be closer (centre to centre) than three times their diameter. Tacking rivets (*i.e.* rivets connecting flange plates and not subject to calculated stress) shall have a pitch not exceeding 24 times the thickness of the thinnest outside plate or 12 ins., whichever is the lesser. For other rivets the straight line pitch in riveted members shall not exceed—

For parts in tension, 16 + the thickness of the thinnest outside plate with a maximum of 8 ins. For parts in compression, 16 + the thickness of the thinnest outside plate with a maximum of 6 ins.

Where two rows of staggered rivets occur in one flange of a single angle, the straight line pitch in the direction of stress shall not exceed 1½ times the distances stipulated above for parts in tension and compression respectively.

(k) Where two or more flange plates are employed, the edge-distance from the centre line of the nearest rivets connecting them to the web construction shall not be greater than 12 times the thickness of the thinnest outside plate, but tacking rivets shall also be used where such edge-distances exceeds 9 times the thickness. Where a single flange plate is used, the corresponding edge-distance shall not exceed 9 times the thickness of the plate.

(l) A flange plate or plate web of steel less than ⅜ in. thick shall not be used in a column.

(m) Subject to special provisions, the calculated working stresses upon structural steel shall not exceed:

(a) For parts in tension.	*Tons per square inch.*
On the net section for axial stresses or extreme fibre stress of all beams	8
On the net section of shop rivets for axial stress . .	5
On the net section of field rivets for axial stress, provided the rivets are of the usual type in accordance with British Standard Specification No. 275—1927, hot driven and that the parts when riveted are in close contact before riveting	4
On the net section of bolts for axial stress, the bolts being not less than ¾ in. diameter corresponding to British Standard Specification No. 28—1932 if black bolts, or No. 190—1924 if turned bolts, except as regards the length of the threaded portion and provided the parts to be bolted together are in contact before tightening the bolts	5

(b) For compression flanges of beams.	*Tons per square inch.*
On the gross section for extreme fibre stress of beams embedded in concrete or otherwise laterally secured . .	8
On the gross section for extreme fibre stress of uncased beams where the unsupported length L is less than 20 times the width b of the compressive flange . .	8
On the gross section for extreme fibre stress of uncased beams where L is greater than 20 times b . .	$11 - \frac{0{\cdot}15L}{b}$

For beams solidly encased, b may be taken as the width of the compression flange of the beam plus the lesser side concrete cover beyond the edge of the flange on one side only with a maximum of 4 ins.

The ratio L/b shall not exceed 50.

(c) For parts in shear.	*Tons per square inch.*
On the gross section of webs	5
On shop rivets and turned bolts of driving fit . .	6
On field rivets	5
On black bolts	4

The strength of rivets and bolts in double shear may be taken as twice that for single shear. The webs of beams shall be sufficiently stiff not to buckle.

(d) For parts in bearing.	*Tons per square inch.*
On packings, seatings and the like	12
On shop rivets and turned bolts of driving fit . .	12
On field rivets	10
On black bolts	8

(n) The span of any filler floor beam encased in concrete should not exceed 32 times the depth from the bottom flange of the beam to the top of the concrete. The span of any other beam shall not exceed 24 times its depth unless the deflections of the beam is less than $\frac{1}{325}$th of the span.

(o) The permissible ratio of effective column length to least radius of gyration in a steel column shall not exceed—

(a) For columns and struts forming part of the main structure	150
(b) For subsidiary members in compression . . .	200

The working loads per sq. in. in the shafts of columns and other compression members of structural steel shall not exceed those specified in the following Table except as provided in (o) and (p).

WORKING LOADS ON PILLARS.

Ratio of effective pillar length to least radius of gyration. $= \frac{l}{r}$	Working loads in tons per square inch of gross section. $= F_1$	Ratio of effective pillar length to least radius of gyration. $= \frac{l}{r}$	Working loads in tons per square inch of gross section. $= F_1$
20	7·2	130	2·6
30	6·9	140	2·3
40	6·6	150	2·0
50	6·3	160	1·8
60	5·9	170	1·6
70	5·4	180	1·5
80	4·9	190	1·3
90	4·3	200	1·2
100	3·8	—	—
110	3·3	—	—
120	2·9	—	—

Intermediate values may be obtained by interpolation.

(p) The effective column length shall be :

Type of Column.		Effective Column Length.
Columns of one storey	Properly restrained at both ends in direction and position.	0·75 × actual column length.
	Properly restrained at both ends in position but not in direction.	Actual column length.
	Properly restrained at one end in position and direction and imperfectly restrained in position and direction at the other end.	A value intermediate between the actual column length and twice that length.
Columns consisting of two storeys.	Properly restrained at both ends in position and direction.	0·75 × distance from floor level to floor level.
	Properly restrained at both ends in position and imperfectly restrained in direction at one or both ends.	A value intermediate between 0·75 and 1·00 × the distance from floor (or roof) level to floor level.
	Properly restrained at one end in position and direction and imperfectly restrained in both position and direction at the other end.	A value intermediate between the distance from floor level to floor level and twice that distance.

The above values are for typical cases only.

(q) In the case of eccentric loading on a steel column, the bending moment about each principle axis shall be calculated and the resulting bending stresses added to the axial load per sq. in. The working load may then be increased above that specified in (m) up to a limit F_2 where

$$F_2 = f_c + 7\cdot5\left(1 - \frac{f_c}{F_1}\right)\left(1 - 0\cdot002\frac{l}{r}\right)$$

where F_1 is the working load per sq. in. specified in (*m*). f_c is the total load on the column (in tons) divided by the gross cross-sectional area of the column (in sq. in.) and $\frac{l}{r}$ is the ratio of effective column length to least radius of gyration.

(*r*) The working loads and stresses specified for beams and columns and all their connections may be increased by 33⅓ per cent. where such increase is due solely to stresses induced by wind pressure. This does not apply to the stresses calculated in grillage beams of structural steel wholly embedded in concrete nor to those mentioned in (*l*).

(*s*) Where a beam is connected to a continuing column, the bending moment in the column due to the eccentricity of the reaction from the beam may be regarded as divided between the column lengths above and below the level of the beam, proportionately to their stiffness.

(*t*) In a continuing column all bending moments due to eccentricities of loading at any one floor level may be considered as entirely dissipated at the levels of the floor beams immediately above and below, provided that, at these latter levels, the column is effectively restrained in the direction of the eccentricity.

(*u*) Where structural steel is used and the loads and stresses are not transmitted through each storey to the foundations wholly by a skeleton framework of steel nor partly by such a skeleton and by a party wall the standards of stability and protection of such steel shall be to the satisfaction of the district surveyor, but not inferior to those required in these rules.

Foundations.

Within the area controlled by the London County Council foundations must conform to the London Building Act (Amendment) Act, 1935, and the By-laws associated therewith (obtainable from P. S. King & Son, Ltd., 4 Great Smith Street, Victoria Street, Westminster, S.W. 1).

Brick walls are made wider at the base, in order to spread the weight of the building over a larger area and produce an equality of settlement.

Figs. 38 to 44 show sections of footings (the wide courses at the base of a wall) of walls in English bond from one to three bricks in thickness. Trenches are excavated for beds of concrete usually from 2 feet to 7 feet in depth, and 12 inches wider than base of footings (6 inches on each

FIG. 38. Elevation.

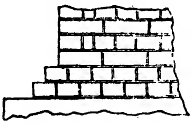


FIG. 39. Section.

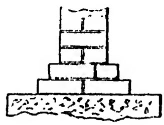
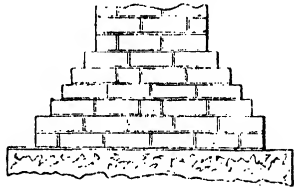


FIG. 40. Frontage for 2½-Brick Wall.



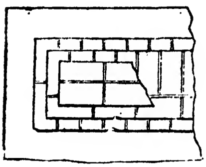


FIG. 41. Plan.

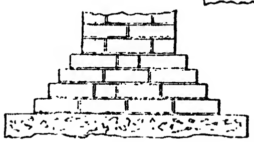


FIG. 42.

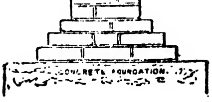


FIG. 43.

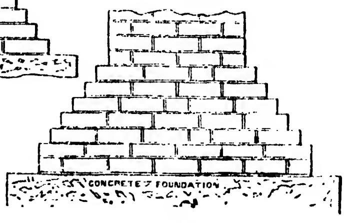


FIG. 44.

side). If the site on which a building is to be built be gravel, sand, or natural virgin soil, no concrete bedding need be used; otherwise the whole site should be covered with a layer of concrete, not less than 6 ins. thick, projecting at least 4 ins. on each side. See also pp. 435, 436.

MASONRY.

Lifting Stones.

There are various ways of laying hold of stones which are too heavy to be moved by hand. The following are those most frequently used :—

1. By a pair of nippers or iron claws or hooks, as in fig. 45, the ends of which catch in holes made, one in each side of the block, in the same vertical plane as, and at a convenient distance above, the centre of gravity of the block. 2. By a *Lewis*, or iron dovetail, as in fig. 46, made in three pieces, the two outer ones being first dropped into a corresponding dovetailed hole cut in the stone directly over its centre of gravity, then the centre-piece slipped in between them, the suspending-tackle put in its place, and finally a bolt passed through the whole and keyed up. 3. By two curved iron plugs, with a chain passing through eyes in their heads, placed in a dovetailed hole directly over the centre of gravity of the stone, as shown in fig. 47. 4. By an iron conical plug, with an eye at the top for a chain or hook to pass through, placed in a rather

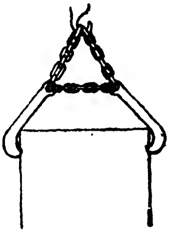


Fig. 45.

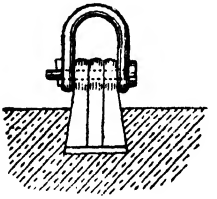


Fig. 46.

larger conical hole directly over the centre of gravity of the stone, with an iron wedge curved so as to fit the plug and the circumference of the hole, tightly driven in by its side, as shown in fig. 48. This plug was used for setting the heavy granite coping on the Thames Embankment

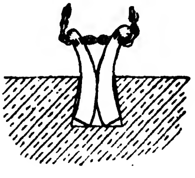


Fig. 47.

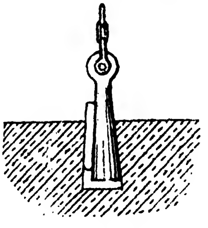


Fig. 48.

wall. 5. By a single iron plug, very slightly tapered, and driven into a cylindrical hole directly over the centre of gravity of the stone. By this simple means hard stones, such as granite, may be lifted with ease, a few side taps being sufficient to loosen the plug when required to be withdrawn.

Setting Stones.

All stones, except under peculiar circumstances, should be laid on their natural or quarry beds, or with their natural beds as far as possible perpendicular to the pressures they have to bear.

All dry and porous stones should be well wetted before being laid, so as not to absorb the moisture required for the proper setting of the mortar. Iron should never be placed in contact with stonework where, by rusting, in might disfigure it with stains, or split the stone, either by its increase in bulk during the process of oxidation, or by its expanding and contracting under the influence of heat and cold.

Descriptions of Masonry.

There are three descriptions of masonry in ordinary use, viz. :—

Rubble. Block-in-Course. Ashlar.

If the stone at disposal is thinly bedded, rough, or intractable, it should be used as *Rubble work*; if obtainable in blocks, and more or less easily wrought, it should be used as *Block-in-Course* or *Ashlar* according to circumstances.

The various systems are illustrated by figs. 49 to 58.

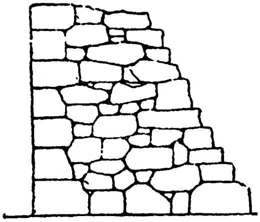


FIG. 49. Common, Rough, Uncoursed, or Random Rubble.

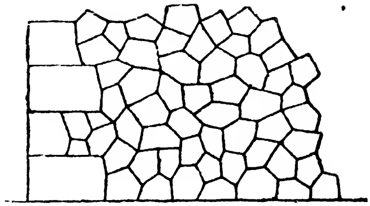


FIG. 50. Random Rubble, with hammer-dressed joints and no spalls on face.

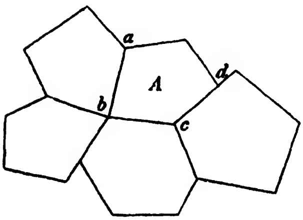


FIG. 51. Hammer-dressed Joints for Random Rubble.

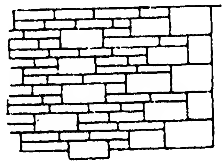


FIG. 52. Irregular-coursed, Random-coursed, Snecked, or Squared Rubble.

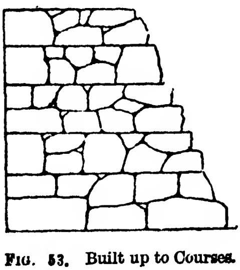


FIG. 53. Built up to Courses.

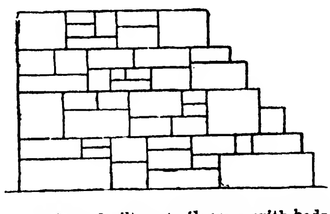


FIG. 54. Built up to Courses, with beds horizontal and joints vertical.

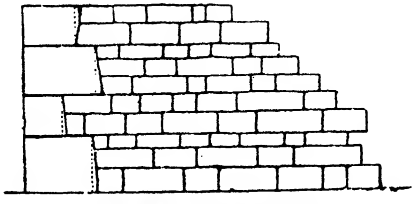


FIG. 55. Coursed Rubble.

RUBBLE MASONRY.

The best stones for rubble masonry are those that scabble freely, and such as lie in 4 or 5 in. beds. Basalts and stones of a crystalline structure are troublesome to use, as they fly under the hammer; but granite and sandstones work in well.

Rubble may be either *uncoursed*, *irregular*, or *random-coursed*, *worked up to courses*, or *coursed*, chiefly depending upon the character of the stone at disposal; some stones, such as Kentish rag, from their intractable nature, and the absence of any distinct lines of bedding, are specially adapted for *uncoursed* rubble, whilst others readily work into courses, and therefore should be used as *coursed* rubble.

In common, rough, uncoursed, or random rubble (fig. 49) the stones are laid at random, without being brought up to any level courses, the only tools required being a trowel, waller's hammer, and plumb or battering rules, as the case may require.

One bond stone should be used to at least every yard super of the face, and these should run about two-thirds to three-fourths through the wall, alternately from the opposite sides; this, in the case of dwelling-houses, diminishes the chances of damp being transmitted to the interior and is in all cases much preferable to a smaller number of *through* stones. In boundary and similar walls *through* stones may, of course, be used. In thick walls, such as piers, abutments, and retaining walls, the bond stones should tail into the work as far as possible; and the whole should be bonded together in the best way the sizes of the blocks will admit of.

It is necessary to see the bond stones actually built into the wall, for builders are apt to set large stones on edge, running only 3 or 4 ins. into the wall, instead of true bond stones—a trick which cannot well be detected after they have once been covered up. In random work (figs. 50 and 51) the stones cleave into irregular shapes, and are fitted closely together. Thus, in order to get A (fig. 51) into place the angles *abc*, *dcb*, are taken with the mason's bevel, and the joints are hammered and punched to fit truly. This is the most expensive kind of rubble work, and requires much more skill to build properly than any other class of rubble walling.

If the rubble stone is distinctly bedded, the beds varying much in depth, and often running very thin, the stones should be put together as in fig. in 52, what is called *irregular* or *random-coursed* rubble. The joints in this case are not necessarily vertical; but in a superior class of the same work all the joints are specified to be vertical and the beds horizontal, by which a more regular appearance is given to the work. The stones are all shaped and roughly squared as required, and hammer or axe faced, in the gross, ready for the waller to set in the wall. This class of masonry is very bold and effective for engineering purposes, when a good proportion of large stones are used, roughly hammer-dressed on face.

Both of the kinds of rubble masonry just described are very commonly worked up at intervals, with the line and level, to level courses, either regular or irregular in depth, the latter being preferable on account of its appearance and more suitable to the irregular sizes of the stones used. From 18 to 14 ins. are the ordinary depths of the courses, according to the nature of the stone. Figs. 53 and 54 show both common and irregular coursed rubble worked up to courses; the latter being also called *coursed header work* when the headers, occurring at intervals, run the full depth of the course. In fig. 55 a superior class of work is shown, the joints being all vertical, by which means a more regular appearance is obtained.

Rankine's 'Civil Engineering,' speaking of coursed rubble (such as fig. 55), says: '*One-fourth part* at least of the face in each course should consist of bond stones or headers, each header to be of the entire depth of the course, of a depth ranging from one and a half times to double that depth, and of a length extending into the building to from three to five times that depth, as in ashlar. These headers should be roughly squared with the hammer, and their beds hammer-dressed to approximate planes; and care should be taken not to place the headers of successive courses above each other, for that arrangement would cause a deficiency of bond in the intermediate parts of the course. Between the headers each course is to be built of smaller stones, of which there may be one, two, or more in the depth of the course. These are sometimes roughly squared, so as to have vertical side-joints; sometimes the stones are taken as they come, so that the side-joints are irregular; but no side-joint should form an angle with a bed-joint sharper than 60°. Care should be taken, not only that each stone shall rest on its natural bed, but that the sides parallel to that natural bed shall be the largest, so that the stone may be flat, and not be set on edge or on end. However small and irregular the stones may be, care should be taken to make the course break joint. Hollows between the larger stones should be carefully filled with smaller stones, completely embedded in mortar.'

The same authority gives the resistance of good coursed rubble masonry to crushing at about four-tenths of that of single blocks of the stone it is built with.

Coursed, or regular coursed rubble, as shown in fig. 55, is applicable where the beds, though thin, are pretty regular, so that a sufficient number of stones of a uniform depth can be got to allow of their being laid in regular courses of one stone only in depth. Larger stones, often of cut stone, equal in depth to two or three courses of the rubble work, are generally used at the quoins.

Whenever cut-stone is used in connection with rubble work, as at the quoins of a building, the ends adjoining or tailing into the rubble should never be cut square, but be left comparatively rough, so as to harmonise with the masonry in the body of the wall, and not appear as if cut off from it by straight lines.

Dry-walling is the simplest class of rubble work, and consists of stone roughly hammered, and bedded by pinning with spalls, without any mortar. It requires considerable skill to build properly, as its stability depends entirely on the firm bedding and bonding of the different stones. It is chiefly used for fencing land, for railway and canal embankments and cuttings, or at the backs of retaining-walls, to diminish the pressure of the earth on them, and to allow of water finding its way down to the drains below. Such walls—which are generally about 5 ft. high, increasing in thickness towards the bottom, and averaging about 18 ins. thick—are usually built to lines strained between trestles (fig. 56), in order to avoid plumbing the face.

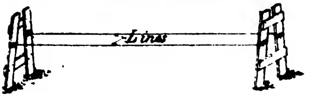


FIG. 56.

Dry-rubble walling is generally built in courses about 12 ins. high, and should have a waterproof top, or *coping*, to keep the water from getting into the body of the work and bursting it in frosty weather. The coping may be made of stones laid on edge in mortar, bituminous concrete, or, for want of anything better, clay puddle, or even sods.

When the stones come out in thin slabs the walls may be specified to be built of 'good sound, flat bedded stones of a fair average size (none less than 6 ins. broad across the narrowest part), set dry, and coped with stones, not less than 9 ins. deep, set on edge; each coping stone to extend across the full breadth of the wall, and to be properly set in good hydraulic mortar.' A high joint with a projecting ledge is very frequently used to mark the joints of rubble work: this is a bad system, and is sure to peel off under the influence of wet and frost.

A cubic yard of rubble masonry will, as a rule, require $\frac{1}{3}$ cubic yard of mortar and $1\frac{1}{4}$ cubic yard of stone.

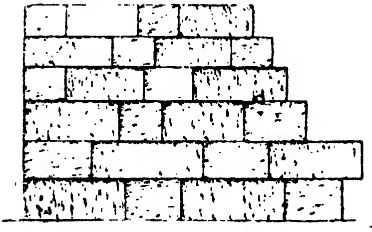


FIG. 57.—Block-in-Course.

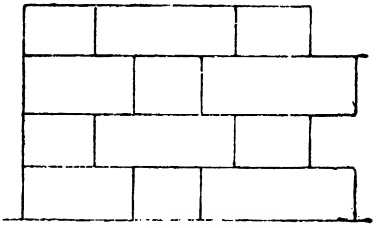


FIG. 58.—Ashlar.

BLOCK-IN-COURSE MASONRY.

Block-in-Course (fig. 57) is a term applied by engineers to a class of masonry much used in piers, abutments, wing walls, &c., when there is a suitable stone in the district and good solid work is required. The stones are all squared and brought to good fair joints, the faces being more or less worked, according to taste, but generally merely hammer-dressed; and the courses consist of single stones only in depth. It chiefly differs from ashlar in the smaller size of the stones and depth of the courses, the stones at command being—ordinarily rougher and more thinly bedded—chiefly *shoddies*, or stones under 12 ins. deep, whereas *ashlar* blocks range, as a rule, from 12 ins. to 14 ins. in depth.

The best stone for the purpose is a hard, self-bedded stone which will work easily into courses from 8 ins. to 10 ins. or 12 ins. in depth, the length of each stone being four or five times its depth. No attention is paid to uniformity in the depth of the courses.

ASHLAR MASONRY.

The term *ashlar* is applied to masonry built up of a thick-bedded stone which admits of being scabbled, or sawn into square blocks of given dimensions. It is also used to denote roughly squared blocks of stone over 12 ins. deep, as delivered from the quarry or stonecutter, or as prepared for setting. Ashlar work (fig. 58) is built in courses of uniform, or nearly uniform, depth—generally from 10 ins. to 14 ins.—ranging throughout with the quoins and dressings. It goes by different names according to the face put upon the stone, from quarry-pitched or rock ashlar up to wrought ashlar.

No stone should be laid in mortar without being first fitted into place and any irregularities in shape corrected. Large ashlars are first accurately fitted into place, and then set on a bed of mortar carefully spread out to receive them; after which the vertical joints have to be filled by stopping them up on the outsides with cement, and pouring in cement or mortar grout, which should be worked about with a piece of hoop iron, so as to ensure its completely filling the joints.

The beds and joints are worked to plane surfaces, so as to allow of close-fitting joints. These surfaces should not be too smooth, otherwise the mortar will not adhere so well to the stones, but in good work should be true enough to allow of joints not over $\frac{1}{8}$ in. thick; they are generally either left as they come from the saw (if sufficiently true), or taken out of winding by running chisel draughts round the margins, and dressing them down with the point, as in half-plain work, taking off any parts left projecting beyond the level of the chisel draughts with a chisel.

Stone Bridges and Arches.—See p. 466.

Piers.—See p. 472.

Stone Stairs.—See p. 315.

Safe Load on Stone Walls.—See p. 436. In many localities stone walls must comply with the Ministry of Health's Model Byelaws, Series IV, Buildings, 1938. See also the notes at the end of British Standard Specification No. 449 - 1937.

Strength of Stones.—See p. 180.

Artificial Stone.—See p. 358.

Concrete.—See p. 343.

Load-bearing Masonry.—See British Standard Specification No. 1145.

Proportions for Stones, Bond, &c.

In both block-in-course and ashlar masonry it is necessary to proportion the length, breadth, and thickness of the stones according to their hardness, in order to guard against their breaking across. Rankine says that in the weaker sandstones and granular limestones the length of a stone should not exceed three times its depth, nor the breadth one and a half to double its depth. In harder stones—such as require $2\frac{1}{2}$ or more tons to the inch to crush them—the length may be four or five times the depth, and the breadth three times the depth.

The best bond in masonry is that which shows on the face of the work alternate headers and stretchers in each course, as in Flemish bond in brickwork, each header coming over the centre of a stretcher in the course below. In such work one-third of the face consists of headers, if the length of the stretchers is twice the breadth of the headers; but as stones are rarely cut to exactly the same dimensions, it may be laid down that not less than one-fourth of the face of the wall should consist of headers, and that the stones should break joint from one to one and a half times the depth of a course.

The thickness of the joints will vary from $\frac{1}{8}$ to $\frac{1}{4}$ inch, according to the smoothness of, or amount of work bestowed upon, the beds, as it must be sufficient to transmit the pressures from stone to stone without permitting of actual contact at any point of their surfaces.

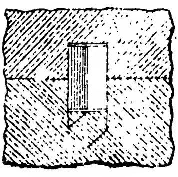


FIG. 59.

Dowelling.

For securing stones in places where the adhesion of the mortar or cement is not sufficient *dowelling* is the best method. Fig. 59 shows a method of doing this. It consists in making, exactly opposite each other, in the two stones two holes, either circular or square in section, and fitting into them a dowel or pin. These may preferably be of good hard slate, run with either brimstone or cement.

Measurement and Valuation of Masonry.

In districts where stone abounds, and forms the principal material for building purposes, ordinary walls are built of rubble stone, and measured by the cubic yard, or some other local standard, such as, in Ireland, by the running perch (21 feet), of a given height and thickness, or by the square perch of 21 feet super, at a standard thickness of 18 inches. In the West of England the square perch is employed, of 18 feet super, at a standard thickness of 2 feet, to which walls of any thickness are reduced; or the price per perch may vary with the thickness of the wall. In other parts masonry walling is measured by the rood of 36 square yards, or 24 cubic yards, the standard thickness being 2 feet; or by the rod of 272 feet super, as in brickwork, at a standard thickness of 18 inches or 2 feet.

In engineering, as distinct from architectural constructions, masonry is rarely valued otherwise than by the cubic yard for the roughest descriptions of work, and by the cubic foot for superior work.

The price in all cases varies with the quality and description of work, as 'coursed' or 'uncoursed' in 'walling' or 'foundations.'

The superficial content of the surface work is paid for separately by the square foot, yard, or rood of 36 square yards, including jointing or pointing, and such squaring to beds and joints as may be required, quoins, &c., of selected stones being often paid for at so much extra per foot run.

Superior work, such as squared masonry, whether in wallings or dressings, is usually valued by the foot cube as block stone, merely scabbled or sawn to the required dimensions and set. The work upon it is taken separately by the foot super.

When different descriptions of masonry occur in a wall, such as rubble faced with ashlar, instead of cubing out the rubble and ashlar work separately—in which case the latter is valued by the cubic foot prepared and set, including all beds and joints, the facework being taken by the foot superficial as tooled, pointed, &c.—the whole may be taken as rubble masonry, and so much extra allowed per foot superficial for the superior facing, including all extra labour, dressing, and pointing.

When walls are built of rubble masonry it is cheaper to use bricks for such parts as jambs and arches.

The block stone for use in a building is ordinarily sold by the cubic foot, or in large rough blocks by the ton, of from 13 to 17 cubic feet, 1 inch being allowed each way for irregularities and waste.

Permissible Pressures on Masonry.

The following are the pressures permitted in the Ministry of Health's Model Byelaws, Series IV, Buildings, 1938, for walls constructed of bricks or solid blocks and having a *slenderness ratio** not exceeding 6.

Column A shows the pressures allowed where the combined dead loading and superimposed loading can be assumed to be uniformly distributed over the area sustaining the load.

Column B shows the pressures allowed (subject to the average pressure not exceeding that in Column A) where the loading is a combination of the uniform pressure in Column A with increases in pressures due to eccentric loading and lateral forces.

Column C shows the pressures allowed at girder bearings, stanchion bases or other similar places where there are concentrated loads, taking into account the pressures in Columns A and B.

Crushing Strength of the individual Brick or Block in Lbs. per Sq. In.	Mortar proportioned by Volume not weaker than	Maximum permissible Pressures in Tons per Sq. Ft. of overall area of the Wall.		
		A.	B.	C.
Not less than 1,500	Lime mortar or black mortar.	4	6	6
	1 : 3 : 12 cement-lime mortar.	5·5	8·25	8·25
	1 : 1 : 6 cement-lime mortar.	7	10·5	10·5
	Hydraulic lime mortar.	5·5	8·25	8·25
	Cement mortar.	8	12	12
Not less than 3,000	Lime mortar or black mortar.	4	6	6
	1 : 3 : 12 cement-lime mortar.	7	10·5	10·5
	1 : 1 : 6 cement-lime mortar.	10	15	15
	Hydraulic lime mortar.	7	10·5	10·5
	Cement mortar.	11	16·5	16·5
Not less than 4,000	Lime mortar or black mortar.	4	6	6
	1 : 3 : 12 cement-lime mortar.	8	12	12
	1 : 1 : 6 cement-lime mortar.	10	15	15
	Hydraulic lime mortar.	8	12	12
	Cement mortar.	13·5	20·25	20·25
Not less than 5,000	Cement mortar.	16	24	24
Not less than 7,500	Cement mortar.	23	34·5	34·5
Not less than 10,000	Cement mortar.	30	40	45

For further information on Load-bearing Masonry, see British Standard Specification No. 1145.

* The *slenderness ratio* is found by dividing the height of a wall by its least overall thickness.

Weights of Various Stones.

DATA COLLECTED BY THE BRITISH STANDARDS INSTITUTION.

(See Standard Specification No. 648—1935.)

	Average Weight per cub. ft. Dry lb.	Average Weight per cub. ft. Wet* lb.
Limestones.		
Ancaster—Free Bed, Lincolnshire	141	150
Ancaster—Weather Bed Brown, Lincolnshire . .	150	160
Anston, Yorkshire . . .	150	156
Barnack, Northamptonshire . .	142	149
Box Ground, Wiltshire . .	121	132
Clipsham, Rutlandshire . .	140	150
Coombe Down, Somersetshire .	127	138
Corngrit, Somersetshire . .	122	137
Corsham Down, Wiltshire .	124	137
Doulting—Chelynch Beds, Somersetshire . . .	145	151
Farleigh Down, Wiltshire . .	125	140
Ham Hill, Somersetshire . .	131	136
Hartham Park, Wiltshire .	127	140
Headington Hard Bed, Oxfordshire	145	149
Headington Soft Bed, Oxfordshire	109	123
Hopton Wood, Derbyshire .	153	156
Huddlestone, Yorkshire . .	135	143
Ketton, Rutlandshire . .	129	139
Ketton—Pink, Rutlandshire .	153	164
Monks Park, Somersetshire .	126	138
Portland—Base, Dorsetshire Max.	146	
	135	146
Min.	124	
Portland—Whit, Dorsetshire Max.	149	
	143	145
Min.	126	
Sicilian Marble	169	170
Stoke Ground, Somersetshire .	118	132
Totternhoe, Bedfordshire . .	113	131
Weldon, Northamptonshire .	120	132
Westwood Ground, Somersetshire	122	135

	Average Weight per cub. ft. Dry lb.	Average Weight per cub. ft. Wet* lb.
Sandstones.		
Ackworth, Yorkshire . .	137	144
Auchinheath, Lanarkshire .	132	139
Binnie, West Lothian . .	134	141
Blaxter, Northumberland . .	128	136
Corncockle, Dumfriesshire .	131	138
Corsehill, Dumfriesshire . .	133	143
Craiglieth, Midlothian . .	139	144
Darley Dale, Derbyshire . .	143	149
Darney Northumberland . .	135	141
Glanton, Northumberland .	131	138
Greenlaw, Northumberland .	137	144
Hermand, Midlothian . .	144	149
Hollington—Mottled, Staffordshire	133	141
Hollington—Red, Staffordshire .	130	138
Hollington—White, Staffordshire	128	136
Leoch, Forfarshire . . .	153	156
Locharbriggs, Dumfriesshire .	127	136
Mansfield—Red, Nottinghamshire	147	153
Mansfield—White, Nottinghamshire	138	145
Polmaise, Stirlingshire . .	137	144
Woolton, Lancashire . .	134	140
Granites and other Igneous Rocks.		
Aberdeenshire Peterhead . .	162	163
Aberdeenshire Rubislaw . .	166	167
Cornish Grey	165	165
Guernsey	183	—
Norway Augite Syenite (Larvikite)	169	169
West of England Penrhyn .	163	164

QUANTITY OF STONE EQUIVALENT TO 1 TON IN WEIGHT.

Description.	Feet Cube.	Description.	Feet Cube.
Vein marble	13	Craigleith	14¾
Statuary marble and granite .	13½	Portland	15 to 16
Purbeck	14	Derby	15
Yorkshire	12 to 15	Bath	16 to 17
Description.	**Feet Super.**	**Description.**	**Feet Super.**
3½-inch York paving (tooled) .	58½	3-inch granite	54
3 „ „ „ „ .	49½	6 „ „	27
2½ „ Purbeck paving . .	68	7 „ curb	23
3 „ „ „ . .	56		

* The wet weight per cub. ft was calculated from the water absorption measured after 24 hours' complete immersion in water at the ordinary temperature.

Weight of Stone.

1 ton of granite = 13¼ cub. ft.

1 „ „ Mount Sorrel granite = 13¾ cub. ft.

1 „ „ Stoke Hall sandstone = 14 cub. ft.

1 ton of Huddersfield stone = 14 cub. ft.

1 „ „ Kentish Rag = 13½ cub. ft.

1 „ „ Blue Lias limestone = 14½ cub. ft.

Effect of Weather on Stone.

Stones vary greatly in their resistance to weather according to the conditions to which they are exposed. Thus, some limestones will last for several centuries 'in the country,' but are rapidly destroyed by corrosion in London and other large towns where sulphurous and sulphuric acids are present in the atmosphere.

Stones which are not laid on their natural bed will corrode more rapidly than when properly laid.

Where atmospheric conditions are trying, as in most large towns, special care should be taken to use only stones which can withstand such conditions. The fact that they have lasted for many years in some other locality is not a sound guide. Some very ancient buildings have corroded more seriously during the last fifty years than in the preceding 1,400 years!

Preservatives for Stonework are numerous, but few are really effective. The best appear to be some form of *sodium silicate* or of *silicon ester.*

Cleaning Stonework should never be effected with strong alkaline or acid solutions. Hot water and brushes are the only safe cleaning agents and the brushes should not have very hard bristles.

See also Descriptive Section IX, Part II.

SECTION X

MORTARS—CONCRETES—CEMENTS—GLUES—PLASTERS

(pp. 337–359.)

Revised and amplified by Alfred B. Searle
(Consulting Advisor on Cementitious Materials).

SECTION X

MORTARS—CONCRETES—CEMENTS—GLUES—PLASTERS

Revised and amplified by Alfred B. Searle
(Consulting Advisor on Cementitious Materials).

MORTARS.*

Mortars are pasty substances which gradually harden on exposure; they are used for uniting stones or bricks in the construction of masonry or brickwork.

The chief kinds of mortar are:

Lime mortar, composed of lime and sand.

Trass mortar, composed of lime, sand, and trass or brick-dust, or of lime and trass without sand.

Cement mortar, composed of Portland cement (or other siliceous cement) and sand.

Hydraulic mortar, composed of either hydraulic lime or cement, and sand, and characterised by its power of hardening under water.†

Clay mortar, consisting of a mixture of fireclay and crushed firebricks, and used chiefly in the construction of furnaces, kilns, and other structures which have to be heated to high temperatures.

Lime Mortar.*

The chief ingredients of ordinary lime mortar are lime and sand, but other materials, such as crushed bricks, ground stone or trass, and waste products such as ground slag, ashes, or coke, may, under suitable conditions, replace some of the sand. The tenacity of the mortar may be increased by the addition of hair or other fibrous material.

The best mortar for use in air † consists of a mixture of lime, sand, and brick-dust, but the latter is usually omitted on account of its cost.

Lime mortars made of fat or pure lime should not be used for roads, bridges, viaducts, waterworks, or any structure exposed to water, and some authorities maintain that even for churches, houses, and other buildings, it should not be used without the addition of trass or its equivalent, so as to prevent the flaking, crusts, and canker which are produced when too fat a lime is used.

Hydraulic lime mortar is composed of hydraulic lime and sand, the proportions varying from 3 to 6 measures for each measure of the lime (see *Lime Concrete*, p. 343). The chief use of this kind of mortar is for structures exposed to water, but unless the hydraulic lime is well selected, the structures may develop crusts and canker. Portland cement mortar is usually much superior to mortar made with hydraulic lime.

Trass Mortar.

Trass mortar is but little used in Great Britain, but has been used extensively abroad, and particularly in the East. The term 'trass mortar' is not confined to mortars containing true trass, but includes those containing similar materials, such as brick- or tile-dust, pozzolana, santorin earth, and other metamorphosed alumino-silicates, which combine with free lime to form an indefinite compound having cementitious properties. The use of trass or similar material overcomes the disadvantages which accompany the use of fat-lime. The best proportions for trass mortars are shown in the table on p. 341.

Trass was highly valued and greatly used by the Ancient Romans, but for several hundred years later its use was not understood and this valuable material fell into disuse. With the greatly extended use of Portland cement the importance of trass has again been realised, and during recent years the use of it has greatly increased in this country and abroad.

When wetted, Portland cement liberates free lime, combines with the latter and so is retained in the mortar. The resulting mortar or concrete is stronger than that in which no trass is used.

Trass is, in no sense, an adulterant but is, on the contrary, a simple means of increasing the strength of concrete and cement mortar.

Cement, mortars and concretes containing trass are more waterproof, more resistant to alkalis and to salt solutions than those in which no trass is used. The addition of trass is particularly important in concrete likely to be subjected to the action of sulphates in solution, e.g. concrete to be used for maritime structures.

Trass should not be added to aluminous cements, as to do so confers no benefit.

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* In many localities the Ministry of Health Model Byelaws, Series IV, Buildings, 1938, Clauses 10–12 are compulsory (H.M. Stationery Office, London).

† Mortars for use under water are termed *hydraulic* mortars.

When trass was difficult to obtain the Ancient Romans used ground roofing tiles as a substitute and, more recently, burned clay has been used as a synthetic trass.

The natural material, when carefully selected and properly prepared, is superior to the artificial product and appears to contain a greater proportion of active colloidal material.

The following mixtures (by volume) are now customary:

TRASS-CEMENT CONCRETE.

	Cement.	Trass.	Sand.	Aggregate.
Mass concrete	1	½–¾	3	6–10
Rammed concrete	1	1–5	3–4	5–10
Reinforced concrete	1	½	–	6
Poured concrete	1	½	–	9
Solid foundations	1	¾	6	12
Road (R.C.) foundations	1	½	3	6
Pier foundations	1	1	–	12–14
R.C. piles	6	1	7	8

TRASS-LIME AND TRASS-LIME-CEMENT CONCRETE.

	Trass.	Lime.	Cement.	Sand.	Gravel.
Concrete poured under water	2	3	–	3	–
Quay walls	1	1	–	1	4–5
Reservoirs, etc.	1	⅔	–	4	8
Rammed concrete for foundations	1½	1	1	5	7
Walls and vaults	1	⅓	⅔	4	8
Concrete	2	1	1	3	5
Watertight vaults	1	¼	½	3	6

TRASS-CEMENT, TRASS-LIME AND TRASS-LIME-CEMENT MORTARS.

	Trass.	Lime.	Cement.	Sand.
Masonry work	2	–	3	2
Masonry work	1	–	1	4
Rendering	1	–	1	3
Watertight rendering	1	–	1	2
Construction work above ground level	1	2	–	4–5
Masonry and brickwork	1½	1	–	2–3
Foundations and cellars	1	1	–	3
Masonry	2½	1½	½	6
Rendering	1½	½	2½	2
Watertight rendering	2	1	1	4–8

Cement Mortar.*

Cement mortar consists of a mixture of sand and Portland cement, the usual proportions being 2 to 6 measures of sand to 1 of cement, according to the strength required. The best proportions for various purposes are shown in the table on p. 341. For structures exposed to the air there should be more than twice as much sand as cement, but for completely waterproof structures the proportion of cement should not be less than half the sand used.

Lime is sometimes added to cement mortar to increase its plasticity and to make it easier to use. The addition of not more than 20 per cent. of hydrated lime slightly increases the

* In localities where the Ministry of Health's Model Byelaws, Series IV, Buildings, 1938 (H.M. Stationery Office, London; price 1*s*. 6*d*. net), have been adopted, all cement used in mortar must conform to British Standard Specification No. 12—1931 or No. 146—1932 or be a high alumina cement having the tensile strength, soundness and setting time mentioned in British Standard Specification No. 12—1931 or any other cement not inferior to the cement first mentioned above.

strength of the concrete, but if the latter is exposed to water (inch rain), the free lime may be washed out, leaving a porous material which soon flakes and disintegrates when exposed to frost. These drawbacks can be avoided by adding trass instead of lime—a cement mortar composed of two measures of cement, one of trass and six measures of sand, being of wide application and easy to work.

All cement mortars are hydraulic—*i.e.* they harden readily under water.

Raw Materials for Mortars.

The chief materials for mortars are *lime* and *sand*, but *Portland Cement* (p. 355) is sometimes used to replace part or all of the lime.

The *lime* should be made by burning a calcareous limestone low in magnesia; magnesian limestone, or dolomite, is much less suitable. The burning should be effected in such a manner that the lime contains at least 85 per cent. of calcium oxide in the form of quicklime. Limes which have been imperfectly burned, and so contain 15 per cent. or more of unaltered calcium carbonate, are not suitable for mortar to be used on important work. A very pure lime (containing 95 per cent. of free calcium oxide) is not necessary for building and is usually too costly. The idea that a pure lime is, in itself, objectionable, is incorrect, provided a suitable sand is used. If a very pure lime is used with a very coarse sand, it may result in a weak mortar.

Lean (*i.e.* less pure) lime * and hydraulic lime, on the contrary, harden in a manner similar to cement. The addition of trass or brick-dust to slaked lime has a similar effect on the manner in which the mortar hardens.

Hydraulic limes differ from the 'fat' or quick limes in consisting of an indefinite compound of lime and clay, which may be regarded as a crude Portland cement (p. 355), containing a large excess of free lime. The hydraulic limes owe their power of setting and hardening under water to this lime-clay compound, and consequently form a stronger brickwork or masonry than is produced by common mortar. Some hydraulic limes consist of a mixture of quicklime, true hydraulic lime, and inert or unburned material; for important work they should be tested and the proportions of these ingredients ascertained, as in hydraulic lime the proportion of cementitious matter present is more important than the percentage of lime.

Lias lime is hydraulic lime obtained from clayey limestones occurring in the Lias formations.

Quicklime should be slaked or hydrated before use; hydraulic lime should be bought in the form of a powder. To save the trouble and time involved in slaking and the risk of 'blowing' and other defects due to imperfectly slaked lime, it is becoming increasingly the custom to use *hydrated lime*. This has been slaked under skilled supervision, so as to produce a fine powder which is free from all coarse material and quicklime; it is sold in bags of 1 cwt. and is ready for immediate use. All limes used for building purposes should comply with British Standard Specification No. 890—1940† and in many localities they must comply with the Ministry of Health's Byelaws, 1938‖.

The *sand* used for mortars should consist of sharp (angular) grains of various sizes. Rounded grains do not interlock sufficiently to produce a strong mortar.‡‖

Crystalline quartz sand is to be preferred to mica or felspar sand; the usual description 'good, clean, sharp sand' is much too wide. The grains of sand which are angular give the best results of tensile strength, those that are round give the highest compression strength. It is advisable to submit the sand to a microscopic examination to ascertain its suitability, as a considerable variation in the strength of mortar may occur, owing to the form and variety of the sand particles. From trials made with mortar composed of 1 of cement and 3 of sand a difference of strength varying from 171 lbs. to 313 lbs. per sq. in. was found, owing entirely to the difference in the sand; normal sand mixed in above proportions gave 256 lbs. per sq. in. in 28 days.

It is very important that the sand used for mortars and concrete should be free from clay and decomposed vegetable matter (*humus*). A simple test, devised by D. Abrams and O. Harder, consists in shaking the sand vigorously with a dilute solution of caustic soda, allowing the sand to settle and noting the colour of the overlying liquid. If it is dark-coloured, the sand is probably unsuitable for use; a very pale yellow colour may be neglected.

The grading of the sand, *i.e.* the proportions of grains of different sizes, deserves far more attention than has yet been paid to it. By the use of a suitably graded sand, it is easy to reduce the proportion of the more expensive lime or cement by about 33 per cent. of that required for a less suitable sand. For most purposes the sand should contain about 30 per cent. of air-spaces or voids between the grains. The proportion of voids may be estimated by filling a measure with sand, and then pouring in water until the water is level with the top of the sand. The ratio of the volume of water, added to the volume of the sand, is that of the volume of the voids to that of the sand.

The *water* used in the preparation of mortar should be free from decaying substances; hence, peaty water should not be used.†‖

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* Limes which produce less than twice their volume of slaked lime in the form of an apparently dry powder are termed 'lean' or 'poor' limes.

† See p. 340.

‡ See *Sand and Gravel*, by A. B. Searle (Contractor's Record Ltd., London).

‖ In many localities the *sand* must comply with Clause 10 and the *water* with Clause 11 in the Ministry of Health's Byelaws, Series IV, Buildings, 1938 (H.M. Stationery Office, London).

BRITISH STANDARD SPECIFICATION FOR LIMES.*

(*Abstract of B.S. Specification No.* 890—1940.)

QUICKLIMES AND HYDRATED LIMES.

Classification—only two classes are recognised:

Class A.—Lime suitable for plastering finishing coat, coarse stuff and building mortar.

Class B.—Lime for coarse stuff and building mortar only.

Either class may be of the non-hydraulic or the semi-hydraulic type.
The purchaser should state:—

(a) Which class and type of lime is required.

(b) Whether a magnesian or calcium lime is required.

The vendor of lime containing more than 5 per cent. of magnesium oxide must always declare it as a magnesian lime.

QUICKLIMES.

Tests required—Class A. (Finishing.)

1. The *calcium and magnesium compounds* present, calculated as oxides, shall not be less than 70 per cent. of the ignited sample. The remainder shall consist essentially of silica and alumina soluble in a 5 per cent. solution of sodium carbonate.

2. The *loss on ignition* on lump quicklime not to exceed 5 per cent.; that on ground quicklime not to exceed 7 per cent.

3. The *carbon dioxide* not to exceed 7 per cent.

4. The *insoluble matter* (after treatment with sodium carbonate solution) not to exceed 3 per cent.

5. The *residue on slaking* left on a B.S. Sieve, No. 18, not to exceed 5 per cent.; the lime passing through that sieve not to leave a residue of more than 2 per cent. on a B.S. Sieve, No. 52.

6. The *volume yield of putty* (made in the prescribed manner) not to be less than 1·7 ml. per g. of quicklime for a calcium lime nor 1·4 ml. for a magnesian lime.

7. The *workability* shall be such that the putty when prepared and 'bumped' in a prescribed manner shall require not less than 13 bumps to cause a mass 11 cm. in diameter to spread to 19 cm. in diameter.

Tests required—Class B. (Coarse stuff and building mortar only.)

Nos. 1–4 same as for Class A.

5. The *residue on slaking* left on a B.S. Sieve, No. 18, not to exceed 5 per cent.

6. The *hydraulic strength* (of semi-hydraulic limes only) after 28 days damp storage to correspond to a modulus of rupture between 100 and 300 lb. per sq. in.

HYDRATED LIMES.

Condition.—The material must be in the form of a fine, dry powder.

Tests required—(identical for both Class A and Class B).

1. The *calcium and magnesium compounds* present, calculated as oxides, shall not be less than 70 per cent. of the ignited sample. The remainder shall consist essentially of soluble silica and alumina.

2. The *carbon dioxide* shall not exceed 5 per cent. by weight.

3. The *insoluble matter* (after treatment with sodium carbonate solution) shall not exceed 1 per cent.

4. The *workability* shall be such that the putty when prepared and 'bumped' in a prescribed manner shall require not less than 10 bumps to cause a mass 11 cm. in diameter to spread to 19 cm. in diameter.

5. *Fineness.*—The residue left on B.S. Sieve, No. 72, not to exceed 5 per cent. and the lime passing through this sieve shall not leave a residue of more than 10 per cent. on a B.S. Sieve, No. 170.

6. *Soundness.*—The expansion in a Le Chatelier mould not to exceed 10 mm. There shall be no signs of disintegration, pitting or popping.

7. The *Hydraulic strength* (of semi-hydraulic limes only) after 28 days' damp storage, to correspond to a modulus of rupture between 100 and 300 lb. per sq. in.

For details concerning modes of sampling and making prescribed tests, see B.S. Specification No. 890—1940.

* Published by the British Standards Institution, 28 Victoria Street, London, S.W. 1.

Proportions for Mortars.*

Mortars are usually made largely by guess-work, but to secure the best results—which are often the cheapest, as a large saving can be effected by preventing a waste of lime—it is necessary to determine the best proportion of sand for the particular lime used.

For many purposes, the most suitable proportions are from 2 to 5 measures of sand to 1 measure each of lime and brick-dust, or trass, but the latter is usually omitted, and large quantities of mortar consist of 1 measure of lime to 3 measures of sharp sand. When ashes † or coke dust are used, one measure of these materials usually replaces an equal amount of sand.

According to J. A. van der Kloes, who has investigated the matter very thoroughly, the best proportions for mortars for various purposes are as shown in the following table :—

Number.	Lean Lime.	Fat Lime.	Lime-Putty. Solid.	Lime-Putty. Fluid.	Trass.	Portland Cement.	Trass or Pozzolana Cement.	Sand.
A.—*Waterproof Mortars for Work which is continually under Water.*								
I.	1	—	—	—	1¼	—	—	1½
II.	—	1	—	—	1⅓	—	—	2
III. (*a*)	—	—	1	—	3	—	—	4
III. (*b*)	—	—	—	1	2¼	—	—	3
IV.	—	—	—	—	—	1	—	2
V.	—	—	—	—	1	1	—	2½
VI.	—	—	—	—	—	—	1	1–1½
B.—*Mortars for Quays, Sluice Walls, Damp-proof Courses, etc.*								
VII.	1	—	—	—	1¼	—	—	2–2½
VIII.	—	1	—	—	1⅓	—	—	2½–3
IX. (*a*)	—	—	1	—	3	—	—	5–6
IX. (*b*)	—	—	—	1	2¼	—	—	3¾–4½
X.	—	—	—	—	—	1	—	3
XI.	—	—	—	—	1	1	—	4
XII.	—	—	—	—	—	—	1	2
C.—*Mortars for Foundations and Buildings.*								
XIII.	1	—	—	—	1¼	—	—	3–4
XIV.	—	1	—	—	1⅓	—	—	4–5
XV. (*a*)	—	—	1	—	3	—	—	8–10
XV. (*b*)	—	—	—	1	2¼	—	—	6–7½
XVI.	—	—	—	—	—	1	—	3
XVII.	—	—	—	—	1	1	—	4–5
XVIII.	—	—	—	—	—	—	1	2½–3

For 1 cu. yd. of cement mortar, the following quantities of sand and Portland cement will be required :—

Proportions. Cement.	Proportions. Sand.	Sand. Cub. ft.	Cement. Lbs.
1	0	—	2,800
1	1	18·8	1,695
1	2	24·9	1,120
1	3	28·7	860
1	4	30·3	682
1	5	31·4	565
1	6	32·2	483

* See footnote on p. 338 and p. 343.

† The use of ashes in mortar is a frequent cause of scum or efflorescence in brickwork.

0·87 cu. ft. of neat cement mortar will result from mixing 0·36 cu. ft. of water with 1 cu. ft. of loose cement.

1 cu. ft. of loose Portland cement will make about—

4·1	**cu. ft. of concrete mixed**		**1 : 2 : 4**
5·0	**„**	**„**	**1 : 2½ : 5**
5·8	**„**	**„**	**1 : 3 : 6**
7·5	**„**	**„**	**1 : 4 : 8**

1 cu. ft. of loose Portland cement neat as cement mortar will cover about 10·4 sq. ft., 1 in. thick.
1 cu. ft. of loose Portland cement to 1 sand will cover about 17 sq. ft. 1 in. thick.
1 cu. ft. of loose Portland cement to 2 sand will cover about 25 sq. ft. 1 in. thick.
1 cu. ft. of loose Portland cement to 3 sand will cover about 34 sq. ft. 1 in. thick.
1 cu. ft. of loose Portland cement to 2 sand will lay about 146 bricks with ⅜-in. joint, and 247 bricks with ¼-in. joint.
1 cu. ft. of loose Portland cement to 3 sand will lay about 212 bricks with ⅜-in. joint, and 317 bricks with ¼-in. joint.
1 yd. super. of pointing brickwork in neat cement requires about 8 lbs. of cement.

The proportion of water required varies with the nature of the lime and sand used. (See the par. on *water*, on p. 339.)

Mixing Mortars.

Small quantities of mortar may be mixed by repeatedly turning over the materials with a shovel, and afterwards with a trowel, so as to mix them very thoroughly. Machine-made mortar is often more reliable than that mixed by hand. The most suitable machine for mixing mortar consists of an edge-runner or pan mill of moderately light construction, in which the materials are treated for 20 to 30 minutes. No pieces larger than ¼ in. diameter should be introduced into a mortar mill, and if any materials more than $\frac{1}{16}$ in. diameter are present, the mixing must be prolonged until they have all been reduced to $\frac{1}{32}$ in. or less in diameter.

Ready-made mortar is obtainable in some localities and has several advantages.

Choice of Mortar.

Mortar made with ordinary lime should not be exposed to water.

Mortar made with hydraulic lime should not be exposed to running water, unless the mortar also contains trass.

Cement-mortar is best where great strength is required a short time after completing the structure, but where a longer period of hardening is permissible, lime-trass mortar is cheaper and equally durable.

For the interior of kilns, furnaces, and flues, clay mortar or a refractory cement should be used.

See footnote on p. 338 and p. 343

Coloured Mortars.

Tints Given to Portland Cement Mortar containing Two Parts of River Sand to One Part of Cement.

Pigment.	Weight of Dry Pigment to 100 Pounds of Cement.			
	½ Lb.	1 Lb.	2 Lbs.	4 Lbs.
Lamp black	Light slate	Light gray	Blue gray	Dark blue slate
Prussian blue	Light green slate	Light blue slate	Blue slate	Bright blue slate
Ultramarine blue	—	Light blue slate	Blue slate	Bright blue slate
Yellow ochre	Light green	—	—	Light buff
Burnt umber	Light pinkish slate	Pinkish slate	Dull lavender pink	Chocolate
Venetian red	Slate, pink tinge	Bright pinkish slate	Light dull pink	Dull pink
Chattanooga iron ore	Light pinkish slate	Dull pink	Light terra cotta	Light brick red
Red iron ore	Pinkish slate	Dull pink	Terra cotta	Light brick red

(*L. G. Sabin.*

The pigments should conform to British Standard Specification No. 1014—1942.
Coloured cements can now be purchased. They are superior to amateur mixtures. See p. 348.

Specifications of Mortars.*

The Department of Scientific and Industrial Research has issued in *Building Research Board Report No.* 9—1927 (p. 340), the following tentative specifications for lime:

Quicklime for mortar—quicklime, Class A or B.

Hydrated lime for mortar—hydrated lime, Class A or B. Hydrated hydraulic lime is *not* suitable.

Hydraulic lime for mortar—hydraulic lime, C2 or C3.

See British Standard Specification for Building Limes (No. 890—1940) and pp. 339–340.

Mortar for structural work (*i.e.* exclusive of plaster, stucco, and parging) to consist of a plastic mixture of lime, putty, or hydrated lime, sand, and water, with or without pozzolanic material. The *lime* for work above ground-level should be Class C1, or preferably C2 or C3 (see table), and for below ground or water level, C2, C3, or a lime-cement mixture. Lime and hydrated lime of Class A should not be used without a pozzolanic material in such proportion that the lime, when mixed with three parts by weight of sand and the proposed proportion of pozzolanic material, shall have a minimum tensile strength of 50 lbs. per sq. in. The pozzolanic material may replace an equal weight of sand up to one-third of the weight of sand stated above. Mixtures containing less sand must have higher tensile strengths (see the *Report*). The *sand* shall not pass more than 10 per cent. of its weight through a Standard I.M.M. No. 80 sieve, and the proportions of particles of different sizes shall be well distributed. In lieu of sand, hard rock, slag, clean bricks, clean clinker, coal-cinder of approved origin or hard-burned clay, may be used if crushed so that all the material passes through a Standard I.M.M. No. 5 sieve. The *water* must be free from unusual proportions of dissolved salts and gases and must not be brackish.

In structures below ground or water level and subject to load, not more than one volume of sand shall be used to one volume of hydraulic lime. In structures above ground-level, subject to load, two volumes of sand may be used to one volume of hydraulic lime. In structures above ground not subject to load and in work of less importance, three volumes of sand to one volume of hydraulic lime may be used. Pozzolanic material may replace part of the lime by agreement with the architect.

CONCRETES.*

Concretes are very coarse mortars, containing stones, gravel or ballast, sand, and a binding agent such as lime or cement. They are known by various names, according to the binding agent employed.

Lime concrete usually consists of a mixture of 1 part lime, 4 parts gravel, and 2 parts sand.

The best way of making lime concrete is by mixing together unslaked or hot, fresh-ground lime with sand and ballast, in the required proportions, on a stone, brick, or wooden floor, turning it over twice to dry, and then, as it is shovelled to a third heap, adding from the rose of a watering-can sufficient water only to slake the lime and make the ingredients cling together in a pasty mass, turning it over well as the water is added. Generally speaking, it will take about ½ gallon to each cubic foot of ballast, but much will depend on the nature of the concrete and the dryness of the ballast employed.

Although frequently employed, fat or quick lime should not be used for lime concrete exposed to the weather, only hydraulic lime being suitable.

Beton is a French term, originally applied to lime concrete, but now used for any kind of concrete.

Cement concrete (more commonly known as *concrete* or *mass concrete*) is a mixture of stones, gravel or ballast, sand, and cement. Coke, clinker, and other materials are sometimes used to replace part of the stones, etc.

Breeze concrete consists of a mixture of 3 measures of small pieces of coke (preferably gas-works coke breeze), 1 of sand, and 1 of Portland cement. It is cheap, and nails can readily be driven into it. Its great disadvantage is that, in the event of a conflagration, it may take fire and increase the resulting damage instead of diminishing it.

Concrete made of ashes or clinker is sometimes, but erroneously, termed 'breeze concrete.'

Slag concrete is a mixture of crushed slag and Portland cement. It has been reported as having a greater compressive strength, less weight per cubic yard, and greater resistance to impact than concrete made with natural stone, but various failures which have occurred when slag concrete has been used have made many engineers unwilling to employ it. According to Dr. J. E. Stead, these failures were chiefly due to (1) using too little cement and (2) the use of fine slag contaminated with ashes and other deleterious substances. Blast-furnace slag for use in concrete should comply with British Standard Specification No. 877—1939.

Reinforced concrete or *ferro-concrete* consists of a concrete structure, the strength of which is increased by the immersion of steel rods or wires in the concrete (see Section XII, p. 381).

* Apart from the Byelaws mentioned in the footnote on p. 337 as a general guide, there is no official specification which applies to all mortars and concretes used in the British Isles, but in localities where the Ministry of Health's Byelaws, Series IV, Buildings, 1938, have been adopted the mortar must conform to them.

Foamed concrete is being increasingly used for sound-proof partitions and other light walls. Several processes are available for introducing air or hydrogen or carbon dioxide into the concrete paste and this imparts a spongy structure resembling pumice.

Refractory concrete is made of aluminous cement and crushed firebricks.

Raw Materials for Concrete.*

All concretes consist essentially of a coarse aggregate, a fine aggregate, and a binding material or cement.

The *coarse aggregate* should consist of suitable stone, gravel or ballast, coke breeze,† clinker, or other hard material of ample resistance to crushing, low porosity (absorbing less than 6 per cent. of its weight of water), and not liable to rapid disintegration by frost. The aggregate should be clean, and if necessary it should be washed before use. Except for the largest and roughest work, all the pieces of coarse aggregate should pass through a 2-in. ring.

The British Standard Specifications for aggregates for concrete are Nos. 877, 882, 1047 and 1165.

Gravel is probably used to a greater extent for concrete than any other one material. As it is a natural substance of very variable composition, even in the same deposit, care should be taken to ascertain its suitability before use. Some gravels contain too large a proportion of soft sandstone to be satisfactory and should not be used.

The best gravel is that with the lowest proportion of voids, *i.e.* that which contains stones of many different sizes in suitable proportion. The preliminary washing and screening of the gravel is usually done by the supplier, but the user should test the proportions of pieces of various sizes, the hardness of the softer pieces, and the percentage of voids, so as to ensure a suitable gravel being used.

The *fine aggregate* should consist of sharp, quartzose sand, similar to that used for mortar (p. 339). If trass is added, the concrete will be more resistant to water. Wet sand requires more cement than dry sand. The sand should be free from clay, and if necessary should be washed before use.

For some purposes, part of the fine aggregate may consist of *Trass* (p. 338) or other *pozzuolana* which combines chemically with any lime set free on wetting the cement and so forms an additional bond.

The *cement* may consist of hydraulic lime, Portland cement, or one of the cruder cements, such as *natural cement*, *Roman cement*, *Grappier cement*, *slag cement* (*iron Portland cement*), or various mixtures of Portland cement with sand, blast furnace slag (*Montan cement*), iron ore, etc. Of these, Portland cement is the strongest and most reliable. It should conform to the Standard Specification (p. 355).*

The *water* used for concrete should be the same as that for mortar (p. 339).

The *pigments* for concrete should conform to British Standard Specification No. 1014—1942.

Proportions for Concrete.*

It is being increasingly realised that the best concrete is that in which the coarse and fine aggregates are most completely immersed in a paste made of cement and water, no excessive amount of such paste being present. A concrete made on this basis is often much cheaper than those compounded on a 'voids' basis. For the newer method no definite proportions can be prescribed in advance; the best ones must be found by trial, as different batches of coarse and fine aggregate differ greatly.

To secure uniformity of consistency a *slump test* is generally used. In this, a conical tube 12 ins. high, with a lower diameter of 8 ins. and an upper one of 4 ins., is placed upright and filled with the concrete to be tested. The filling is effected in three stages, the layer formed in each stage being lightly tamped with a metal rod 21 ins. long and ⅝ in. in diameter. Immediately afterwards the tube is lifted off and the slump or settlement is measured.

According to Duff Abrams, a concrete with a slump of 1 in. or less will contain a little more water than is necessary for the maximum strength but will be too stiff for most constructional work. If 10 per cent. more water is present, the slump will be increased to 3–4 ins., 25 per cent. more water will increase the slump to 6–7 ins.

Instead of using the amount of subsidence or slump as a direct measure, some engineers prefer the term *consistency*. The following table shows the relation between the two :—

Slump in inches.	Consistency.	Slump in inches.	Consistency.
0–½	0·90	3–4	1·10
½–1	1·00	5–6	1·15
1–2	1·05	6–7	1·25
2–3	1·08	8–10	1·30

* In many localities only materials and proportions complying with the Ministry of Health's Byelaws, Series IV, Buildings, 1938, may be used. See also British Standard Specification No. 882—1940.

† On the danger of coke breeze, cinders, and other coal residues see p. 349.

Some engineers still prefer to base the proportions of the various materials on the voids of interstices between each; others make experimental mixtures with various proportions and determine the crushing strength of the concrete after 28 days.

As the determination of the voids is often troublesome, many engineers adopt definite proportions, such as 1 of Portland cement to 2 of sand to 4 of coarse material, or 1 : 2½ : 5, or even 1 : 4 : 8. In large masses of concrete and in foundations, large stones (commonly called 'plums') can often be placed in the mass some little distance apart so as to allow them to be surrounded by the concrete; these economise considerably, without greatly reducing the strength of the mass.

For concrete to be used under water, a mixture corresponding to a cement-trass mortar is preferable (see the table on p. 338).

The following table shows the relative weights required to produce the proportions mentioned in the first column: from it the cost of materials per cu. yd. of concrete for various mixturer can be ascertained.

MATERIALS FOR 1 CU. YD. OF CONCRETE.

Based on loose cement weighing 90 lbs. per cu. ft., with an average specific gravity of 3·12 and a cu. ft. of loose, moist, coarse sand weighing 89 lbs. when dried. (*Only approximate.*)

Proportions.	Kind of Coarse Material.	Lbs. Portland Cement in 1 cu. yd.	Sand cu. yds. in 1 cu. yd.	Coarse Material, cu. yds. in 1 cu. yd.
1 : 1½ : 3	Shingle (40 per cent. voids) .	666	·41	·82
Do.	Broken stone (45 per cent. voids)	697	·43	·86
1 : 1¾ : 3½	Shingle	610	·42	·84
Do.	Broken stone	640	·44	·88
1 : 2 : 4	Shingle	520	·43	·86
Do.	Broken stone	548	·45	·90
1 : 2½ : 5	Shingle	430	·44	·88
Do.	Broken stone	450	·46	·92
1 : 3 : 6	Shingle	364	·45	·90
Do.	Broken stone	383	·47	·94
1 : 4 : 8	Shingle	280	·46	·92
Do.	Broken stone	294	·48	·97

This table does not apply to concrete made with a single aggregate, such as Thames ballast, to which no sand is added, but where proper scientifically proportioned concrete is employed, as should always be the case, the values given in the table correspond with experience.

In measuring out the materials, the fact must not be overlooked that about 34 cu. ft. of stone, sand, and cement will be required for each cu. yd. of concrete, as the sand and cement occupy the interstices between the pieces of coarse aggregate.

Gravel passing the same screens as stone always has less voids than the stone.

Fine sand concrete has a smaller weight per cubic foot than coarse sand concrete.

Fine sand plus large aggregate (without cement) gives a *smaller* volume than coarse. Fine sand plus large aggregate (with cement) gives a *larger* volume than coarse.

Fine sand concrete is easier worked than coarse sand concrete for equal amounts of sand.

The finer the aggregate the more deleterious material and air it carries with it into the concrete.

The finer the sand the less should be used.

Mixing Concrete.

Concrete may be mixed by hand, as described on p. 343, but large quantities are preferably mixed by machinery. Various types of mixing machines have proved satisfactory, but the ones most largely used consist of a cylinder, fitted with baffle plates internally and mounted on trunnions so as to be revolved rapidly.

Combined Table of Working Capacities and Leading Dimensions of Concrete Mixers.

No. of Mixer.	00	0	1	2	3	4	5
Nominal batch capacity (cu. yds.)	0·08	¼	½	¾	1	1½	2
Hopper batch (cu. ft.)	3	6·75	13·5	20	30	56	80
Approximate output per batch of broken stone (cu. ft.)	2	5	11	16	24	38	54
Approximate output per batch of ballast or gravel (cu. ft.)	2·25	5·5	12	18	27	42	60
Approximate output per 10-hour day of broken stone (cu. yds.)	22	55	118	180	270	415	592
Approximate output per 10-hour day of ballast or gravel (cu. yds.)	25	60	133	200	300	470	660
Capacity output which has been obtained per 10-hour day (cu. yds.)	50	120	266	400	600	940	1,320
R.P.M. (driving shaft)	174	132	118	122	94	99	90
R.P.M. (drum)	24	18	16	15	14½	14	12
Mean speed of elevator (ft. per min.)	50	50	45	47	50	50	50
Size of pulleys (ins.)	15×2½	18×4½	21×6½	24×7½	32×9½	42×10½	60×12½
Diameter of driving shaft (ins.)	1½	1¾	2¼	2¼	2½	3	3½
Capacity of water tank (galls.)	4	10	20	30	35	35	70
Diameter of drum	2′ 9″	3′ 6″	4′ 6″	5′ 0″	5′ 6″	6′ 6″	7′ 6″
Width of drum	2′ 0″	2′ 6″	3′ 0″	3′ 6″	4′ 6″	5′ 0″	5′ 9″
Approximate B.H.P. absorbed (varies according to material)	2	5	7	12	15	20	40

Note.—In the production of concrete it is impossible to guarantee definitely the exact amount of mixed material which will be produced from a given quantity of aggregate, owing to the varied nature and size of the stone used, but the above figures, based on actual results, may be taken as approximately correct.

The mixing should be as rapid as possible, consistent with thoroughness, as the concrete should be placed in position before it has begun to set.

Remixed concrete should never be used.

For descriptions of mechanical mixers and further information on mixing concrete, see 'Cement, Concrete, and Bricks,' by A. B. Searle (Constable).

Placing Concrete.

In ordinary foundations, the concrete should be deposited in horizontal layers, not more than a foot thick, and care should be taken to cover any joints in one layer by the succeeding one, as the joint between two days work is always a weak part; moreover, the last-laid layer should be well wetted, to ensure a proper connection with the next.

The chief mistake made in placing mass concrete is in failing to secure sufficient adhesion to the previously placed material. In reinforced concrete there is often a danger of incomplete filling, particularly around the more complex portions of the reinforcement; this may be minimised by the use of a wetter mixture or by efficient damping or vibration. If the latter is effected mechanically, care must be taken not to continue it so long that the setting of the concrete is disturbed.

Fine concrete or cement mortar may be sprayed on to the surface to be covered by means of a Cement Gun or Sprayer, operated by compressed air. This method is particularly effective for 'plastering' with cement mortar, or in the construction of walls about 2 ins. thick on a basis of expanded metal or wire netting.

As the right placing of concrete is of great importance, it may be desirable to consult one or more text-books on the subject.

Various articles formerly made by carving stone are now made of concrete at a much lower cost (see 'Artificial Stone,' p. 358).

Volume of Concrete

The volume of a mass of concrete is much less than that of the separate ingredients, owing to the voids or interstices between the particles of coarse aggregate being occupied, in the concrete, by the sand and cement. Roughly,

Mixture.	Volume of coarse aggregate.	Volume of concrete.
1 : 2 : 4	1 cu. yd.	1·07 cu. yd.
1 : 2½: 5	1 cu. yd.	1·02 cu. yd.
1 : 2 : 8	1 cu. yd.	1 cu. yd.

1 cu. ft. of loose Portland cement will make about :—

4 cu. ft. of 1 : 2 : 4 concrete.	6 cu. ft. of 1 : 3 : 6 concrete.
5 cu. ft. of 1 : 2½: 5 ,,	7½ cu. ft. of 1 : 4 : 8 ,,

Weight of Concrete.

The weight of 1 cu. ft. of concrete varies with the proportion and nature of the aggregate used.

The average weight of 1 : 2 : 4 concrete, using sand as fine aggregate and one of the following coarse aggregates, is : coke breeze, 100 lbs. per cu. ft. ; clinker, 110 lbs. per cu. ft. ; sandstone, 120 lbs. per cu. ft. ; bricks, 125 lbs. per cu. ft. ; limestone, 135 lbs. per cu. ft. ; shingle, 145 lbs. per cu. ft. ; granite, 145–150 lbs. per cu. ft.

Strength of Concrete.*

The proportion of water used in mixing concrete is much more important than is commonly realised, as it has a great influence on the strength of the concrete. According to D. A. Abrams,

$$\text{Compressive strength of concrete in lbs. per sq. in.} = \frac{14{,}000}{7^x},$$

where x is the ratio (by volume) of water : cement. If this is the case, a comparatively small excess of water must have a very harmful effect. The American practice of using a very wet mixture is not to be recommended where great strength is required.

Too little water, as well as a large excess of water, are adverse to the production of strong concrete.

Concrete containing too much cement may be weaker than that containing less cement.

Badly mixed concrete is weak in some parts and strong in others.

The addition of 10 per cent. calcium chloride to the mixing water increases the strength of mass concrete, but cannot be used for reinforced concrete as it corrodes the reinforcement.

The compressive strength of most concretes, as commercially made, can be increased 25 to 100 per cent. or more by employing rigid inspection, which will ensure proper methods of fabrication of the materials.

For equal working consistency and equal cement, gravel concrete is as strong as stone concrete.

30–40 per cent. higher strengths are obtained with 3·16 in. cubes than with 6 in. cubes.

Small cubes are more uncertain and inconsistent in the strength values than larger cubes.

The requirements of the London County Council and the Specification drawn up by a Joint Committee of the Royal Institute of British Architects, District Surveyors' Association, Institute of Builders, Institution of Municipal and County Engineers, War Office, Admiralty, London County Council, and the Concrete Institute, call for a crushing strength of 1,800 lbs. per sq. in. at 28 days and 2,400 lbs. per sq. in. at 90 days, for a 1 : 2 : 4 mixture.

The figures 1,400 at 28 days and 2,000 at 90 days more nearly represent the results obtained under ordinary working conditions, with an aggregate like Thames ballast.

The strength of concrete continues to increase long after it has been made, and is by no means at its maximum after 90 days.

According to the U.S. Bureau of Standards, if S_{28} be the strength of concrete at 28 days, and S_7 the strength at 7 days, then :

$$S_{28} = S_7 + 30\sqrt{S_7}.$$

Weather conditions have a marked effect on concrete work. Heat hastens the hardening process; cold delays it. The effect of cold becomes noticeable when temperatures fall below 50° F., and becomes more marked with lower temperatures. As a rule, concrete does not show any serious effects from having once been frozen if, after it thaws out, it is not again frozen until early hardening is complete. It is better, however, to protect the concrete from freezing for

* See footnote on p. 343.

from forty-eight hours to four or five days, depending upon the degree of the cold, rather than to expose it to the possibility of freezing. Warmth and moisture are necessary to the proper hardening of concrete.

It is especially important to keep concrete thoroughly saturated with water for at least a week, as if it is allowed to dry too soon, it never develops its full strength.

Colouring Concrete and Effect of Addition of Pigments.

Coloured concrete may be obtained by adding the substances described on p. 342, or by using a coloured Portland cement.

Trials have been made by the British Portland Cement Association and others to test the effect on the strength of concrete produced by the addition of pigments. Red oxide of iron produced no effect; green chromium oxide, ultramarine blue, yellow ochre, and manganese dioxide slightly reduced the strength. Carbon black reduced the strength very noticeably.

Effect of Sea-water on Concrete.

The effect of sea-water on concrete depends on the permeability of the latter, dense, waterproof concrete being largely resistant to sea-water, whilst porous concrete corrodes readily. For exposure to sea-water, the concrete should contain a considerable proportion of trass, or equivalent, a suitable mixture consisting of 1 measure of trass, 1 of Portland cement, and 2½ to 3 measures of aggregate. The aggregate should consist of coarse and moderately fine particles, in such proportions as to produce a concrete of maximum impermeability.

Condensation on Concrete.

Interior plastering will prevent condensation on any sort of concrete. A good method is to cover the interior of the concrete walls with ordinary plasterer's lime and sand mortar (about 3 of sand to 1 of lime), lightly keyed in the usual manner to receive the final setting coat. This coat must be left until strong pressure from the thumb makes no impression. It may be necessary to leave it for several days, according to the weather. The finishing coat of lime, putty, sand and plaster of Paris may then be applied with safety. Immediately before this latter coat sets, it should be finished with a soft hair brush, which produces a granular surface, a further method of absorbing moisture.

The best and most easily obtainable finishing coat is the following:

3 parts of lime putty or chalk lime; 6 parts of washed sand; 1 part of plaster of Paris. It gives a good finish, sets well, and works quickly.

Instead of making dense concrete for walls, it is much cheaper and more hygienic to use a very porous concrete, and to render it with a cement waterproofed exterior coating, for the fiercest driving rain cannot penetrate a ¾-in. pudloed cement rendering. Porous concrete, like all porous materials, has also the quality of retaining the heat engendered in the room, thus giving a warmer dwelling. Therefore porous concrete has the dual advantage of assisting the absorption of condensation and of conserving heat.

Preventing Scaling and Cracking of Concrete.

The best method of preventing *scaling* is to keep the concrete thoroughly wet for a week or more after it has been placed, the surface being protected with wet cloths, sawdust, or other porous material. The surface may also be covered with about 2 in. of calcium chloride.

Cracking may be prevented in the same manner, except in those cases where insufficient allowance has been made for stresses due to load or to expansion.

Permeability of Concrete.

The permeability of concrete depends on the number and size of the pores or interstices between the grains. If all the particles were spherical and of the same size, permeability would be at a maximum. It can be reduced by the use of very small particles of sand or of lean clay, but these tend to reduce the strength of the concrete. The addition of lime-soaps (sold under registered names) or of trass also reduces permeability and so does spraying the surface with an impermeable medium (such as a lacquer) or with a substance which combines with some constituent in the concrete (such as silicon ester, some silicofluorides and soluble silicates). (See p. 349.)

Rust Prevention in Reinforced Concrete.

Various methods have been proposed for preventing the reinforcement from rusting. Under ordinary circumstances, no special precautions are necessary, and if the corrosion is due to electric currents no application of chemicals can be satisfactory.

The use of galvanised reinforcement has been stated to prevent rusting, and also to secure greater adhesion between the metal and the concrete.

Steel cannot rust when embedded in a strongly alkaline substance. In the setting of Portland cement concrete a relatively large quantity of caustic lime (lime hydrate) is produced, and the concrete becomes saturated with it. This is strongly alkaline, and by decreasing the number of hydrogen ions present makes any corrosion of the steel impossible.

Coal Residues in Concrete.

A Joint Committee appointed by certain institutes and associations to investigate the subject of the use of coal residues in concrete, has issued a report * giving a summary of the objections and dangers attendant on the use of such, especially in contact with steel.

The objections to this material arise mainly under two headings: (1) The corrosive effect on the steel; (2) expansion.

The report gives in Part 1 the existing laws in Great Britain. Part 2 gives the results of an exhaustive examination as to corrosion, and Part 3 as to expansion. Part 4 contains personal commentaries arising on the two foregoing, together with practical experience of various authorities and individuals on the subject.

There is no regulation whatever as to material used in other concrete in contact with steel.

Treatment of Dusty Concrete Floors.

Various hardening solutions are sold for preventing concrete floors from being dusty, and to give a better wearing surface. The composition of most of these solutions is a trade secret.

Two excellent hardening solutions are the following :—

(1) Mix 1 part silicate of soda to 4 or 5 parts water. Brush or sprinkle the floor well with this solution, and allow to dry. Then, after 4 hours and within 24 hours, wash clean with clean water. Repeat the treatment three times.

(2) The following solution has been recommended by *Concrete* :—

To 10 gallons of cold water add 1 fluid ounce of sulphuric acid. Heat the liquid to boiling point and stir in 25 lbs. of sulphate of alumina. Let the solution cool and strain. Sweep and thoroughly wash the concrete surface. After drying, lay on the hardener in four applications of varying strength, ranging from 30 per cent. solution and 70 per cent. water, up to 100 per cent. solution. The above quantity suffices for the complete treatment of an area of from 50 square yards to 100 square yards, depending upon the porosity of the surface treated.

To Waterproof Concrete.

Concrete may be made waterproof by taking care to use such proportions of the various ingredients as will produce a dense and impermeable material.

As this is not always practicable, it is often preferable to add a waterproofing agent prior to mixing the various ingredients. The chief of these additions are :

(1) *Oil* or a solution of paraffin wax in benzoline or other cheap volatile solvent.

(2) *Insoluble soap* such as a lime-soap, 'Pudlo' being one of the best known. It is not advisable to attempt to prepare a soap *in situ*.

(3) *Water-glass* (silicate of soda) applied in the form of a wash or spray, several applications being necessary at suitable intervals.

(4) *Special mixtures* sold under various trade names, such as Pudlo Waterex, Prufito Prufit, etc.

See also p. 359.

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* Report of Joint Committee of R.I.B.A., Inst. C.E., and other Institutions in conjunction with the Association of Floor Constructors, Victory House, Leicester Square, London, W.C.2. Price 1*s*. 6*d*., postage 1½*d*.

CONCRETE BLOCKS, TILES, AND OTHER ARTICLES.

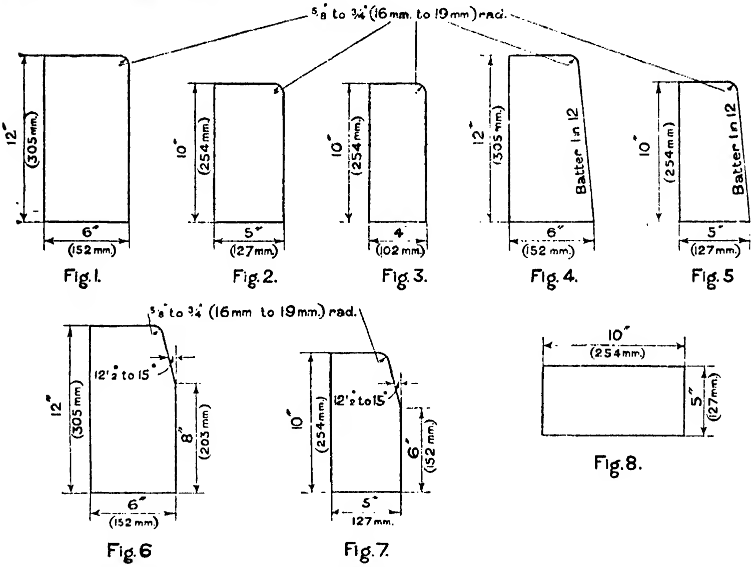
In addition to masses of concrete moulded *in situ*, large quantities of blocks, tiles, garden vases, fencing posts, troughs, and other articles made of concrete or of cement-sand mortar (p. 338) are in regular use, and afford clear proof of the manifold uses of this material.*

See British Standard Specifications Nos. 192 and 834; also see pp. 352, 353, and below.

BRITISH STANDARD SPECIFICATION FOR CONCRETE KERBS, CHANNELS AND QUADRANTS.†

(No. 340—1936.) (*Abstract.*)

The kerbs, channels and quadrants shall be composed of one part by volume of cement ‡ and not more than four parts by volume of aggregate, thoroughly mixed in a dry state Clean water shall then be added and the materials again thoroughly mixed. All the mixing shall be in an efficient mechanical mixer.



FIGS. 1–8.

The *Cement* shall conform to British Standard Specification No. 12 (see p. 355) for Portland cement, or No. 146 for Portland blast-furnace cement or of rapid-hardening cement which complies with the foregoing specifications and, in the case of neat cement and cement and sand, attains the specified breaking strengths in not less than two days and eight days respectively.

*** For further details see *Concrete and Constructional Engineering*, *Everyday Uses of Portland Cement*, and various text-books on concrete.**

† By permission of the British Standards Institution.

‡ The volume is based on an assumed density of 90 lb. per cu. ft. If rapid hardening cement is used, it must be assumed to have a density of only 80 lb. per cu. ft., so that 1·12 volumes will be required to each 4 volumes of aggregate.

The *Aggregate* shall be of best quality, thoroughly clean, and of a material approved by the purchaser. The whole shall pass through a sieve having meshes not more than ¾-in. square.

The *Moulds* shall be accurate in size and form. The inner surfaces shall be clean and true.

Moulding shall be begun immediately after mixing and shall be followed by consolidating by pressure, vibration, or other effective method. Hydraulically-pressed articles shall be subjected to an even pressure over the entire surface of not less than 1,000 lb. per sq. in.

The escape of fine particles shall be prevented as far as practicable. The material shall not be disturbed after the initial set has begun.

No material subjected to frost shall be used until it has been fully thawed.

Dimensions.—Straight kerbs shall be 3 ft. long and to sections shown in figs. 1–7.*

Radius kerbs shall be made to the sections shown in figs. 1–7, and to radii of 6 ft., 8 ft., and 10 ft. respectively, these radii being measured half-way down the outer face of the kerb.

Straight channels shall be 3 ft. long and to the section shown in fig. 8.

Radius channels shall be made to the section shown in fig. 8, and to three radii, 6 ft., 8 ft., and 10 ft. respectively, these radii being those of the inner faces of the channels.

Quadrants shall be 10 in. deep, 18 in. radius, and with faces to match the sections shown in figs. 2, 5, and 7.*

Angles.—All angles (except those resulting from the battered face in figs. 4–7) shall be true right-angles

All *edges* shall be clean and, with the exception of the radiused edge, sharp. The wearing surfaces shall be true, out of winding, and free from excrescences and depressions. All fractures shall present a clean, homogeneous appearance.

Maturity.—No kerbs, channels, or quadrants manufactured less than three months prior to delivery shall be supplied under this Specification unless the whole of the cement used is rapid-hardening cement, in which case they may be delivered fourteen days after manufacture.

Tests are restricted to (i) accuracy of size ; (ii) transverse strength ; (iii) absorption ; and (iv) volume-weight.

Transverse Strength.—The article shall be supported on two steel bearers, each ½ in. wide on the supporting surfaces, placed parallel to each other and 30 in. apart. A kerb shall be placed with the greatest width of wearing face uppermost. A channel with the wearing face uppermost and with the longer sides at right angles to the supports. The load shall be applied to a space 2 in. wide in the centre of the article, extending across its whole width and parallel to the bearers. The load shall be applied steadily and uniformly at a rate not exceeding 112 lb. per foot of width in 10 seconds.

Where necessary the kerb shall be restrained from moving on the bearers, and on battered kerbs a taper strip shall be applied between the kerb and the weight, so that the weight may be truly applied.

Absorption Test.—From each sample kerb or channel a test piece approximately cubicle in form, weighing between 4 and 6 lb. and having not less than three cut faces, shall be taken. The test pieces shall be dried for 72 hours in a suitably ventilated drying oven, the temperature of which, as measured by a thermometer suspended centrally, is between 185° and 204° F. On removal from the oven they shall be weighed and immediately submerged in water, the temperature of which is between 58° and 64° F. for a period of ten minutes, at the end of which time they shall be taken out, immediately wiped with a dry cloth for a period of half a minute and again weighed. They shall then be submerged in water again for a total period of twenty-four hours, at the end of which time they shall be taken out, wiped with a dry cloth and weighed.

Volume-Weight.—A sample which has been dried at 100° C. shall not weigh less than 140 lb. per cu. ft.

* The ⅜–½-in. radii shall in each case be formed along the edge joining the two wearing faces.

BRITISH STANDARD SPECIFICATION FOR CONCRETE FLAGS.*

(No. 368—1936.) (*Abstract.*)

The flags shall be composed of one part by volume of cement † and not more than three parts by volume of aggregate.

The *cement* shall be as for kerbs, channels, etc. (see p. 350).

Flags made with rapid-hardening cement shall have the letter R impressed on the back. This impression shall be 2 in. high and $\frac{1}{4}$ in. deep.

The *aggregate* shall be as for kerbs, etc. (see p. 351), but the whole shall pass through a sieve having meshes not more than $\frac{3}{8}$ in. square.

The conditions for moulds, moulding and freedom from frost and defects are the same as for kerbs, etc. (see p. 351).

Dimensions.—The flags shall be of the following sizes :—

3 ft. × 2 ft. × 2 in.	2 ft. × 2 ft. × 2 in.
3 ft. × 2 ft. × $2\frac{1}{2}$ in.	2 ft. × 2 ft. × $2\frac{1}{2}$ in.
2 ft. 6 in. × 2 ft. × 2 in.	1 ft. 6 in. × 2 ft. × 2 in.
2 ft. 6 in. × 2 ft. × $2\frac{1}{2}$ in.	1 ft. 6 in. × 2 ft. × $2\frac{1}{2}$ in.

Tests.—(i) The aggregate shall be tested for size and the flags for (ii) transverse strength; (iii) rate of wear; (iv) absorption; and (v) volume-weight.

Transverse Strength.—The flag shall be supported on horizontal hard, unyielding bearers, each $\frac{1}{2}$ in. wide on the supporting surface, placed parallel to each other and 18 in. apart. The wearing surface of the flag shall be uppermost and the longer sides at right angles to the supports. The load shall be applied to a space 2 in. wide in the centre of the flag, extending the whole width and parallel to the bearers, and shall be applied at a uniform rate not exceeding 112 lb. per ft. of width per ten seconds.

Flags 2-in. thick shall support a load of at least 1,120 lb. for each foot of width for at least one minute. For flags $2\frac{1}{2}$ in. thick this figure shall be increased to 1,736 lb.

Rate of Wear.—The samples shall be 2 ft. × 1 ft. They shall be dried at not more than 37° C. and then tested by an apparatus which consists of two end-plates mounted on a shaft so as to form, with four samples, a rectangular drum with the samples as sides.

A charge of 1,000 balls of hard steel or chilled cast iron, each with a diameter of $\frac{3}{8}$–$\frac{7}{16}$ in., is placed in the drum, which is then revolved for 24 hours at a regular speed of 60 r.p.m. in one direction, and a further 24 hours in the opposite direction.

The wear on the faces of the samples shall be uniform, and when the faces have been dried as before and the faces brushed free of dust, the loss in weight shall not exceed 3 lb.

Absorption.—Samples about 18 in. square, having two moulded and two cut edges shall, after being dried at 37° C., be weighed, immersed in water for 48 hours, wiped and re-weighed, and shall not show an increase in weight of more than $1\frac{1}{2}$ per cent.

Volume-Weight.—A sample, prepared as for the absorption test, shall weigh not less than 140 lb. per cu. ft.

BRITISH STANDARD SPECIFICATIONS FOR CONCRETE BLOCKS.*

(Nos. 492 and 834.)

The sizes, composition, mode of curing and tests are all specified.

CONCRETE PIPES AND TUBES.

Concrete pipes are of two kinds :—

(i) *Plain pipes* made by simple moulding without reinforcement and only suitable for situations in which great strength is not required.

(ii) *Reinforced pipes* which are designed to resist any pressure likely to be applied to them in the ordinary course of use.

For *plain pipes* see British Standard Specification No. 556—1934 ‡ with a revised absorption test (described in OE(PW)3124 ‡ issued in March 1937), also No. 1194.

* By permission of the British Standards Institution.
† See footnote ‡ on p. 350.
‡ Issued by the British Standards Institution.

Its chief provisions are:—

A *tube* is defined as a hollow cylinder of uniform diameter throughout its length or with an ogee or rebate at each end.

A *pipe* is defined as a hollow cylinder with a socket at one end.

The cement, aggregate, composition, some of the dimensions and conditions of maturing are specified.

The tests are: a hydraulic test to show impermeability to water under pressure, an absorption test, and a crushing test.

For *reinforced pipes* see p. 405. There is no Standard Specification for reinforced concrete pipes.

BRITISH STANDARD SPECIFICATIONS FOR PIPES OF ASBESTOS CEMENT ETC.

Rainwater Pipes, Gutters and Fittings,* No. 569—1934.

Flue-Pipes and Fittings for Stoves,* No. 835—1939.

Soil, Waste and Ventilating Pipes,* No. 582—1934.

Pipes for Gas-Fired Appliances,* No. 567—1934.

Pressure Pipes,* No. 486—1933.

BRITISH STANDARD SPECIFICATION FOR ASBESTOS SHEETS.

British Standard Specification No. 690—1940 relates to asbestos sheets—both flat and corrugated.

BRITISH STANDARD SPECIFICATION FOR CONCRETE ROOFING TILES.*

For Plain Tiles—No. 473—1944.

For Interlocking Tiles—No. 550—1934.

In both these Specifications, the nature of the cements, aggregate, and pigment to be used, also shape and size of the tiles are specified. The only tests are for the modulus of rupture and the permeability.

Adhesive Cements.

Cements are materials used for uniting other materials or articles so that they adhere permanently. There are many kinds of cements, but the more important ones may be grouped as follows:—

Mastic cements, in which the chief ingredient is a gum (such as resin, shellac, wax, or rubber), boiled oil, or bituminous material to which is added a 'filler,' such as sand, rock dust, or other inert powder. If boiled oil is used red lead must usually be added as well as the filler. Mastic cements are used for uniting leather, paper, wood and other materials to each other or to metal, stone or slate. They are also used to fill joints in building- and road-construction as they are waterproof.

Rubber cements are chiefly made by dissolving shredded rubber in carbon disulphide or other solvent. They are chiefly used for uniting leather or rubber.

Fish-glues or *liquid glues* are made by treating fish bones with acetic acid. *Seccotine* is of this nature.

Casein cements consist of casein (milk curd) and an alkali; they are resistant to moderate heat, water, and oil vapours, but not to acid fumes. For use, the cement is mixed with water; if mixed with a weak solution of gelatine glue the strength will be about 10 per cent. greater. Average breaking stress, about 1,000 lbs. per sq. in.

Bituminous cements are those in which the chief ingredient is asphalt, tar, pitch or bitumen (see also *Mastic cements*). They are used for uniting glass for photographic and microscopical uses, also for coating wood, concrete, brick work, steel, stone, slate, stoneware, etc., and for *caulking*.

Water-glass or *silicate cements* consist chiefly of an inert material mixed (such as asbestos, sand, glass-powder or silica flour) with water-glass (syrupy silicate of soda) or solid silicate of soda.

* Published by the British Standards Institution, London.

Calcareous cements, such as hydraulic lime, Roman cement, the various natural cements and Portland cement. *Hydraulic limes* are described on p. 339. *Roman cement* is a brown, quick-setting cement, made by calcining and grinding the septaria or calcareous claystones found in London clay. It is one of the natural cements.

Natural cements (so-called) are made by heating natural mixtures of clay and limestone to redness, so as to effect a combination between the lime and clay similar to that in Portland cement. Such natural cements are comparable to a mixture of Portland cement with either an excess of lime or clay, as the case may be. Their composition is irregular and uncertain, so that they are inferior to Portland cement for all work of importance.

Portland cement is described on pp. 355–357.

Ciment Fondu or *Electric Cement* is composed of lime and alumina, heated to incipient fusion in an electric furnace and afterwards ground to powder. This high alumina cement attains a great strength very much more quickly than Portland cement, and is said to be unaffected by sea-water. It should comply with the requirements of British Standard Specification No. 91b — 1940.

Rapid hardening cements are a special variety of Portland cement or of ciment fondu which attains almost its maximum strength in three days, and is, therefore, invaluable in road work and in some building construction (see p. 357).

Gypsum cements include plaster of Paris, Keene's cement, Parian cement (see *Plasters*, p. 357).

Oxychloride cements, of which the best known is *Sorel cement*, are composed of dense magnesia and syrupy magnesium chloride, or zinc oxide and zinc phosphate. They depend very largely on the use of suitably calcined oxides and on the chloride or phosphate being of suitable density, so that it is wise to follow the manufacturers' instructions.

Magnesium oxychloride cements may be diluted with fillers such as china clay, or may be coloured with mineral pigments such as ultramarine, ochre, etc. They are largely used for forming *floors*; these should not be repeatedly subjected to the action of water as this tends to soften them and causes them to wear badly (see p. 305).

Pastes made of flour or starch are largely used for fastening papers to each other or to walls, and ceilings. (See p. 359.)

Celluloid cements are made by dissolving celluloid or similar materials in a suitable solvent, such as amyl acetate. They are excellent adhesives for general use and will satisfactorily unite pottery, paper and wood, china, silk or cotton fabrics, leather, etc. They are waterproof.

Celluloid cements for various purposes are on the market; they are sometimes known as *dopes*.

Refractory and *fire-resisting cements* are those which are specially resistant to heat and are used in repairing furnaces, etc. (see p. 373)

Glues.

Commercial glue is made by boiling animal flesh, bones, etc., in water, removing fat and other mpurities and concentrating by evaporating the greater part of the water. It is usually sold in thin cakes or flakes, or in the powdered form, and should be kept in a dry place.

See British Standard Specifications Nos. 745—1937, and 647—1938.

Ground glue should be soaked in cold water at least three hours, flake glue, eight hours and cake glue, twelve hours. It should then be placed in a glue-kettle and heated until soft and tacky; its temperature should not exceed 150° F.

After being heated, the glue may be diluted with hot water to the required consistency. Glue should never be heated sufficiently to make it boil. It is also bad practice to prepare large quantities of glue at a time, because it loses its strength rapidly after being under heat for about four hours.

Glue that has cooled should never be reheated or it will lose about 50 per cent. of its strength.

Core glues are largely used in foundries. They consist of sand mixed with an adhesive, such as dextrine, resin soap or powdered glue to unite the sand grains into a firm mass.

Waterproof glues are best if bought ready-made, as most of the published recipes are unreliable. Casein cements, celluloid cements, and mastic cements are all waterproof.

Marine glue is a mixture of rubber, shellac, and naphtha. It has now been largely replaced by mastic cements.

For *fish glues* and *liquid glues*, see p. 353.

Portland Cement.

Portland cement is usually made from a mixture of chalk, or limestone, and clay ground together to an extreme degree of fineness, and subsequently dried, calcined, and ground to the fineness of flour. Pats or briquettes made from it should, when broken, exhibit a bluish-grey colour in the fracture.

Portland cement * may be obtained quick or slow setting. If there are no special reasons which make quick setting desirable, it is advisable to employ a cement which requires at least two hours to pass from the pasty state into the hard, solid one, as quick-setting cements require exceptional care and skill in gauging; otherwise they begin to set and to spoil before they have been placed in their proper position in the structure.

Most manufacturers add a little gypsum when grinding the cement clinker, and thereby adjust the rate of setting of the cement to suit the user's requirements. If required, the time taken for setting may be increased by exposing the cement to moist air before use, but this is not altogether satisfactory, and it is better to purchase a cement which sets at the desired rate.

Temperature has a very great effect upon the setting time of cement. A sample tested in a temperature of 61° Fahr. and gauged with water at the same temperature took six hours to set hard. The same sample gauged in a temperature of 90° Fahr. and gauged with water of equal temperature set hard in one hour fifty minutes. This is a point which is often lost sight of.

Testing Portland Cement.

The tests suggested by the British Standard Specification (*below*) are usually regarded as sufficient, but others are used for special purposes. The following notes indicate the purpose of the various tests:

Chemical Composition.—This test is for the detection of adulterants, or to determine whether certain constituents are present in amounts exceeding that believed to be safe.

Fineness.—The fineness of the cement is a measure of its cementing value; a fine cement will be much stronger when mixed into a mortar, or it can be mixed with a larger proportion of sand than a coarse one, and yet attain the same strength.

Crushing or Compression Strength.—The most generally comprehensive test of a Portland cement or of a mixture of it with sand would be a determination of the crushing strength of a cube of 4 in. side. Unfortunately, this requires a very costly machine, so that it is usual to substitute a tensile test which can be made much more cheaply.

Tensile Strength is determined in order to gain some idea of the cohesion between the particles. This property is not required in cement and concrete, but as it is roughly proportional to the crushing strength and the tensile strength is easier to determine, it is usually substituted for a crushing test.

Time of Setting.—This is determined in order to ascertain whether a cement sets at a rate suitable for a given piece of work. If a cement sets before it has been properly placed in position the work will be defective; hence where there is any doubt, a slow-setting cement should be used. There is no necessary relationship between the time of setting and that of hardening or attaining the maximum strength; a slow-setting cement may harden more rapidly than a quick-setting one, and *vice versâ*. *Rapid Hardening* cements and the aluminous cements, such as *Ciment Fondu*, set slowly but harden very rapidly.

Soundness.—The object of this test is to hasten the rate at which cement or concrete attains its maximum strength and to subject it to conditions more severe than are likely to occur in regular use. By thus exaggerating the conditions some idea is gained of the lasting properties of the material much more rapidly than would otherwise be the case. The soundness test is in some respects, the most important of all, for if a sample passes all other tests satisfactorily yet fails in the soundness test, it is worthless as a constructional material. To pass the soundness test a cement or concrete must retain a *constant volume* under the conditions of the test; failure is revealed by the test-piece cracking, swelling, or disintegrating in any other way.

The actual testing of cement and concrete requires considerable skill. The methods described in the British Standard Specification for Portland Cement (see below) are satisfactory, but require considerable practice before accurate results can be obtained.

British Standard Specification for Portland Cement.†

Ordinary and Rapid Hardening—No. 12—1940. (*Abstract.*)

The cement shall be manufactured by intimately mixing together calcareous and argillaceous or other silica and alumina bearing materials, burning them at a clinkering temperature and grinding the resulting clinker, so as to produce a cement capable of complying with the specification.

* The composition of Portland cements is highly complex. Some further indications of it are given in 'Cement, Concrete, and Bricks,' by A. B. Searle (Constable & Co. Ltd., London.).

† By permission of the British Standards Institution.

No addition of any material shall be made after burning other than calcium sulphate, or water, or both. For a specification which permits added slag, see No. 146—1932 (*below*).

The sample or samples shall be tested for :—(*a*) fineness ; (*b*) chemical composition ; (*c*) tensile strength (cement and sand) ; (*d*) setting time, and (*e*) soundness.

Fineness.—The residue on a sieve of B.S. Mesh No. 170 not to exceed 10 per cent. With a Rapid Hardening cement the residue not to exceed 5 per cent.

Chemical Composition.—

(*a*) The percentage of lime, after deduction of that necessary to combine with the sulphuric anhydride present, not to be more than 2·8 times the percentage of silica + 1·2 times the percentage of alumina + 0·65 times the percentage of iron oxide nor less than two-thirds of that amount.

(*b*) The Loss on Ignition not to exceed 3 per cent. in temperate climates, or 4 per cent. in hot climates.

(*c*) Insoluble Residue not to exceed 1 per cent.

(*d*) Magnesia not to exceed 4 per cent.

(*e*) Total Sulphur (calculated as sulphuric anhydride) not to exceed 2·75 per cent.

Tensile Strength (cement and sand).—3 : 1 sand-cement mortar not less than 300 lb. per sq. in. at 3 days. Strength at 7 days to be greater and not less than 375 lb. per sq. in.

Tensile strength of *Rapid Hardening* cement-sand mortar not to be less than 300 lb. per sq. in. at 1 day. Strength at 3 days to show an increase and not be less than 450 lb. per sq. in.

Compressive Strength (alternative to Tensile Strength) on 2·78 in. cubes made of cement 185 g., sand 555 g., water 74 g. at 3 days not less than 1,600 lb. per sq. in. At 7 days to show an increase and not be less than 2,500 lb. per sq. in. *Rapid Hardening* cement-sand mortar at 1 day not to be less than 1,600 lb. per sq. in. At 3 days to show an increase and not be less than 3,500 lb. per sq. in.

Setting Time.—The normal initial setting time should not be less than 30 minutes and the final setting time not more than 10 hours.

A quick setting cement shall have an initial setting time of not less than 5 minutes, and a final setting time of not more than 30 minutes.

Soundness.—The cement shall be tested for soundness by the Le Chatelier method, in the manner specified, and shall not expand more than 10 mm.

Effect of Climate.—Cement intended to be used in hot climates may be tested at any temperature up to 95° F. (instead of 58–64° F. required in temperate climates) and shall then conform to the Standard Specification. The requirements as to the setting time and strength may be altered if agreed between vendor and purchaser.

British Standard Specification for Portland Blast-Furnace Cement.*

(*No.* 146—*Revised* 1941.) (*Abstract.*)

This cement shall consist of a mixture of Portland cement clinker and granulated blast-furnace slag in such proportions as the manufacturer may prefer, but in no case shall the proportion of slag exceed 65 per cent.

The Portland cement clinker and the granulated blast-furnace slag shall be thoroughly and intimately mixed, and shall produce a cement capable of complying with the specification.

The cement shall comply with the following conditions of fineness :—100 grammes (or, say, 4 ozs.) of cement shall be continuously sifted for a period of 15 minutes on a B.S. Mesh 170 sieve and the residue, by weight, shall not exceed 10 per cent.

The cement shall comply with the following conditions as to its chemical composition. The percentage of insoluble residue shall not exceed 1 per cent. ; that of magnesia shall not exceed 5 per cent. Sulphur present as, and calculated as, sulphuric anhydride shall not exceed 2 per cent., and sulphur present as sulphide shall not exceed 1·2 per cent. sulphur, these being equivalent to a maximum total of 5·00 per cent. of sulphuric anhydride. The total loss on ignition shall not exceed 3 per cent. The composition of the Portland cement clinker portion of the mixture shall comply with the requirements of the current British Standard Specification for Portland Cement.

Tensile Strength at 3 days not to be less than 300 lb. per sq. in. The tensile strength at 7 days shall show an increase on the breaking strength at 3 days and shall be at least 375 lb. per sq. in.

The methods of making the tests are the same as for Portland cement (see p. 355).

* By permission of the British Standards Institution.

Rapid Hardening Cements.

Concrete made with some Portland cements and also concrete made with aluminous cement (*Ciment Fondu*) (see p. 354) are as strong after 24 hours as ordinary Portland cement concrete which is 90 days old.

Concrete made with 'Ferrocrete' (a rapid hardening Portland cement) has in 4 days the same strength as concrete made with ordinary cement has in 28 days; there is also a continuous increase in strength. Most rapid hardening cements and aluminous cements set slowly, but harden rapidly (see p. 354).

The British Standard Specification for Rapid Hardening Cements is No. 12—1940; that for Aluminous Cements is No. 915—1940.

Waterproof Cements.

Strictly speaking, no calcareous cement is wholly waterproof, but concrete can be made sufficiently so for most purposes by using suitable aggregates and the correct proportion of water, cement and aggregate. Where additional certainty is needed one of the materials described on p. 349 (waterproofing concrete) should be used.

PLASTERS.

Plasters are pasty materials applied to walls, ceilings, etc., to give a hard, smooth finish. They resemble lime mortars in many respects but serve a different purpose. Common plaster is made of lime and sand, with or without ½ lb. of hair to each cu. ft. of plaster. It is made harder and has smoother finish if 20 per cent. of plaster of Paris is added.

According to a tentative specification issued by the Department of Scientific and Industrial Research (*Building Research, Special Report, No.* 9, 1927) the lime to be used for plaster, fine and coarse stuff, should be *Quicklime, Class A*, or *High Magnesium Quicklime, Class A*, and, for coarse stuff only, *Quicklime, Class B* (see p. 340).

The British Standard Specification for gypsum and anhydrite plasters is No. 1191.

For *floors*, a hard-burned plaster of Paris is used, or a mixture of ordinary plaster with a slight excess of lime is laid, and then treated with zinc sulphate (white), iron sulphate (red), boiled oil (mahogany), or copal varnish.

Stucco for external use consists of 1 part of hydraulic lime to 3 parts of sand, but for more durable work a cement mortar (p. 338) is often used.

Rough cast is sometimes made of the same materials as stucco, or of cement mortar (p. 338).

Plaster of Paris is made by calcining gypsum and grinding the product. The resulting powder, when mixed with water, forms a smooth paste which rapidly sets to a white, solid mass.

Hard Plasters.

Keene's, Martin's, and Parian cements are hard-setting forms of plaster of Paris. Keene's cement is made by soaking calcined gypsum in a solution of alum or borax and cream of tartar. For Martin's cement, a solution of potassium carbonate is used, and Parian cement is an intimate mixture of gypsum and borax which has been calcined and then ground to powder. Both thes and other hard cements can be purchased and are more satisfactory than home-made preparations.

Plastering.

Coarse stuff is mortar composed of 1 part lime (carefully-slaked fat or pure lime is generally used) and 1 or 2 of sharp fresh-water sand, with 1 lb. hair to 3 cu. ft of mortar.

Hydrated lime is usually superior to lime which has been slaked by hand and can be used at once, whereas ordinary slaked lime must be kept in the form of a putty until slaking is complete.

Fine Stuff is made of pure lime slaked with a little water, after which water is added to bring it to the consistency of cream; it is then left to settle, the superfluous water poured off, and the water evaporated until it is of proper thickness. In ceilings a small portion of white hair is mixed with it.

Bricklayers' and plasterers' putty is fine stuff brought to the proper consistency.

Gauged stuff is ¾ fine stuff mixed with ¼ plaster of Paris.

Common stucco is 1 lime to 3 or 4 clean-washed sand.

Bastard stuff is ⅔ fine stuff and ⅓ clean sand, with a little hair.

Lath-and-Plaster, or *One-Coat Work.*—A coat of coarse stuff, ½ inch to ⅝ inch thick. laid on the laths and smoothed off with the trowel.

Lath, Plaster, and Set, or *Two-coat Work.*—The first, or 'pricking-up' coat of coarse stuff is scored with a birch broom or undercut with a lath, and a thin finishing coat of fine stuff or gauged stuff added.

Lath, Plaster, Float, and Set, **or *Three-coat Work.*—When the pricking-up coat is dry, the surface is scored, and divided into 8-foot squares by *screeds* of coarse stuff, 8 inches wide, carefully levelled. These are then filled in with coarse stuff, floated carefully level. The finishing coat is then put on as before.**

Materials.	1 in. Thick.	¾ in. Thick.	½ in. Thick.
1 bushel of cement, or 1·28 cubic feet, will cover .	1 1/7 sq. yds.	1½ sq. yds.	2¼ sq. yds
1 cement and 1 sand ,, .	2¼ ,,	3 ,,	4½ ,,
1 cement and 2 sand ,, .	3⅓ ,,	4½ ,,	6¾ ,,

1 cubic yard of lime, 2 yards of sand, and 3 bushels of hair will cover 75 yards superficial, render and set, on brick, or 70 yards on lath.

Reconstructed or Artificial Stone.*

ARTIFICIAL BUILDING STONE, NO. 1.—Mix 100 parts of hydraulic lime which has fallen to a powder with water to form a paste. To this add 250 parts of gravel and 50 of coal ashes or lixiviated wood ashes. The mass is then thoroughly mixed, and a sufficient quantity of water added to make the volume of the mass equal to 500 parts. It is then poured into moulds made of pine-boards, where it is allowed to remain until set.

No. 2.—125 parts of hydraulic lime which has fallen to a powder are mixed with a sufficient quantity of water to form a paste. To this are added 250 parts of ground oyster shells and 150 parts of ground peat ashes, and a sufficient quantity of water to make the bulk of the mass equal to 500 parts. It is then poured into moulds as in No. 1, and dried.

ARTIFICIAL STONE FROM QUARTZ SAND AND PLUMBIC OXIDE.—Ground quartz sand is mixed with 2 to 10 per cent. of finely-ground plumbic oxide. The harder the stones are to be, the more plumbic oxide must be used. The mixture is moistened with water-glass, again thoroughly mixed, and then pressed firmly into moulds. The resulting stone is dried and then burned.

VICTORIA STONE.—The refuse of granite quarries is broken up into pieces of suitable size, and 4 parts of the fragments thus obtained are mixed with 1 part of Portland cement, and sufficient water added to bring the whole to the consistency of dough. The mass is run into moulds, in which it is allowed to remain for several days, or until it has set solid. The blocks are then immersed in a solution of silicate of soda.

RANSOME'S ARTIFICIAL STONE.—Clean and dry sand and other suitable siliceous and earthy ingredients are thoroughly incorporated in a mixing mill with silicate of soda. The resulting pasty mass is then pressed into moulds of any required pattern or size, and when set sufficiently immersed in a solution of chloride of calcium. In the case of large pieces the saturation with chloride of calcium is facilitated by the use of an air-pump. The resulting reaction is the formation, by double decomposition of the ingredients of an insoluble calcium silicate out of sodium chloride. The first-named forms a solid and indurate binding material for the stone, and the sodium chloride is removed by a subsequent thorough washing with water. This last operation is important, since, if the sodium chloride is not completely removed from the stone, it will make its appearance subsequently in the form of a white efflorescence on the surface.

TO IMITATE VARIEGATED MARBLE.—Mix hydraulic lime and ground marble and incorporate with the mixture a solution of alum and suitable colouring substances. Differently coloured masses are then mixed together and cut into slabs.

ARTIFICIAL MARBLE.—The following mixtures have been recommended for making artificial marble. Grind and thoroughly mix comminuted stone, 280 parts; limestone or chalk, 140 parts; burned calamine, 5 parts; calcined felspar, 3 parts; fluorspar, 2 parts; calcium phosphate, 2 parts; water-glass, 40 parts.

On the addition of the water-glass the ingredients are quickly mixed, and thereupon pressed into moulds. The finished pieces are dried at a temperature gradually rising to 125° F.

RECONSTRUCTED STONE.—Many artificial or reconstructed stones sold under proprietary names are really carefully prepared concrete which has been moulded to the desired shapes. See p. 350 and British Standard Specification No. 834—1939.

*** The manufacture of artificial stones, *lime-sand* bricks and blocks, is dealt with at length in 'Bricks and Artificial Stones of Non-Plastic Materials,' by A. B. Searle (J. & A. Churchill, London).**

To Remedy Dampness in Walls.

Before attempting to cure dampness, serious efforts should be made to ascertain the cause which is often remote from the damp spot. For instance, a leaky channel or valley in the roof may introduce water into the wall, but the water may not be revealed until it reaches a more porous part of the structure or one from which it cannot drain freely. Too low a flashing behind a parapet wall is a common cause of dampness. When the source has been definitely found it is usually easy to prevent further trouble. The serious cases are those in which the source is not discovered.

Other methods of reducing or preventing dampness in walls include:

1. Cut out a course of bricks foot by foot, insert an asphalt damp-course (British Standard Specification No. 743—1937), and re-brick.

2. Remove plastering and re-point wall outside after wall has become dry. If it is inconvenient to renew plaster, the latter (although wet) may be varnished over with a solution of shellac in naphtha (4 ozs. shellac, 1 quart naphtha); this almost immediately hardens, and paper may be pasted over it.

3. A good coating of hot coal-tar laid on the exterior of a brick wall will prevent the entrance of rain, but should only be applied where its appearance is not objectionable. Spraying the wall with a silicon ester (silicone) will often prevent the further infiltration of rain and is less disfiguring than tar.

4. Cover the surface of the wall with a mixture of 1 part of Portland cement with 2 parts of fine sand, and water enough to bring the ingredients to the consistency of thick cream. Apply two coats of the mixture, and when it is quite dry finish with a coat of paint.

5. Re-plaster the wall with sand and cement impregnated with one of the special water-proofing agents, such as 'Pudlo,' 'Bareau,' 'Waterex,' or 'Silicon Ester.'

6. Spray or paint the wall with: (*a*) a solution of water-glass (sodium silicate), followed by a solution of calcium chloride, which forms an insoluble silicate; (*b*) a solution of various fluo-silicates with a subsequent application of calcium chloride to form an insoluble calcium fluosilicate in the pores of the wall; or (*c*) one of various substances, dissolved in acetone, alcohol, or other volatile solvent, which remain in the pores when the solvent has evaporated; (*d*) silicon ester.

Each of these methods is suitable for certain walls, but some walls appear to resist every kind of treatment yet applied to them.

British Standard Specification No. 743—1941 relates to *Damp-Proof Courses.*

Effects of Moisture Changes on Building Materials.

A Bulletin (No. 3) on this subject has been issued by the Building Research branch of the Department of Scientific and Industrial Research. Price 1*s*. net.; postage extra. H.M. Stationery Office, London, York House, Kingsway, W.C. 2.

FLOUR PASTE. (See p. 354.)

The most satisfactory method of making flour paste is to mix flour with a little cold water, stirring until it forms a smooth cream. This is then poured slowly into boiling water, with constant stirring. The total weight of water is usually five times that of the flour (4 oz. of flour to each pint of water). The mixture is boiled for a short time, with constant stirring, and is then allowed to cool.

For some purposes an adhesive paste with twice as much water is satisfactory.

Before the mixture is quite cold a little formalin or carbolic acid may be added and thoroughly stirred in: it acts as a preservative.

See also Descriptive Section X.

SECTION XI

CLAYS AND ALLIED PRODUCTS USED BY ENGINEERS

CLAYS AND EARTHS — BRICKS — TILES — BLOCKS AND TERRA-COTTA — STONEWARE, SANITARY WARE, AND DRAIN PIPES — ELECTRICAL WARE — REFRACTORY MATERIALS — FIREBRICKS — FURNACE LININGS — RETORTS — CRUCIBLES — HEAT-INSULATING MATERIALS — FIRE-RESISTING CEMENTS.

(pp. 363–377)

Contributed by Alfred B. Searle
(Consulting Advisor on Clay Products).

SECTION XI

CLAYS AND ALLIED PRODUCTS USED BY ENGINEERS.

CLAYS AND EARTHS—BRICKS—TILES—BLOCKS AND TERRA-COTTA—STONEWARE, SANITARY WARE, AND DRAIN PIPES—ELECTRICAL WARE—REFRACTORY MATERIALS—FIRE-BRICKS—FURNACE LININGS—RETORTS—CRUCIBLES—HEAT-INSULATING MATERIALS—FIRE-RESISTING CEMENTS.

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CLAYS AND EARTHS.

For engineering purposes, clays and earths may be regarded as (1) substances which must be excavated and transported from their present situation; (2) the raw materials from which earthwork, puddle, etc., are made; and (3) the raw materials used in the production of bricks, drain pipes and many other articles employed by engineers. The term 'clay' in the broadest possible sense is applied to a material which is tenacious and plastic when wet. Such clays may contain only a small proportion of true clay, together with a large proportion of inert material such as sand. For many engineering purposes a perfectly pure clay would be useless, as it might be of so fine a texture and so highly plastic that it would shrink excessively on drying, and it would be extremely difficult to make into bricks or other desired articles. When made, these articles would be deficient in certain important properties, such as resistance to crushing, impermeability to water, etc.

The term 'earth' is usually applied to a friable material partaking of the general properties of soil, though not necessarily of any agricultural value. It usually consists chiefly of inert material, the particles of which are united by a very variable proportion of clay, humus, or other adhesive. Hence the 'earths' are often regarded as 'clays' of low plasticity, but both these terms are used very loosely by many engineers and no really satisfactory definition of them has yet been formulated.

In selecting a 'clay' or 'earth' for any given purpose, the engineer must, therefore, decide what properties are essential to his requirements and must endeavour to find a natural material possessing these properties, or to prepare a mixture of two or more materials which suit his purpose.

The chemical and physical properties of 'clays' depend chiefly on the composition and plasticity of the material, it being obvious that a 'clay' containing 50 per cent. of quartz-grains must behave very differently from one which consists wholly of 'true clay.' The *plasticity* of clays varies with the nature of the clay and the amount of water present. Dry clays are friable and devoid of plasticity; on mixing clay with an equal weight of water a liquid slurry or slip is formed which is not really plastic, but becomes so when a suitable proportion of the water has been removed by evaporation or otherwise. No generally accepted explanation of the cause of plasticity has been published; this property may be due to the presence of colloidal particles which may be derived from the clay itself or from impurities which have become associated with it. The purest clays—e.g. 'china clay'—are very feebly plastic; the highly plastic ball clays which are supposed to have been derived from them are rich in carbonaceous matter, to which their plasticity is sometimes attributed. The clays commonly used for bricks, pipes, terra-cotta, etc., are never pure; they are naturally occurring complex mixtures possessing the requisite properties to make them useful for specific purposes.

It is a great mistake to suppose that any plastic 'clay' can be made into bricks, or that any 'system' of manufacture is equally suitable for all 'clays.' On the contrary, the manufacture of clay products is in reality so complex (notwithstanding its apparent simplicity) and the ordinary manufacturer depends on the simultaneous occurrence of so many fortuitous circumstances that it is not surprising that many firms fail to manufacture clay goods of uniform quality.*

BRICKS.

Many kinds of bricks are in use. The most important are: *Common Bricks*, used for ordinary building purposes; *Facing Bricks*, used for external work where appearance is important; *Paviours*, used for yards, footpaths, and other locations where a hard-wearing and impervious brick is required (for *Road Bricks*, see p. 562); *Blue Bricks*, which are similar in many respects to 'paviours' and are employed where exceptional strength and impermeability are important; *Stock Bricks*, which are practically the same as 'common bricks,' but have usually been burned in a clamp instead of in a kiln; they have proved exceptionally resistant to the climatic conditions of London, but are seldom, if ever, used in other large industrial centres (in some localities the term 'stock bricks' relates to 'facing bricks').

Bricks are also known according to the manner of their production. Thus, *Malms* contain a large proportion of chalk, either naturally or artificially mixed with the clay; *Wire-cut Bricks* are made by cutting up a column of clay issuing from a machine, by means of wires; *Pressed Bricks* have been made in presses; *Semi-plastic* or *Stiff Plastic Bricks* have been made from a paste which is stiffer and less plastic than that used for hand-moulded or wire-cut bricks; *Semi-dry Bricks* are made by compressing the powdered clay in powerful presses instead of first converting it into a paste.† *Place Bricks*, *Grizzles*, and *Chuffs* are inferior bricks which are soft through insufficient burning; *Crozzles* are excessively hard, misshapen bricks which have been partially melted and overheated. *Rubbers* and *Cutters* are soft, sandy bricks.

Glazed Bricks are used where a smooth and impervious surface is required (see 'Glazed Tiles').

Sand-lime Bricks.—Sand-lime bricks are made of high silica content mixed with about 8 per cent. of lime. After pressing they are placed in autoclaves, or 'hardening chambers,' fed by steam under pressure where an irreversible chemical change occurs and calcium is formed; the absence of 'burning' results in a brick of perfect shape and square corners. The efficiency of the manufacturing process enables them to be sold at a price comparable with the cheapest clay bricks (see p. 366). Sand-lime bricks have been made in this country for nearly fifty years but their serious entry into the market became noticeable only in the last fifteen years, during which time they have made marked commercial progress. Their strength is equal to that of good quality clay bricks and their porosity is no higher. Unjustifiable doubts have often existed on the latter point through their tendency to show change of colour on immersion.

Cement-concrete Bricks.‡—These bricks have attracted attention, but are not so suitable as larger blocks of concrete for general purposes, as the cost of production is relatively high. See also p. 367.

Engineering Bricks is the term usually applied to bricks of an exceptionally dense and vitreous character of which the best known are the *Staffordshire* blue bricks. These owe their unusual strength and impermeability to the fact that they have been burned under such conditions that the particles composing them are united very firmly by means of a film of clinker surrounding each. For purposes where a somewhat weaker brick has ample strength, well-made *pressed* or *stiff plastic* bricks are usually satisfactory and much less costly, but they are more porous and therefore less suitable for supporting heavy masses of material on a boggy or damp subsoil. Bricks for *boilers* should be accurate in size and shape so as to enable the thinnest possible joints to be used. In order to prevent undue leakage they should be glazed or painted with tar or other impervious material.

For *fireboxes*, *furnaces*, etc., firebricks and blocks are employed.

Colliery Bricks are usually of very poor quality, due to their being made from unsuitable material, or from good clay which has been spoiled by mixing it with rubbish.

Panel Bricks have a hollow portion (known as a *frog* or *panel*) which reduces the weight and may serve as a key for the mortar. According to the Ministry of Health Model Byelaws, Series IV, Buildings (1938), the volume of solid material shall not be less than one-half of the total volume and the aggregate width of the solid material (measured horizontally at right angles to the face) shall not be less than one-third of the width of the brick.

Cavity Bricks usually have a larger hollow portion than panel bricks; the same regulation as to the volume of the cavity applies to them as to panel bricks.

Hollow Bricks and Blocks.—See p. 369.

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* For further information, see 'The Chemistry and Physics of Clays and other Ceramic Materials,' by A. B. Searle (E. Benn Ltd., London).

† For further details respecting the manufacture of bricks see 'Modern Brickmaking,' by A. B. Searle (E. Benn Ltd., London).

‡ See 'Bricks and Artificial Stones of Non-Plastic Materials,' by A. B. Searle (J. & A. Churchill, London).

Crushing Strength of Bricks.

Opinions differ greatly as to the respective values of different kinds of bricks, and as the crushing strength of bricks taken from the same works at different times varies 25 per cent. and even more, it is clear that no comparison of crushing strength tests can be of much value in deciding between various kinds of bricks. The chief use of the tests is in fixing a minimum crushing strength which shall ensure the safety of the structure, but as the strength of even the commonest bricks is much greater than that of average brickwork (the mortar being usually weak), it is difficult to stipulate any satisfactory minimum strength for bricks.

The following figures are the averages of a large number of bricks, but it must be clearly understood that bricks from any given works may be either much stronger or much weaker than the ones mentioned.

Description.	Crushing Commences.	Ultimate Crushing.
	Tons per square foot.	Tons per square foot.
Common stocks	40–60	80–120
London best stocks	80–140	100–180
Gault white, wire-cut	40–120	135–200
Aylesford, red pressed	70	140
Manchester, common red	75	120
" wire-cut	85	260
Accrington, plastic *	690	1005
Leicester, wire-cut	115–250	230–340
Rugby, common red	160	190
Fletton	125–200	170–250
Notts.	60–300	120–400
Ruabon.	360	—
Staffordshire blue bricks	250–450	400–750
Terra-cotta blocks	—	100–260
Glazed bricks	70–165	165–175
Stourbridge fire bricks	90–160	110–210
Sand-lime bricks	150–200	180–230

The crushing strength of brickwork is shown on p. 180.

See also Building Research Special Report No. 22 (1934) (H.M. Stationery Office, London), and London County Council Byelaws, 1938.

Sizes of Bricks.

In many countries there is no official standard; the following are average sizes:—

Standard Sizes of Bricks in Different Countries.†

Country.	Length, inches.	Breadth, inches.	Thickness, inches.
Great Britain	9	$4\frac{3}{8}$	$2\frac{5}{8}$ ‡
Germany	10	$4\frac{3}{4}$	$2\frac{1}{2}$
Austria	$11\frac{3}{8}$	$5\frac{5}{8}$	$2\frac{5}{8}$
France	$8\frac{5}{8}$	$4\frac{3}{8}$	$2\frac{1}{4}$
Italy	9	$5\frac{7}{8}$	$2\frac{3}{8}$
Holland and Belgium	$9\frac{3}{8}$	$4\frac{1}{4}$	$2\frac{3}{8}$
Switzerland	10	$4\frac{3}{4}$	$2\frac{3}{8}$
Russia (varies greatly).	$11\frac{1}{4}$	$5\frac{1}{4}$	$2\frac{3}{4}$
United States	8	4	$2\frac{1}{4}$

Standard Specification for Bricks.

The sizes of bricks vary in different parts of Great Britain, but there are many advantages in adopting the British Standard Specification No. 657—1936 and 1941, summarised on p. 366.

Unfortunately, many users still order bricks of non-standard sizes and so fail to secure the advantages they might obtain, viz. readier and more accurate designing of clear openings in brickwork, more accurate assessment of the number of bricks required, reduction in the cost of bricks and cost of laying which arise when different sizes are used for facings and backing, and quicker delivery from stock.

* These figures supplied by The Accrington Brick and Co., 1926.

† See also p. 316.

‡ In Scotland, $2\frac{7}{8}$ in. or 3 in.

BRITISH STANDARD SPECIFICATION FOR BRICKS MADE OF CLAY,* No. 657—1941.
(*Abstract.*)

The dimensions of the bricks shall be as shown in the following table :—

Length.		Width.		Depth.					
				Type I.		Type II.		Type III.	
In.	Total in.	In.	Total in.	In.	Total in.	In.	Total in.	In.	Total in.
8¾	± ⅛	4 3/16	± 1/16	2	± 1/16	2⅝	± 1/16	2⅞	± 1/16

Method of Measurement.—The bricks laid dry shall measure as follows :—

(*a*) Eight bricks laid dry, end-to-end, in contact, in a straight line.
Maximum length, 71 in.
Minimum length, 69 in.

(*b*) Eight bricks laid dry, side-by-side, in contact, in a straight line.
Maximum length, 34 in.
Minimum length, 33 in.

(*c*) Eight bricks laid on edge, bedding surface to bedding surface, in contact, in a straight line.

	Type I.	Type II.	Type III.
Maximum length	16½ in.	21½ in.	23½ in.
Minimum length	15½ in.	20½ in	22½ in.

Snap Headers shall be 4 3/16 ± 1/16 in. in length, with width and depth as in the table.

Closers shall have the length and depth shown in the table, and the width shall be 2 ± 1/16 in.

All backing bricks should have the same dimensions as the facing bricks.

Joints should be ¼ in. thick, and an extra 1/16th, making 5/16th, for the bed joints to cover irregularities in the bricks. This gives a standard length of 9¼ ins. centre to centre of joints.

TESTS OF BRICKS.

All dry bricks should *ring clearly* when two, held in the hands, are struck sharply together. If a dull sound is emitted the bricks are either saturated with water or they have not been sufficiently well burned.

The *porosity* of bricks is usually tested by soaking them in water, wiping dry and noting the increase in weight. Whilst many good building bricks will absorb one-fifth of their weight of water, this is no criterion of the value from an engineering standpoint ; on the contrary, engineers usually prefer bricks with low porosity on account of their greater strength.

Rattler Test.—In the United States, and to a much smaller extent in this country, engineering bricks are tested for toughness and resistance to abrasion by placing a weighed quantity of them in a cylinder with steel balls and rotating the cylinder a thousand times in about half-an-hour.† The contents of the cylinder are then removed, all pieces of brick larger than 1 in. diameter are picked out and reweighed. The loss in weight is regarded as a measure of the inferiority of the bricks. The rattler test is particularly applicable to bricks used for road making and for lining furnaces.

For standard methods of testing bricks, see British Standard No. 1257.

SPECIFICATION FOR SAND-LIME BRICKS.‡

(No. 187—1942.) (*Abstract.*)

(Sand-lime bricks are described on p. 364.)

Four classes of sand-lime bricks are recognised: —
Bricks for special purposes.
Building Bricks, Class A (i).
Building Bricks, Class A (ii)..
Building Bricks, Class B (for internal use only).

The length of the bricks shall be 8¾ in. ± ⅛., and the width shall be 4 3/16 ± 1/16 in. unless otherwise agreed.

* The term 'clay' is used in a general sense to exclude sand-lime bricks and glazed bricks.

† The weight of the balls, the number of revolutions, and the size of the rejected pieces, differ with various engineers.

‡ By permission of the British Standards Institution.

The thickness shall be agreed between the manufacturer and the purchaser and shall not differ from the agreed figure by more than ± $\frac{1}{16}$ in.

Samples.—Fifteen bricks per 100,000 (or other agreed number from stock) shall be taken; twelve to be used for the Crushing Test or Transverse Test and three for the Chemical Tests.

Fifteen other bricks to be sealed and reserved in case of dispute.

Strength Test may be either Crushing Test or Transverse Test, as agreed.

Crushing Test.—On twelve whole bricks tested flat in wet state after soaking in cold water for 24 hours:—

Bricks for special purposes . . . 2,500 lb. per sq. in.
Building bricks, Class A . . . . 1,750 lb. per sq. in.
Building bricks, Class B . . . . 1,000 lb. per sq. in.

Transverse Test.—On twelve whole bricks in wet state, laid flat with frog uppermost on rounded supports of ½ in. radius and 7 in. apart, the load being applied at mid-span through a similar bar of ½ in. radius. Modulus of rupture to be not less than:—

Bricks for special purposes . . . 500 lb. per sq. in.
Building bricks, Class A . . . 350 lb. per sq. in.
Building bricks, Class B . . . . 200 lb. per sq. in.

Uniformity.—The average crushing strength (or the average modulus of rupture) of the seven bricks giving the lowest results expressed as a percentage of the average crushing strength of the twelve bricks should not be less than:—

Bricks for special purposes . . . 90 per cent.
Building bricks, Class A . . . . 80 per cent.
Building bricks, Class B.

Composition.—The bricks shall consist essentially of an intimate uniform mixture of siliceous sand and slaked lime, combined by the action of high-pressure steam and, after being crushed and screened through a British Standard test-sieve No. 52, shall correspond on testing in approved manner to:—

	Total Silica not less than per cent.	Total Free Lime not more than per cent.	Total Calcium Oxide present as Silicate not less than per cent.
Bricks for special purposes .	75	1·0	3·5
Building bricks, Class A . . .	75	1·5	3·0
Building bricks, Class B . . .	75	2·0	3·0

For further details of the tests, see Specification No. 187—1942.

Concrete Bricks.

Bricks made of 6 parts of sand and 1 part of Portland cement are cheap to produce when the sand is on the building site. Ordinarily they cost more than clay bricks. They are made in plain moulds, and must be kept wet for a fortnight. Concrete bricks are as durable as sandstone: they are suitably made in isolated places where sand is available and carriage is costly. See British Standard Specification No. 657—1941.

TILES.

A 'Code of Practice for Roofing with Plain Tiles,' published by the Building Industries National Council, 110 Bickenhall Mansions, W.1. (Price 9*d.*) should be read.

Roofing Tiles are made of materials similar to those used for bricks, but of a somewhat finer texture. No official specifications have been published, but it is recognised that roofing tiles should be light, tough, and ring clearly when struck, be impervious to water in the sense that even under a continuous supply of water no dripping will occur from the underside of the tile when placed at the same angle as when in use on a roof. It is desirable that tiles should be moderately porous, so as to avoid undue condensation of their underfaces. (See p. 299.)

BRITISH STANDARD SPECIFICATION FOR CLAY OR MARL PLAIN ROOFING TILES.*

(No. 402—1930.) (*Abstract.*) (See also No. 1424—1948.)

British Standard Plain Roofing Tiles may be either hand-made or machine-made, as specified. They shall be made from well-weathered and ground clay or marl.

Particles of lime, visible to the naked eye, either on a surface or a fracture, shall be a cause for rejection, unless the lime has been water-slaked by *docking* the tiles as they come from the kiln.

* By permission of the British Standards Institution.

The tiles shall be true in shape, free from fire-cracks, dense, tough, show a clean fracture when broken, and well-burned throughout.

Nibs.—Hand-made tiles shall have not less than two nibs, each not less than $\frac{3}{4}$ in. wide and a depth of $\frac{3}{8}$–$\frac{1}{2}$ in.

Machine-made tiles shall have not less than two nibs not less than $\frac{3}{4}$ in. wide or a continuous nib $\frac{3}{8}$–$\frac{1}{2}$ in. deep.

Note.—Nibless tiles may be used by agreement.

Nail-holes.—Two nail-holes shall be provided, the centre of each being 1–1$\frac{1}{4}$ in from the side of the tile and $\frac{5}{8}$ in. from the under side of the nib. The holes shall not exceed $\frac{1}{4}$ in. in diameter.

Size.—The standard size is 10$\frac{1}{2}$ in. × 6$\frac{1}{2}$ in., with a tolerance of ± $\frac{1}{8}$ in. on the width and ± $\frac{1}{4}$ in. on the length.

Tiles 11 in. × 7 in. may be used if this size is definitely stated.

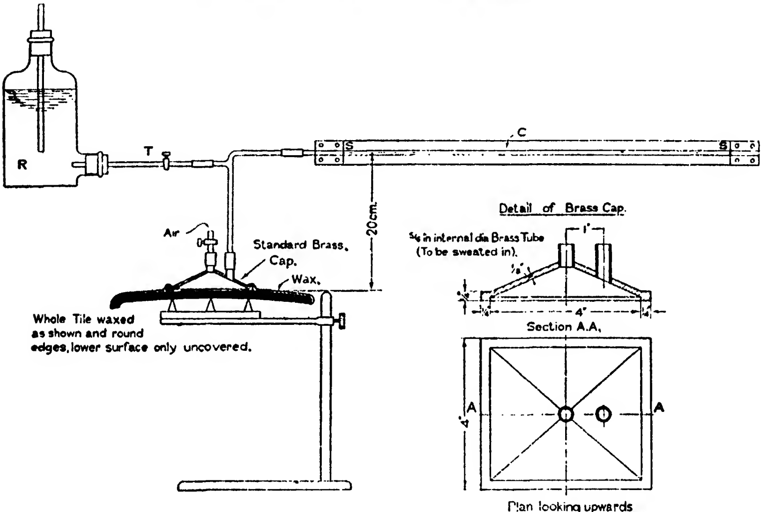

FIG. 1.

Thickness.—Hand-made tiles shall not be less than $\frac{1}{2}$-in. thick. Machine made tiles shall not be less than $\frac{3}{8}$ in. thick.

Camber.—Hand-made tiles shall have a camber of $\frac{3}{16}$–$\frac{7}{16}$ in.

Machine-made tiles shall have a camber of $\frac{1}{8}$–$\frac{1}{4}$ in.

Transverse Test.—The average breaking load applied along the centre line of a tile at right angles to its length shall not be less than :—

	10$\frac{1}{2}$ × 6$\frac{1}{2}$ in. tile.	11 × 7 in. tile.
Hand-made tiles . . .	175 lb.	189 lb.
Machine-made tiles . . .	125 lb.	189 lb.

The tiles shall be soaked in water for 24 hours prior to testing and shall be tested wet. They shall be supported on wooden bars 1 in. wide, rounded on the bearing edges to $\frac{1}{8}$ in. radius, and placed at 7$\frac{1}{2}$ in. centres. The load shall be applied through a similar wooden bar, placed parallel to the supports. The average of six tiles should be taken. If the test fails a further six tiles may be tested.

The test shall be made before delivering.

Freezing Test.—Four tiles shall be soaked by gradual immersion in water, 4 hours being required for immersion, followed by a further 20 hours. The tiles shall then be wrapped in wet cloths and immersed in a freezing mixture of ice and salt (4 :1 by volume) for 24 hours at a temperature below − 10° C. They shall then be thawed for 24 hours at 16° C. and be again immersed in the freezing mixture, the process being repeated ten times.

The tiles shall not show any sign of lamination, pitting, or cracking.

The test shall be made before delivery.

Permeability Test.—Three tiles shall be dried at 90–100° C., then waxed on to a special metal cap with Faraday wax (fig. 1). The upper surface and sides shall be coated with Faraday wax as far as the four sides of the cap. Tubing is used to connect the interior of the metal cover to a head of water and to a capillary tube. The rate of flow of water is deduced from the rate at which the water travels along the capillary tube when the tap, T, is closed.

The rate of flow through the specimen with the tap, T, open shall not in 24 hours exceed that indicated by a rate of flow of 4 in. per min. along a capillary tube of 1 mm. bore under a head of 20 cm. The test shall be made before delivery.

Concrete Roofing Tiles should comply with British Standard Specification No. 473—1932 or No. 550—1934.

Glazed Tiles are used partly for decorative purposes, but chiefly for sanitary reasons. The glaze should be sufficiently hard to resist all reasonable abrasion and it should not craze (forming hair-like cracks) after prolonged use. For chemical laboratories and other special purposes a felspathic glaze burned at about 1300° C. must be used in order that it may be sufficiently resistant to corrosive fluids, etc. (see 'Stoneware,' p. 370).

Paving Tiles, like paving bricks, should be very hard and resistant to abrasion. To prevent animals slipping on them, the surface may be channelled with small grooves forming a diamond pattern.

Paving tiles are usually made of the same material and in the same manner as paving or engineering bricks.

BLOCKS AND TERRA-COTTA

The chief requirements of *blocks* made of clay and allied materials are accuracy in shape and size, and ample strength to carry the superimposed loads. For external work, appearance is also important.

Terra-cotta is largely used in conjunction with structural steel work, the latter taking the bulk of the stresses and the terra-cotta forming a sort of filler and cover of great resistance to climatic conditions and of pleasing appearance. There is no necessity to impose any conditions as to its strength, as in a properly designed structure it would not be exposed to any severe stresses.

Solid Terra-cotta weighs 117–148 lbs. per cu. ft.

Blocks for boilers are made of special shapes. They must be sufficiently heat-resisting not to be affected by the conditions to which they are subject when in use. They must be very accurate in shape and size or there will be a serious leakage into the side flues.

HOLLOW BLOCKS.

For many years in France and in some of the Colonies, but more recently in this country, the advantages which hollow blocks possess over building bricks have been realised. Such blocks are only one-third of the weight of the same number of bricks; they can be laid about four times as rapidly, and they are of ample strength for all purposes for which ordinary bricks are used. In addition, they automatically form a cavity wall, with all its advantages of warmth

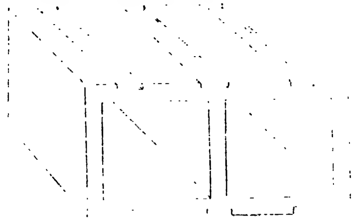

FIG. 2.

and soundproofness. Their great strength is really due to the fact that the bricks in a solid mass of brickwork have about ten times the crushing strength of the brickwork as a whole. Consequently a large proportion of the solid interior is quite useless and only adds to the cost of material, carriage, and labour; these costs are saved when hollow blocks are used.

Hollow building blocks are made in various shapes and sizes, according to the purposes for which they are to be used, and numerous patents have been granted in connection with them.

The *clays* used for making hollow blocks are the same as those employed in the manufacture of terra-cotta and roofing tiles. Such clays must vitrify sufficiently to form a strong, weather-resisting mass when fired. They must retain their shape perfectly at any temperature to which they may be exposed in the kiln, and whenever possible they should have a pleasing colour so as to avoid the necessity of 'rough casting.' Suitable clays and shales are found in various parts of the country, particularly in the Midlands. The lower grades of fireclays found in the various coalfields are also suitable, provided their buff colour is not regarded as objectionable.

The *process* of manufacture consists in preparing a clay paste of suitable consistency and extruding it through a die or mouthpiece, the interior of which is fitted with a number of cores, corresponding to the hollow portions of the blocks, fig. 2. The extruded column of clay is then cut by wires into suitable lengths, and the resulting blocks are dried and burned like bricks or architectural terra-cotta.

No standard specifications for hollow blocks have as yet been issued, but it is obvious that if the crushing strength of such blocks when they are laid in the same position as that in which they will be used, is greater than the normal strength of brickwork, such blocks will have ample strength. In most cases the blocks themselves have more than twice the strength of normal brickwork.

Where hollow blocks of exceptional lightness or great insulating power are needed, the clay is mixed with diatomite (kieselguhr) or sawdust.

Hollow Concrete Blocks 8½ in. thick weigh 4·25 lb. per sq. ft. for each inch of overall thickness of block, as laid. Similar blocks 2½ in. thick weigh 6·25 lb. per sq. ft. for each inch of overall thickness of block, as laid.

Hollow Blocks of Diatomaceous Earth weigh 2·50–2·65 lbs. per sq. ft. for each inch of thickness of block, as laid.

STONEWARE, SANITARY WARE, AND DRAIN PIPES.

Stoneware, as used by engineers, is chiefly employed in chemical works in the construction of appliances for dealing with corrosive fluids both at low and high temperatures.

The conditions under which it is used are exceptionally severe and yet are so varied that no widely acceptable specifications have been found possible. For chemical and industrial purposes, stoneware should consist of an impervious body of great crushing strength and with a minimum sensitiveness to sudden changes in temperature. The ideal is closely approached by the porcelain crucibles used in chemical laboratories, but this is too costly and too fragile for most commercial purposes and various coarser substitutes are therefore employed. To a very large extent the manufacturers of stoneware appliances may be relied upon to supply a satisfactory product if they are fully informed as to the conditions to which it will be subjected. Where troubles arise, they are often due to an engineer requiring a piece of stoneware of a design or shape which is incompatible with the nature of the material used.

Sanitary Ware is not subjected to the same conditions as stoneware, and may therefore consist of a porous body covered with a hard felspathic glaze, though an impervious body is preferable where expense is no object. It is highly important that the glaze should be of a suitable character, earthenware and lead glazes seldom being sufficiently durable. In designing sanitary ware it is desirable to work in close co-operation with the manufacturer, as in this way much delay in production and a considerable saving in cost may be effected, and the ware is likely to be more satisfactory. As far as possible, joints and sharp arrises or corners should be avoided.

Drain Pipes are frequently made of a porous body which is covered with a salt-glaze, but for the best work a vitreous or impervious body should be used. If the glaze is defective or damaged an impervious body will maintain the efficiency of the pipes, but a porous body may permit seepage of a disagreeable or even harmful character.

British Standard Specifications No. 65—1937 and 539—1937, deal with salt-glazed drain pipes, bends, junctions and fittings, and No. 540—1937 deals with salt-glazed glass (vitreous) enamelled fireclay pipes, *i.e.* pipes and fittings which have been glazed internally with a preparation of ground glass or other suitable materials and externally with salt-glaze. The required dimensions and permissible variations are stated and the hydraulic test and absorption test are described. All pipes marked 'Tested' must have been tested hydraulically and have shown no leakage when subject to a hydraulic pressure of 20 lb. per sq. in. for one minute.

The Specifications are illustrated and are of such a nature that they cannot be briefly summarised so that direct reference to them is essential.

According to the Ministry of Health's Model Byelaws, Series IV, Buildings, 1938, all drain-pipes must conform to British Standard Specification No. 65—1937, or No. 540—1937, or No. 437, or No. 556, or No. 1191, according as they are made of burned clay, cast iron or concrete.

ELECTRICAL WARE.

Electrical Ware is chiefly used in the form of conduits, insulators, and fittings. The conduits are usually made of coarse stoneware and are similar in many respects to salt-glazed drain pipes (*supra*).

Other electrical ware is made of fine stoneware or porcelain. Insulators must have a fine and uniform texture and their fractured surface should be free from holes and fissures; they must be able to resist the passage of electric currents of the desired voltage without appreciable leakage, and should be tested at a considerably higher voltage to ensure this.

REFRACTORY MATERIALS.*

The term '*Refractory*' is applied to various heat-resisting materials, such as firebricks, crucibles, and furnace linings. There is no official definition of this term, but it is becoming increasingly customary to regard 1580° C. (2876° F.) as a minimum temperature, materials which

* See also *Refractory Materials, their Manufacture and Uses* (3rd edn.), by A. B. Searle (Griffin & Co., Ltd., London).

suffer deformation when heated rather slowly to this temperature being excluded from what are generally regarded as refractory materials.

As the rate of heating has an important influence on the temperature at which deformation occurs, it is necessary, for comparative purposes, to specify the conditions of heating.

MELTING POINTS OF REFRACTORY MATERIALS.

	°C.		°C.
Alumina (pure)	2010*	Ganister	1700 to 1800
Bauxite	1820*	Kaolin	1835 to 1740*
Bauxite brick	1565 to 1785*	Magnesia brick	2165
Bauxite clay	1795*	Silica—Pure	1750*
Chromite	2180*	Pure (fine grains)	1650
Chromite brick	2050*	Pure with 14·5 per cent. alumina	1680
Fireclay brick—Highest grade	1843	Pure with 63·0 per cent. alumina	1790
General grade	1680 (average)	Brick	1700 to 1750
General grade	1649 (average of 41 samples)*		

The term 'melting point' is misleading when applied to refractory materials, as these are seldom, if ever, melted, and the term is generally used to indicate the temperature at which a small pyramid or 'cone' of the material bends over until its apex is on a level with its base, or, alternatively, at which the angular edges of the pyramid begin to lose their angularity.

FIREBRICKS.

Fireclay bricks are made of clays derived chiefly from the coal measures. These clays are crushed to powder, mixed with water and made into bricks either by hand-moulding or machines of the stiff-plastic or semi-dry type; the bricks are then dried and are afterwards burned in kilns. The temperature attained in burning should be at least as high as that to which the bricks will be subjected when in use, as otherwise shrinkage and other defects may occur. It is seldom that bricks burned at such high temperatures are obtained unless they are definitely specified; and for some purposes, such as lining steel-melting furnaces, users must be content with bricks fired at a lower temperature than that at which they will be employed. The highest temperature ordinarily attained in burning fireclay bricks is about 1350° C.; in silica bricks 1500° C.

Firebricks are also made of silica (in which case they are known as *Ganister* or *Silica Bricks*), alumina (*Bauxite* or *Semi-bauxite Bricks*), magnesia (*Magnesia* or *Magnesite Bricks*), chrome iron ore (*Chromite Bricks*), sillimanite (*Sillimanite Bricks*), zirconia (*Zirconia Bricks*).

Firebricks made of clay or silica are acid in character and are readily attacked by the limestone used as a flux in steel furnaces.

The materials of which *acid* bricks are made may be classified into three groups:

(*a*) Aluminous clays.
(*b*) Siliceous clays.
(*c*) Silica rock (sometimes erroneously termed 'silica clays').

The *Aluminous clays* include the Stourbridge, Yorkshire, Scotch, Welsh, and other Coal Measure fireclays, which contain, on an average, about 65 per cent. of silica. The *Siliceous clays* include those which contain 80–92 per cent. of silica, such as Ewell clay. *Silica rock* contains more than 92 per cent. of silica, such as Dinas rock and ganister. The composition of some of these materials is shown in the following table †:

	Stourbridge Clay.	Ewell Clay.	Dinas Clay.	Black Ganister.‡
Silica	64·62	87.57	97.60	98·5
Alumina	21·65	3·60	0·50	0·3
Ferric oxide	1·48	4·80	1·50	1·3
Lime	1·88	0·74	0·20	0·2
Magnesia	0·62	0·41	—	—
Potash	—	0·69	0·09	trace
Soda	—	0·10	0·08	trace
Moisture	9·62	—	—	—
Combined water	—	2·09	—	—

* Kanolt. The temperatures are those at which the materials were first seen to flow easily.

† Analyses of these and other heat-resisting materials will be found in *Refractory Materials*, by A. B. Searle (Griffin & Co., Ltd., London).

‡ The Dinas material is not a true clay, but a silica rock composed of almost pure quartz, ganister is similar, but consists of much smaller particles.

Ordinary fireclays contain 58 to 75 per cent. of silica, 25 to 36 per cent. of alumina, and ½ to 2 per cent. of iron oxide. Good silica bricks contain upwards of 95 per cent. of silica, little or no alumina, and under 2 per cent. of lime. Ganister bricks contain about 90 per cent. of silica, 5 per cent. of alumina, and less than 2 per cent. of iron oxide. Magnesia bricks consist almost wholly of magnesia, unless about 5 per cent. of clay or iron oxide has been used to bind the particles together.

Basic Bricks are made of magnesia, whilst *Neutral Bricks* are made of bauxite, sillimanite, zirconia, chromite, or carbon; the two latter are chiefly used as an intermediate course between acid and basic bricks in furnace linings. The materials used for silica and neutral firebricks usually need a 'binder' such as fireclay or lime, but the clays are self-binding. Firebricks made of sillimanite appear to be specially suitable for use at very high temperatures. As natural sillimanite ($Al_2O_3.SiO_2$) is a rare mineral which has to be imported from Northern India, an artificial material made by heating fireclay and bauxite in a blast-furnace is frequently used instead; the artificial product is not true sillimanite, but more closely resembles millite ($3Al_2O_3.2SiO_2$). Those made of zirconia have proved disappointing—chiefly on account of the impurities in the zirconia used.

The impurities in fireclays, i.e. those constituents other than the silica and alumina, have a most harmful effect on the refractoriness of the material owing to their low fusing point. These constituents should, if possible, never be allowed to exceed more than 4 to 5 per cent. Much must, of course, depend upon the type of clay, and it is possible to point to certain substances containing so much as 4 per cent. of iron alone, and yet they are highly refractory. Green clay when fired is subject to considerable shrinkage, and it is for this reason that chamotte, or 'grog,' is added in varying proportions before the article is placed in the kiln. This 'grog' is usually prepared from a clay which originated in the same seam as the clay with which it is mixed. It should be prepared specially for the purpose, and should not consist of the crushings from defective and broken articles which have passed through the kiln. The quantity of 'grog' added varies, of course, with the material required and the kind of clay made use of. For firebricks and smaller articles it is usually added in a proportion of 15 to 20 per cent. For larger articles, such as retorts, the amount is increased to 30 to 40 per cent.

The deformation or softening points of firebricks vary considerably. The following are fairly typical:—

Fireclay bricks, 1610°–1770° C.; Silica bricks, 1670°–1790° C.; Magnesia bricks, 1730°–2000° C.

Highly porous fireclay bricks—see p. 377.

Specification for Fireclay Bricks.

(Institution of Gas Engineers, 1934.)

(1) *Refractoriness.*—Two grades of material are covered by the specification:—

Grade No. 1.—The material which shows no sign of fusion when heated to a temperature of not less than Seger Cone 29 (about 1650° C.).

Grade No. 2.—Material which shows no sign of fusion when heated to a temperature of not less than Seger Cone 26 (about 1580° C.).

Note.—Firebricks with a refractoriness of Cone 32 are deemed to be 'Special Refractory Materials,' for which no Specification has been prepared.

(2) *Surfaces and Texture.*—The material shall be evenly burnt throughout and the texture regular, containing no holes or flaws. All surfaces shall be reasonably true.

(3) *Contraction or Expansion.*—Two test pieces, when heated to and maintained for two hours at a temperature of 1410° C. shall not show more than the following linear contraction or expansion:—

No. 1 grade, 0·75 per cent.; No. 2 grade, 1·15 per cent.*

The test pieces shall be 2½–3 inches long and 1½–2 inches square, the ends being ground flat, and the contraction being measured by means of vernier calipers reading to 0·1 mm., a suitable mark being made on the test piece, so that the calipers may be placed in the same position before and after firing.

(4) *Variations from Specified Dimensions.*—In the case of ordinary bricks, 9 ins. by 4½ ins. by 3 ins. or 2½ ins. thick, there shall not be more than ± 1½ per cent. variation in length, nor more than + 1½ per cent. or − 2½ per cent. variation in width and ± 2 per cent, variation in thickness. In the case of special bricks, blocks or tiles, there shall not be more than ± 2 per cent. variation from any of the specified dimensions.

(5) *Crushing Strength.*—The cold material shall be capable of withstanding a crushing strain of not less than 1,800 lbs. per sq. in. when applied to whole bricks placed with their long side vertical between the jaws of the machine, giving a vertical thrust. Not less than three bricks must be used.

* A tolerance of 0·1 per cent. is allowed.

SPECIFICATION FOR SILICA BRICKS

(Institution of Gas Engineers, 1934.)

The material covered by this specification is divided into two classes:—

(*i*) That containing 92 per cent. and upwards of silica, and hereinafter called 'silica' material.

(*ii*) That containing from 78 to 92 per cent. of silica, and hereinafter called 'siliceous' material.

(1) *Refractoriness.*—Test pieces of the material shall show no sign of fusion when heated to the following temperatures:—
'Silica' material not less than Seger Cone 31 (about 1690° C.).
'Siliceous' material not less than Seger Cone 29 (about 1650° C.).
The test shall be carried out in an oxidising atmosphere, the temperature of the furnace being increased at the rate of about 50° C. per five minutes.

(2) *Surface and Texture.*—The material should be evenly burned throughout, and the texture regular, with no holes or flaws. All surfaces shall be reasonably true.

(3) *Contraction and Expansion.*—Two test pieces when heated in a gas muffle or electric furnace to a temperature of 1450° C., and maintained at that temperature for two hours, shall not show on cooling more than 0·5 per cent. linear contraction or expansion, with a tolerance of 0·1 per cent.

Siliceous material shall pass the same test, but at 1410° C.

The test pieces shall be the same size and shape as in Clause 3 in the specification for Fireclay Bricks (above).

(4) *Variations.*—In the case of all bricks, blocks and tiles there shall not be more than $\pm 1\frac{1}{4}$ per cent. variation in length, width or thickness except for dimensions of 3 ins. or less, when the permissible variation shall be $\pm \frac{1}{16}$ in.

Marking.—All bricks or blocks shall be distinctly marked to indicate the grade to which they belong.

Inspection and Testing.—The arrangements for inspection and for making tests are specified.

SPECIFICATIONS FOR REFRACTORY MORTAR OR CEMENT.

(Institution of Gas Engineers, 1911, but withdrawn in 1934.)

Cementing clay or fireclay mortar shall be machine-ground, and, at the discretion of the manufacturer, may contain a suitable percentage of fine grog; but in all cases the cement-clay shall be quite suitable for the purpose of binding together the bricks, blocks or tiles for which it is supplied, and shall be capable of withstanding the same test for refractoriness.

FIREBRICKS USED IN GASWORKS.

The firebricks used in gasworks should conform to the specifications of the Institution of Gas Engineers (p. 372). They are usually, but by no means invariably, made of the materials mentioned on p. 371.

BRICKS OR BLOCKS FOR FURNACE LININGS.*

The bricks or blocks used for lining furnaces should be selected according to the conditions to which they will be subjected. It is almost impossible to obtain a material which is wholly efficient in every respect, and some compromise is essential. Thus, great resistance to abrasion can only be obtained by increasing the amount of vitreous material and thereby lowering the refractoriness of the bricks. Answers to the following questions will usually provide most of the information required:—

(*a*) What is the highest temperature to which the brick will be exposed (excessive or only moderate), and will it be constant or fluctuating? (*b*) Will the material come into contact with any flame, and will the latter be of an 'oxidising' or 'reducing' character? (*c*) Is mechanical strength required? (*d*) Is heat to be conducted or retained? (*e*) Is the brick expected to withstand sudden and repeated changes in temperature? (*f*) Will the brick come into contact with fluid slags or glass, or with flue-dust, cement or other fluxes? (*g*) Must the brick resist blows, shocks and other unavoidable strains? (*h*) What is the nature of the fuel used and is it fed by hand or mechanically? (*i*) Is the electrical conductivity important?

To meet (*a*), refractoriness is of much importance, but, in most cases, other factors deserve even greater attention. For (*b*) and (*c*) the texture of the surface of the brick must be right or the brick will not be durable, no matter how great its refractoriness. If mechanical strength

* For a much fuller description of the properties required in bricks and blocks used for various furnaces, kilns, etc., see *Refractory Materials*, by A. B. Searle (Griffin & Co., Ltd., London. 5th Edn.).

is needed, the brick cannot be as refractory as otherwise, for great strength is dependent on the presence of a large amount of bond, which is necessarily less refractory than the particles to be bound together. For (*d*), it will be found that the conduction of heat depends chiefly on the density of the material if made of clay or silica and on its chemical composition if made of graphite, plumbago or other carbonaceous material. Graphite, fused alumina and zirconia are good heat conductors, whilst clay, silica and magnesia are good heat retainers. For (*e*) a lean, coarse and porous material should be used, whilst for (*f*) a fat, close material is required, i.e. exactly opposite to that for (*e*), and for (*g*) a hard-burned material which has not been heated more than once is preferable; broken bricks and other refractory articles should not be used as grog in this case, and the mass should not be either too lean or too fine-grained.

Consideration of the above points will show that there is some difficulty in deciding upon the correct description of the material for use in the ordinary type of industrial furnace. Where bricks of high refractoriness are required to withstand the temperature employed, heat must be retained and radiation reduced to a minimum, though most of these requirements are satisfied by using an open-grained, porous material. Still greater difficulty arises in dealing with furnaces in which fluxes are brought into contact with the firebricks, and in these the material must be close-grained, even if this is accompanied by the loss of some heat resistance.

Where great resistance to fluxes (flue-dust, lime and other bases) is required, the refractoriness of the brick is of secondary importance to the closeness of texture. Under such conditions, a porous, highly refractory brick will be far less durable than a dense brick of much lower fusing point. The fluxes act less rapidly on a comparatively smooth, dense surface than on a more porous one, so that the 'life' of the former brick will be longer than the latter, notwithstanding the fact that its actual heat resistance may be lower. For this reason, many fireclays which are of No. 2 quality as regards heat resistance are preferable for use in some furnaces and for flue work generally.

In some heating appliances, the draught and the atmosphere to which the refractory material is exposed have a far greater influence on the durability of the latter than is generally supposed. Unless properly managed, such furnaces may cause firebricks to be destroyed in a few weeks, whereas under proper conditions of draught they would last almost as many years. This is a matter which can only be properly investigated by a specialist in refractory materials.

RAMMED FURNACE LININGS.

Furnaces are sometimes lined with a stiff paste which is rammed into position. The composition of such rammed linings varies with the purpose for which the furnace is used; a well-known material consists of ganister to which a little lime has been added and the mixture then made into a very stiff paste with water.

Various kinds of sand, with or without sufficient clay or other binding agent to make the mass adhesive, are largely used for lining and repairing furnaces. (See also 'Refractory Cements,' p. 376.)

GAS RETORTS.

The chief forms of gas retorts are briefly described in Section XXXIII. They are usually made of a mixture of fireclay and 'grog,' the latter consisting of fireclay which has previously been burned and reduced to a very coarse powder. The grog should not consist of particles of uniform size, but should be graded so as to produce a material capable of withstanding sudden changes in temperature. The proportions of grog and clay must be adapted to the nature of the clay and the size and shape of the retort. The retorts require to be dried very slowly and carefully, and to be burned at a sufficiently high temperature to prevent them shrinking unduly when in use.

A useful test code for horizontal retorts and intermittent vertical chambers is contained in British Standard Specification No. 819—1938.

The following paragraphs contain the salient features of the Specification for Gas Retorts issued by the Institution of Gas Engineers:

Specification for Moulded Gas Retorts.*

(Institution of Gas Engineers, 1934.)

Refractoriness.—Test pieces of the material shall show no signs of fusion when heated to a temperature not less than Seger Cone 28 (about 1630° C.), the heat being increased at a rate of about 50° C. per five minutes in an oxidising atmosphere.

The material is to be chipped to the form and size of a Seger Cone, 1½ ins. high, with a triangular base of ½ in. sides, and tested against standard Seger Cones.

Surface and Texture.—All surfaces shall be reasonably true and free from flaws or winding, and after burning no 'washing' shall be done without the consent of the purchaser. The material shall be evenly burned throughout and of regular texture.

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* Segmental retorts come within the specifications for silica, siliceous or fireclay bricks (p. 372).

Contraction or Expansion.—Test pieces, when heated to a temperature of 1410° C. and maintained at that temperature for two hours, shall not show, when cold, more than 0·75 per cent. linear contraction or expansion.* The test piece shall be 2½–3 ins. by 1½–2 ins. square, the ends being ground flat, and the contraction being measured by means of vernier calipers, reading to 0·1 mm., a suitable mark being made on the test piece, so that the calipers may be placed in the same position before and after firing.

Variation from Dimensions.—Moulded retorts shall be longitudinally straight, true to template in cross-section, of even bore throughout, uniform in thickness, and not warped or twisted. The permissible variation from the straight shall not exceed ⅜ in. in 7 ft. of length or more than ± ¼ in. in any cross-sectional dimension.

Inspection and Testing.—Arrangements for sampling, inspection and testing are specified.

CRUCIBLES.

Crucibles are made of (*i*) fireclay to which a suitable proportion of sand has been added, (*ii*) graphite or plumbago with a sufficient amount of clay or other binder, and (*iii*) various other materials of a special character such as magnesia, zirconia, etc. Crucibles must be sufficiently refractory to withstand the heat to which they are subjected, and whilst at the highest temperature at which they are used they must be sufficiently strong to be lifted out of the furnace, with their contents. Equally important is the resistance offered to the action of their contents, and as no crucible is equally suitable for all purposes careful selection is necessary.

A common test for the ability of a crucible to withstand corrosion is to half fill it with copper and to add a little borax. As the copper melts, part of it will be oxidised, the borax will absorb the oxide and will rapidly corrode the crucible, unless the latter is of excellent quality.

Another useful test consists in melting red lead and copper oxide in the crucible to be tested. Most crucibles will be perforated by the fused mixture after a few minutes' heating, but a satisfactory crucible will stand prolonged heating.

Silica is more readily corroded than clay or grog, but graphite is not affected at all. Hence, for highly corrosive contents, grog or graphite crucibles and not siliceous ones should be used.

Resistance to sudden changes in temperature is an essential property of a good crucible. How necessary is this resistance may be realised from the fact that a crucible may be withdrawn from a furnace at 1800°C., with its contents of 80 lbs. of steel at 1500°C. The steel is at once poured out and replaced by air at a temperature of only 20°C. Yet, notwithstanding the severity of such conditions, crucibles are regularly used for this purpose. On the other hand, many clay crucibles which have been used once and allowed to become cold cannot be used again, as they crack when reheated.

Further information will be found in *Refractory Materials*, by A. B. Searle (Griffin & Co., Ltd., London).

HEAT-INSULATING MATERIALS.†

Attention is being increasingly paid to the saving of fuel by retaining heat in structures from which it ordinarily escapes very rapidly. Thus the loss of steam and of heat in hot gases passing through pipes, cylinders, etc., may be greatly reduced by coating the pipes or cylinders with a highly porous material, such as kieselguhr, light magnesia, felt, cork, or various fibrous materials such as wood-wool, slag-wool and spun glass. The choice of a material must depend partly on the temperatures to which it will be raised when in use; for instance, felt and fibrous vegetable materials should not be raised to more than 220° F., but kieselguhr can be used at a bright red heat.

The prevention of the transference of heat through partitions and furnace-walls is best effected by constructing them of highly porous materials because air is a very poor conductor of heat though its insulating power is much lower at temperatures above 2200° F. With most materials, the larger the number of separate pores per cubic foot, the better will be the insulating power. Where the temperature is relatively low (*e.g.* below 500° F.) the insulating medium may consist of thin slabs or 'containers,' with porous or fibrous material between and blankets made of glass silk and particularly efficient. Highly porous slabs or bricks may also be used.

For higher temperatures it is necessary to use highly porous bricks or blocks made chiefly of kieselguhr or of a highly plastic fireclay or ball clay mixed with a very large proportion of sawdust; the latter burns away during manufacture, leaving a very highly porous mass of great resistance to heat. Such 'insulating bricks' are sold by several firms and are being increasingly used in the construction of kilns and furnaces, in which they effect a notable saving in fuel.

The most effective insulation of kilns and furnaces is in the form of a 'wall' of insulating bricks between the firebrick lining and the exterior. Such bricks should not be placed where their temperature will exceed 2000° F., or they will not long remain effective. They are most efficient at temperatures below 1800° F. They must be protected from damp, which would fill the pores, and so are not suitable for exterior work which is exposed to the weather.

* A tolerance of 0·1 per cent. is allowed.

† See British Standard Specification No. 874—1939 for definitions.

FIRE-RESISTING OR REFRACTORY CEMENTS.

These cements—usually sold under a fancy name—are used for patching furnaces, and other appliances subjected to heat. The precise nature of most of these cements is regarded as a trade secret, but many of them are composed of fireclay, ground firebricks, and Portland cement or water-glass. These materials, when properly selected and mixed in the proper proportions, form a paste which adheres readily to cold brickwork, and when the latter is heated the nature of the cement is changed and a strong heat-resisting material is produced.

The commonest cements for use with various bricks consist of a slip made from the same material as that of the bricks. Thus, a mixture of fireclay and brick-dust (grog) is used with fireclay bricks, whilst a mixture of sand and sodium silicate is often used for bonding silica bricks.

Ground ganister or a mixture of silica rock and fireclay is largely used for lining furnaces.

Special cements are used in certain cases; fused alumina and carborundum cements are sometimes used in retorts and muffles, a small proportion of clay being used as a bonding agent. Powdered zirconia, ground so as to pass through a 200-mesh sieve, is very effective at exceptionally high temperatures, whilst other special mixtures may also be employed where a mixture of unusual composition is needed.

Other refractory cements include:

(*a*) A mixture of fireclay and plumbago made into a paste with water; it may also be used to coat glass to protect it from flame.

(*b*) Equal parts of flour and lime and half a part of fireclay are made into a paste with water; white of egg is sometimes added. This is a fire-resistant cement and not highly refractory. It is sometimes used for patching retorts.

For patching *crucibles and melting pots*:

(*c*) Sift brickdust and mix with equal quantity of red lead; rub together with boiled linseed-oil, which has been mixed with sand to a stiffness of cement.

(*d*) Put freshly slaked lime with concentrated solution of borax, apply with a stiff brush and allow to dry. Upon heating, a fused glaze is obtained.

(*e*) Litharge, 6 parts; fresh burnt lime, 4 parts; china clay, 2 parts; and mix with cold nseed-oil. This is only fire-resisting.

(*f*) Alumina, 1 part; sand, 4 parts; slaked lime, 1 part; borax, ½ part; water to make a stiff paste.

FIRE-RESISTING MATERIALS AND FIRE-PROOFING.

Some substances may be raised to almost any temperature without being melted; they are known as *refractory materials*, and include firebricks of all kinds, crucibles, ladles, retorts, and other appliances as well as the substances from which these are made. Refractory materials and articles differ greatly in their resistance to heat, but all can safely be heated to a temperature of 1500° C., and many of them to 1800° C., or even higher. At about 2000° C. all refractory articles tend to collapse (see p. 371).

Many substances are not sufficiently refractory to stand such intense heating for long periods, but they can endure short periods of intense heat without much damage. Such substances are termed *fire-resistant* and sometimes *fireproof*, though the latter term implies a greater resistance than the former.

The most fire-resistant materials are bricks, tiles, and other articles made of burned clay, stones of various kinds and, to a lesser degree, iron, steel, plaster, and concrete.

Girders, stanchions, and other structures of iron or steel are highly resistant to fire, but they expand so much when heated that they tend to bring about the disruption of structures in which they are used. They must, therefore, be protected by covering them with some material of low thermal conductivity such as blocks or bricks of fireclay, or even with plaster or concrete (see figs. 7–10, p. 306).

Concrete is only moderately resistant to fire. If the aggregate consists of burned clay, trap rock, or blast-furnace clay, the concrete will be more resistant to fire than one with a limestone or silica aggregate. The fire resistance of concrete structures can be increased by surrounding or facing them with expanded metal, or equivalent netting and embedding this in concrete.

Natural Stone varies greatly in its resistance to fire. Some sandstones and limestones crumble readily when heated and then cooled suddenly, as with water.

Artificial stone resembles concrete in its resistance to fire.

Sheet iron is fairly resistant to fire, if free to expand, but under normal conditions it tends to 'buckle,' and may then be useless.

Asbestos, in various forms, has a high fire-resistance, though it is not regarded as a refractory material.

Wood, **when uncovered, is easily set alight, but if coated with a fireproof paint, or with waterglass, followed by a solution of calcium chloride and dried, its resistance is considerably increased. A solution of alum or one of sodium tungstate has a similar effect.**

Painted surfaces **may have their fire resistance increased by the use of suitable paints, but no coating of equivalent thickness to paint can render wood wholly inert to fire. Even the famous ' American Government Whitewash ' will only retard the effect of fire for a short time.**

Definitions of Fire-Resistance of Materials and Structures, and methods of Testing, see British Standard Specification No. 476—1932.

Fire-resisting doors **are classified as shown in Section XXXIV, Part IV.**

Fire-resisting buildings**—see Section XXXIV, Part IV.**

Fire-resisting cements **are described on p. 376.**

Highly Porous Fireclay Bricks.

Highly porous fireclay bricks are used :

(i) as Insulating Bricks (p. 375).

(ii) as permeable bricks through which the products of combustion pass to an annulus or ring-flue and thence to the main flue and chimney. By this means the waste gases act as a heat-insulating medium and much heat is saved. Such bricks are usually known as *hot face insulating bricks*.

SECTION XII

REINFORCED CONCRETE CONSTRUCTION

(pp. 381-418.)

(Revised by J. D. W. Ball, A.M.I.C.E.)

SECTION XII

REINFORCED CONCRETE CONSTRUCTION.

(Revised by J. D. W. Ball, A.M.I.C.E.)

LONDON COUNTY COUNCIL.

By-laws for the construction and conversion of buildings, made by the London County Council in pursuance of the London Building Act (Amendment) Act, 1935.

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. . . .

PART I—LOADING.

2. Every part of a building shall be so constructed as to be capable of safely sustaining and transmitting all the dead and superimposed loading thereon without exceeding the appropriate limitations of permissible stresses provided in these by-laws.

3. For the purpose of calculating dead loading the weights of materials shall be taken to be as set forth in the British Standard Specification (Schedule of Unit Weights of Building Materials) No. 648—1935 unless otherwise agreed with the district surveyor.

4. (a)—*Schedule of loading*—The minimum superimposed load on each floor and on the roof shall be estimated as equivalent to the following dead loads :—

Class No.	Type of Building or Floor.	Slabs : lb. per sq. ft. of Floor Area.	Beams : lb. per sq. ft. of Floor Area.
1	Rooms used for residential purposes ; and corridors, stairs and landings within the curtilage of a flat or residence.	50	40
2	Offices, floors above entrance floors . . .	80	50
3	Offices, entrance floor and floors below entrance floor ; retail shops ; and garages for private cars of not more than two and one quarter tons net weight.	80	80
4	Corridors, stairs and landings not provided for in class 1.	Loading to be provided for to be ascertained to the satisfaction of the district surveyor, but not less than :— 100	100
5	Workshops and factories ; and garages for motor vehicles other than private cars of not more than two and one-quarter tons net weight.	Loading to be provided for to be ascertained to the satisfaction of the district surveyor, but not less than :— 150	120
6	Warehouses, book stores, stationery stores and the like.	Loading to be provided for to be ascertained to the satisfaction of the district surveyor, but not less than :— 200	200
7	Any purpose not herein specified . . .	Loading to be provided for to be ascertained to the satisfaction of the district surveyor.	

Beams and ribs not spaced further apart than two feet six inches between centres shall be designed for slab loads.

Class No.	Roofs.	Slabs: lb. per sq. ft. of Covered Area.	Beams: lb. per sq. ft. of Covered Area.
8	Flat-roofs and roofs inclined at an angle with the horizontal of not more than twenty degrees.	50	30

Subject to the provision of paragraph (*d*) of this by-law, all columns, piers, walls, foundations, and other supports to beams shall be calculated for the superimposed loads tabulated above in this paragraph for beams.

On roofs inclined at an angle with the horizontal of more than twenty degrees a minimum superimposed load (deemed to include the wind load) of fifteen pounds per square foot of surface shall be assumed acting normal to the surface inwards on the windward side, and ten pounds per square foot of surface acting separately and not simultaneously outwards on the leeward side. This requirement shall apply only in the design of the roof construction, and a vertical superimposed load of ten pounds per square foot of covered area shall be substituted for it in estimating the vertical superimposed roof load upon all other parts of the construction.

(*b*) In all cases of floors where the positions of partitions are not definitely located in the design, a uniformly distributed load sufficient to allow for them shall be added to the dead floor load. For all floors of rooms used for offices the minimum total allowance for internal partitions shall be at the rate of twenty pounds per square foot of floor area.

(*c*) Slabs and beams shall be capable of carrying in accordance with these by-laws the following superimposed loads in any position on an otherwise unloaded floor:–

Class of Floor.	Minimum Superimposed Load.	
	Slabs.	Beams.
Floors tabulated under class 1.	¼ ton uniformly distributed per foot width	1 ton uniformly distributed.
All floors tabulated under classes 2 to 6 except garage floors tabulated under class 5.	¾ ton uniformly distributed per foot width	2 tons uniformly distributed.
Garage floors tabulated under class 5.	1·5 × maximum possible combination of wheel loads, but each wheel load not less than 1 ton.	

Provided that beams and ribs spaced not further apart than two feet six inches between centres shall be calculated for the slab loads tabulated in this paragraph; provided also that non-load bearing beams such as beams solely employed as ties to columns shall be exempt from any load calculation under this paragraph.

The reactions due to the superimposed loads tabulated in this paragraph need not be allowed for in calculating the loads on columns, piers, walls or foundations.

(*d*) For the purpose of calculating the total load to be carried on columns, piers, walls and foundations in buildings of more than two storeys in height, and in which the loads and stresses are transmitted through each storey to the foundations:

(i) wholly by a skeleton framework of structural steel; *or*

(ii) partly by a skeleton framework of structural steel and partly by a party wall or party walls; *or*

(iii) wholly by a skeleton framework of reinforced concrete; *or*

(iv) partly by a skeleton framework of reinforced concrete and partly by a party wall or party walls:

the superimposed loads for the roof and topmost storey shall be calculated in full in accordance with the schedule of loading in paragraph (a) of this by-law, but for the lower storeys a reduction of the superimposed loads may be allowed in accordance with the following table :—

Next storey below topmost storey .	10 per cent. reduction of its superimposed load.
Next storey below	20 " " " "
Next storey below	30 " " " "
Next storey below	40 " " " "
Each succeeding storey below . .	50 " " " "

The above reduction may be made by estimating the proportion of floor area carried by each foundation, column, pier and wall. No such reductions shall be allowed on any floor tabulated in this by-law for a superimposed beam loading exceeding one hundred pounds per square foot.

(e) In any case where the superimposed load on any floor or roof is to exceed that hereinbefore specified for the floor or roof, such greater load shall be provided for in accordance with by-law 2.

In the case of any floor intended to be used for a purpose for which a superimposed load is not specified herein, the superimposed load to be carried on that floor shall be provided for in accordance with by-law 2.

(f) In cases where a superimposed load may move, proper provision in accordance with these by-laws to the satisfaction of the district surveyor shall be made for all effects of such movement, including vibration, impact, acceleration and deceleration.

5. In every storey the floor of which is constructed for superimposed loading exceeding one hundred pounds per square foot there shall be exhibited by the owner permanently in a conspicuous position in every room a notice in the following form stating the superimposed loading for which the floor has been constructed.

'LONDON BUILDING ACT (AMENDMENT) ACT, 1935.
NOTICE.'

The floor of this room is constructed for superimposed loading to an intensity not exceeding pounds on any square foot of its surface.'

6. A building shall be so constructed as to resist a wind pressure in each horizontal direction of not less than fifteen pounds per square foot on the upper two-thirds of its surface up to the general roof level or ridge which is or which may be exposed to wind pressure and an additional pressure in each horizontal direction of not less than ten pounds per square foot upon all projections above the general roof level or ridge. If the height of a building be less than twice the width (measured in a direction parallel with that of the wind pressure) of the base upon which the building depends for its resistance to the overturning action of the wind pressure in that direction and provided the district surveyor be satisfied that all loading due to wind pressure will be transmitted safely to the earth, the above wind pressure need not be calculated for the building as a whole; but provision shall be made in accordance with by-law 2 for all the local loading due to wind pressure.

7. Where loading is transmitted through plain concrete, brickwork or other material of similar consistency, the angle of dispersion of such loading through such material shall be taken as not more than forty-five degrees with the direction of such loading.

8. A building or any part thereof shall not be subjected to loading beyond its proper load-bearing capacity at any time when such loading is applied. This provision shall not apply to any loading which may be required by the district surveyor for the purpose of testing.

PART II.—MATERIALS OF CONSTRUCTION.

9. The following provisions shall apply to the aggregates for reinforced concrete :—

Aggregate shall be sand and gravel. or crushed natural stone. It shall be hard strong and durable and shall be reasonably clean and free from clay, organic matter, coal and coal residues (including clinker, ashes, coke-breeze, pan-breeze, slag and other similar material), copper slag, forge breeze, dross (and other similar material), soluble sulphates (including gypsum and other similar material), porous material and other materials liable to reduce the strength or durability of the concrete, or to attack the steel reinforcement.

Fine aggregate shall be of such a size that it will pass through a $\frac{3}{16}$-inch mesh. Not more than five per cent. by weight shall pass a No. 100 mesh.

Coarse aggregate shall be of such a size that it will be retained on a $\frac{3}{16}$-inch mesh and will pass a mesh of a size one-quarter of an inch less than the minimum lateral distance between the reinforcing bars.

Aggregate shall be so graded between the limits as to make a dense concrete of the specified proportions and consistence that will work readily into position without segregation and without the use of an excessive water content.

10. Aggregate for plain concrete shall consist of such materials as are specified in by-law 9 or of hard well-burned brick, hard well-burned tile, pumice or other material which the district surveyor may approve as of like suitability and shall be so graded and contain sand in such proportion as to produce a dense concrete.

11. Sand shall be clean and shall be composed of hard silicious grains reasonably free from clay or any animal, vegetable or bituminous matter. The grains shall be of such a size as to pass through a $\frac{5}{16}$-inch mesh.

12. Cement shall be Portland cement, Portland blast furnace cement or high alumina cement, but no two of such cements shall be used in combination.

Portland cement shall comply with the British Standard Specification for Portland Cement numbered 12—1931.

Portland blast-furnace cement shall comply with the British Standard Specification for Portland Blast-furnace Cement numbered 146—1932.

High alumina cement shall consist of aluminous and calcareous materials which have been fused to a molten state and ground to such a degree of fineness that the cement will not leave a residue of more than 12 per cent. by weight on a No. 170 mesh and not more than 1 per cent. on a No. 72 mesh. The cement shall contain not less than 35 per cent. by weight of alumina and the ratio of percentage by weight of alumina to that of lime shall not be less than 0·9. When gauged with 22 per cent. by weight of water it shall not begin to set before the expiration of two hours but shall begin to set within 6 hours of gauging and the final setting shall take place within 2 hours of the initial setting. The strength of high alumina cement shall be such that when a mortar is composed of one part by weight of high alumina cement to three parts by weight of white Leighton Buzzard sand graded to pass a No. 18 mesh and stop on a No. 25 mesh, and the whole is gauged with a weight of water equal to 8 per cent. of the dry materials, such mortar shall have a tensile strength of not less than 475 lb. per square inch within 24 hours after gauging and within seven days its tensile strength shall have increased and shall be not less than 550 lb. per square inch.

Wherever cement is used it shall not be moved or disturbed after one hour from the time it has come into contact with water until it has set hard.

13. Water used shall be clean and free from deleterious matter.

14. Concrete shall consist of aggregate mixed with cement. The proportions of fine aggregate to coarse aggregate to cement shall be as set out in Tables I and II or in any intermediate proportions in which the volume of coarse aggregate is twice that of the fine aggregate. Provided that in any particular case where specially authorised by the district surveyor the proportion of coarse aggregate may be varied within the limits of one-and-a-half and two-and-a-half times the fine aggregate. If in any particular case the district surveyor so requires, the proportion of coarse aggregate shall be varied within the aforesaid limits. Where the proportion of coarse aggregate to fine aggregate is so varied the requirements of this by-law shall be based on the ratio of the sum of the volumes of fine and coarse aggregates, each measured separately, to the quantity of cement and shall be obtained by proportion to the two nearest specified mixes. If the district surveyor so requires, the builder shall make tests in accordance with Schedules I and III of this Part of these by-laws as may be necessary to prove the quality of the concrete. The grades of concrete designated I, II and III in column I of Table I and mixed in the proportions set out in Column 2 of such Table against each such designation shall be deemed to be " ordinary concrete " and shall, within the period of twenty-eight days, possess the respective minimum resistance to crushing set out in Column 3 of such table against each such designation. Where intermediate proportions of cement to aggregate are used, as heretofore provided, the minimum crushing strength shall be in proportion to the prescribed minima crushing strengths of the two nearest specified mixes.

TABLE I.

(1)	(2)		(3)
Designation of Concrete.	Cubic Feet Aggregate per 112 lbs. of Cement.		Minimum Resistance to Crushing in lbs. per square inch within 28 Days after Mixing.
	Fine Aggregate.	Coarse Aggregate.	
I	1¼	2½	2,925
II	1⅞	3¾	2,550
III	2½	5	2,250
IV	7½		1,480
V	10		1,100
VI	12½		740
VII	15		370

The grades of concrete designated IA, IIA and IIIA in Table II shall be deemed to be ' Quality A concrete ' and when such concrete is to be used, if the district surveyor so requires, preliminary tests shall be made in accordance with Schedule II of this Part of these by-laws before the commencement of the work and subsequently whenever any change is to be made in the materials or in the proportions of the materials to be used. Works tests shall be made in accordance with Schedules I and III of this Part of these by-laws as and when the district surveyor shall require to prove the quality of the concrete. A record of such tests identifying them with the part of the work executed shall be kept by the builder on the works.

The grades of concrete designated IA, IIA and IIIA in column 1 of Table II and mixed in the proportions set out in column 2 of such Table against such designation shall, within the period of twenty-eight days, possess the respective minimum resistance to crushing set out in column 3 of such Table against each such designation. Provided that such concrete may possess a lesser strength but (not less than that required for ordinary concrete in Table I) subject to the appropriate maximum permissible stresses in such concrete being proportionately less than the maximum stresses specified in by-laws 99 and 101 of these by-laws.

TABLE II.

1	2		3	
Designation of Concrete.	Cubic Feet of Aggregate per 112 lbs. of Cement.		Minimum Resistance to Crushing in lbs. per square inch within 28 Days after Mixing.	
	Fine Aggregate.	Coarse Aggregate.	Preliminary Test.	Works Test.
IA	$1\frac{1}{4}$	$2\frac{1}{2}$	5,625	3,750
IIA	$1\frac{7}{8}$	$3\frac{3}{4}$	4,950	3,300
IIIA	$2\frac{1}{2}$	5	4,275	2,850

Concrete containing more than 10 cubic feet of aggregate per 112 lbs. of cement, or having resistance to crushing less than 1,110 lbs. per square inch, shall not be used in the construction of a building or any part thereof.

Concrete containing more than 15 cubic feet of aggregate per 112 lbs. of cement, or having resistance to crushing less than 370 lbs. per square inch shall not be used for any purpose in connection with the construction of a building or chimney shaft.

The quantity of water for making reinforced concrete shall be sufficient but not more than sufficient to ensure that the concrete shall surround, cover, embed and grip adequately all the reinforcement.

The quantity of water for making Portland cement concrete and Portland blast furnace cement concrete shall be sufficient but not more than sufficient to bring the entire mixture to a uniform colour and to ensure that the concrete shall be suitable for its intended purpose.

The quantity of water for making high alumina cement concrete shall be sufficient but not more than sufficient to produce a sound concrete and the concrete shall be kept wet for 24 hours after gauging.

Material shall be so mixed as to secure uniformity throughout the mixture.

The concrete shall be deposited without segregation of the materials and shall be properly consolidated by punning, rodding, vibrating or other means after depositing and before the cement begins to set.

After such consolidation, the concrete shall remain undisturbed and shall be protected from frost, heat, running water, evaporation, vibration or any other cause which may reduce its strength or tend to form voids in it.

During mixing, depositing and setting, the temperature of concrete shall not fall below 40 degrees Fahrenheit.

Where formwork is employed it shall be sufficiently rigid to retain the concrete in position and shape during depositing, punning and consolidating.

. . . .

25. The district surveyor may for the purpose of due supervision by notice require the builder or other person causing or directing the work, to furnish him with proof by means of adequate tests or otherwise that the materials used or to be used conform to the requirements of these by-laws, and no material shall be so used unless such material conforms to such requirements.

SCHEDULE I.

Standard method of test for consistence of concrete.

The test is to be used both in the laboratory and during the progress of the work for determining the consistence of concrete.

The test specimen shall be formed in a mould in the form of the frustrum of a cone with internal dimensions as follows:—Bottom diameter, 8 inches; top diameter, 4 inches, and height, 12 inches. The bottom and the top shall be open, parallel to each other, and at right angles to the axis of the cone. The mould shall be provided with suitable foot pieces and handles. The internal surface shall be smooth.

Care shall be taken to ensure that a representative sample is taken.

The internal surface of the mould shall be thoroughly clean, dry and free from set cement before commencing the test.

The mould shall be placed on a smooth, flat, non-absorbent surface, and the operator shall hold the mould firmly in place, while it is being filled, by standing on the foot pieces. The mould shall be filled to about one-fourth of its height with the concrete which shall then be puddled, using 25 strokes of a ⅝ inch rod, 2 feet long, bullet pointed at the lower end. The filling shall be completed in successive layers similar to the first and the top struck off so that the mould is exactly filled. The mould shall then be removed by raising vertically, immediately after filling. The moulded concrete shall then be allowed to subside and the height of the specimen measured after coming to rest.

The consistence shall be recorded in terms of inches of subsidence of the specimen during the test, which shall be known as the slump.

SCHEDULE II.

Standard method of making preliminary cube tests of concrete.

The method described applies to compression tests of concrete made in a laboratory where accurate control of materials and test conditions is possible.

Materials and proportioning.—The materials and the proportions used in making the preliminary tests shall be similar in all respects to those to be employed in the work. The water content shall be as nearly as practicable equal to that to be used in the work, but shall be not less than the sum of 30 per cent. by weight of the cement and 5 per cent. by weight of the aggregate unless specially authorised by the district surveyor. For porous aggregates additional water shall be used to allow for the amount absorbed by the aggregates.

Materials shall be brought to room temperature (58° to 64° F.) before beginning the tests. The cement on arrival at the laboratory shall be mixed dry either by hand or in a suitable mixer in such a manner as to ensure as uniform a material as possible, care being taken to avoid intrusion of foreign matter. The cement shall then be stored in air-tight containers until required. Aggregates shall be in a dry condition when used in concrete tests.

The quantities of cement, aggregate and water for each batch shall be determined by weight to an accuracy of 1 part in 1,000.

Mixing concrete.—The concrete shall be mixed by hand or in a small batch mixer in such a manner as to avoid loss of water. The cement and fine aggregate shall first be mixed dry until the mixture is uniform in colour. The coarse aggregate shall then be added and mixed with the cement and sand. The water shall then be added and the whole mixed thoroughly for a period of not less than two minutes and until the resulting concrete is uniform in appearance.

Consistence.—The consistence of each batch of concrete shall be measured, immediately after mixing, by the slump test made in accordance with the method of test for consistence of concrete given in schedule I. Provided that care is taken to ensure that no water is lost the material used for the slump tests may be re-mixed with the remainder of the mix before making the test specimen.

Size of test cubes.—Compression tests of concrete shall be made on 6-inch cubes. The mould shall be of metal with inner faces accurately machined in order that the opposite sides of the specimen shall be plane and parallel. Each mould shall be provided with a metal base, having a smooth machined surface. The interior surfaces of the mould and base shall be slightly oiled before concrete is placed in the mould.

Compacting.—Concrete test cubes shall be moulded by placing the fresh concrete in the mould in three layers, each layer being rammed with a steel bar 15 inches long and having a ramming face of 1 inch square and a weight of 4 lbs. For mixes of 1½ inches slump or less, 35 strokes of the bar shall be given for each layer; for mixes of wetter consistence the number may be reduced to 25 strokes per layer.

Curing.—All test cubes shall be placed in moist air of at least 90 per cent. relative humidity and at a temperature of 58° F. to 64° F. for 24 hours (± ½ hour) commencing immediately after moulding is completed. After 24 hours the test cubes shall be marked, removed from the moulds, and placed in water at a temperature of 58° to 64° F. until required for test.

Method of testing.—All compression tests on concrete cubes shall be made between smooth plane steel plates, without end packing, the rate of loading being kept approximately at 2,000 lbs.

per square inch per minute. One compression plate of the machine shall be provided with a ball seating in the form of a portion of a sphere, the centre of which falls at the central point of the face of the plate.

All test cubes shall be placed in the machine in such a manner that the load shall be applied to the sides of the cubes as cast.

Distribution of specimens and standard of acceptance.—For each age at which tests are required, three cubes shall be made and each of these shall be taken from a different batch of concrete.

The acceptance limits are a difference of 15 per cent. of the average strength between the maximum and minimum recorded strengths of the three cubes. In cases where this is exceeded repeat tests shall be made, excepting where the minimum strength test result does not fall below the strength specified.

SCHEDULE III.

Standard method of making works cube tests of concrete.

The method described applies to compression tests of concrete sampled during the progress of the work.

Size of test cubes and moulds.—The test specimens shall be 6-inch cubes. The moulds shall be of metal, with inner faces accurately machined in order that opposite sides of the specimen shall be plane and parallel. Each mould shall be provided with a metal base plate, having a smooth machined surface. The interior surfaces of the mould and base shall be slightly oiled before concrete is placed in the mould.

Sampling.—Wherever practicable concrete for the test cubes shall be taken immediately after it has been deposited in the work. Where this is impracticable samples shall be taken as the concrete is being delivered at the point of deposit, care being taken to obtain a representative sample. All the concrete for each sample shall be taken from one place. A sufficient number of samples, each large enough to make one test cube, shall be taken at different points so that the test cubes made from them will be representative of the concrete placed in that portion of the structure selected for tests. The location from which each sample is taken shall be noted clearly for future reference.

In securing samples the concrete shall be taken from the mass by a shovel or similar implement, and placed in a large clean pail or other receptacle, for transporting to the place of moulding. Care shall be taken to see that each test cube represents the total mixture of concrete from a given place. Different samples shall not be mixed together, but each sample shall make one cube. The receptacle containing the concrete shall be taken to the place where the cube is to be moulded as quickly as possible and the concrete shall be slightly re-mixed before placing in the mould.

Consistence.—The consistency of each sample of concrete shall be measured, immediately after re-mixing, by the slump test made in accordance with the method of test for consistence of concrete given in schedule I.

Providing that care is taken to ensure that no water is lost the material used for the slump tests may be re-mixed with the remainder of the mix before making the test cube.

Compacting.—Concrete test cubes shall be moulded by placing the fresh concrete in the mould in three layers, each layer being rammed with a steel bar 15 inches long and having a ramming face of 1 inch square and a weight of 4 lbs. For mixes of 1½ inches slump or less, 35 strokes of the bar shall be given for each layer; for mixes of wetter consistence the number may be reduced to 25 strokes per layer.

Curing.—The test cubes shall be stored at the site of construction at a place free from vibration, under damp sacks for 24 hours ($\pm$ ½ hour), after which time they shall be removed from their moulds, marked and buried in damp sand until the time for sending to the testing laboratory. They shall then be well packed in damp sand or other suitable damp material and sent to the testing laboratory, where they shall be similarly stored until the date of test. Test cubes shall be kept on the site for as long as practicable and for at least three-fourths of the period before test, except for tests at ages less than seven days.

The temperature of the place of storage on the site shall not be allowed to fall below 40° F., nor shall the cubes themselves be artificially heated.

Record of temperatures.—A record of the maximum and minimum day and night temperatures at the place of storage of the cubes shall be kept during the period the cubes remain on the site.

Method of testing.—All compression tests on concrete cubes shall be made between smooth plane steel plates without end packing, the rate of loading being kept approximately at 2,000 lbs per square inch per minute. One compression plate of the machine shall be provided with a ball seating in the form of a portion of a sphere, the centre of which falls at the central point of the face of the plate.

All test cubes shall be placed in the machine in such a manner that the load shall be applied to the sides of the cubes as cast.

Part IV.—Walls and Piers.

Section 1—General Requirements.

39. Every building shall be enclosed with walls. Provided that openings may be made in such walls subject to the following conditions—(1) That the total elevational area of openings in any such wall above the soffit of the first floor do not exceed one-half the elevational area of such wall measured from the soffit of the first floor of the building to the roof; (2) that the total elevational area of openings in any storey-height of such wall above the soffit of the first floor of the building do not exceed two-thirds of the total area of such wall within such storey-height; (3) that the total width of openings at any level above the soffit of the first floor do not exceed three-quarters of the total length of the wall at that level. For the purposes of this by-law, the expression 'walls' shall be deemed to include piers and for the purpose of this by-law and of by-laws 43 and 51 (*g*) any glazing or glass in the thickness of such walls shall be deemed to be an opening.

. . . .

43. . . . No reinforced concrete external wall nor reinforced concrete part or panel of an external wall shall be of less thickness in any part than four inches exclusive of plastering, rendering, rough cast or other applied covering. No reinforced concrete party wall shall be of less thickness in any part than eight inches.

. . . .

Part VI—The use of Reinforced Concrete.

92. Concrete shall comply with these by-laws and shall not be relied upon to support, collect, or transmit loading otherwise than as provided in these by-laws.

Reinforced concrete shall be, as regards composition and quality, not inferior to that designated III in by-law 14.

93. Construction which will support or transmit loading supported, collected or transmitted by reinforced concrete shall comply with the requirements of these by-laws.

94. Loading supported, collected or transmitted by reinforced concrete shall be distributed upon the earth by concrete which shall:—

(*a*) comply with the requirements of by-law 32 in the same manner as is required for concrete which is to support walls or piers;

(*b*) if plain, be of composition and quality not inferior to that designated V in by-law 14. The angle of dispersion through such plain concrete shall be taken as not less than forty-five degrees with the horizontal; and such plain concrete shall not be relied upon to resist shearing or tensile stresses otherwise than in accordance with this by-law; and

(*c*) if reinforced, comply with the requirements of these by-laws.

The pressure upon such distributing concrete shall be calculated, and such concrete, if plain, shall comply in all respects with the requirements of by-law 35 in the same manner as is required for plain concrete which is to support walls or piers.

95. Where metal is used in combination with concrete which supports, collects or transmits loading in a building or part thereof, or which distributes such loading upon the earth, proper protection shall be provided to prevent such damage to the metal as would, in the opinion of the district surveyor, affect adversely the stability of such building or of any part thereof.

96. Reinforcement shall be of structural steel complying with these by-laws so combined with the concrete that the reinforcement will be sufficient to provide, in accordance with these by-laws, all necessary:—

(*a*) resistance to tension;

(*b*) assistance for the concrete to resist shearing actions; and

(*c*) assistance for the concrete to resist compression.

Reinforcement shall, immediately before being placed in the concrete, be free from loose mill scale, loose rust, oil or other matter which might affect adversely the proper combination of such reinforcement with such concrete.

97. Reinforcement shall have concrete cover, and the thickness of such cover (exclusive of plaster or other decorative finish) shall be:—

(*a*) for each end of a reinforcement rod or bar which is anchored otherwise than by means of a hook, not less than 2 inches, nor less than twice the diameter of such rod or bar beyond such anchorage;

(*b*) for a longitudinal reinforcement rod or bar in a column, not less than 1½ inches, nor less than the diameter of such rod or bar;

(*c*) for a longitudinal reinforcement rod or bar in a beam, not less than 1 inch, nor less than the diameter of such rod or bar;

(*d*) for tensile, compressive, shear or other reinforcement in a slab, not less than half an inch, nor less than the diameter of such reinforcement;

(*e*) for any other reinforcement (not being a binding), not less than half an inch, nor less than the diameter of such reinforcement.

98. The following by-laws 99 to 112 inclusive shall relate only to the use of reinforced concrete in a building wherein the loads and stresses are transmitted through each storey to the foundations wholly by a skeleton framework of reinforced concrete or partly by a skeleton framework of reinforced concrete and partly by a party wall or party walls.

99. The compressive, shearing and bond stresses in reinforced concrete shall be calculated, and, subject to the requirements of by-law 101, such stresses shall not exceed those shown as appropriate for each designation of concrete in Tables IX and X following:—

TABLE IX—*Ordinary Concrete.*

Designation of Concrete.	Modular Ratio.	Permissible Concrete Stresses. Lbs. per square inch.			
		Compression.		Shear.	Bond.
		Due to Bending.	Direct.		
I	15	975	780	98	123
II	15	850	680	85	110
III	15	750	600	75	100

Where other proportions of fine to coarse aggregate are used the permissible concrete stresses shall be based on the ratio of the sum of the volumes of the fine and coarse aggregates, each measured separately to the quantity of cement, and shall be obtained by proportion from the two nearest designations.

TABLE X—*Quality A Concrete.*

Designation of Concrete.	Modular Ratio.	Permissible Concrete Stresses. Lbs. per square inch.			
		Compression.		Shear.	Bond.
		Due to Bending.	Direct.		
IA	15	1,250	1,000	125	150
IIA	15	1,100	880	110	135
IIIA	15	950	760	95	120

Where other proportions of fine to coarse aggregate are used the permissible concrete stresses shall be based on the ratio of the sum of the volumes of the fine and coarse aggregates, each measured separately, to the quantity of cement, and shall be obtained by proportion from the two nearest designations.

Provided that where reinforcement in the form of plain bars is used to resist tensile stresses induced by bending action the calculated bond stress due to a variation in tensile stress shall not exceed twice that shown as appropriate for each designation of concrete in tables IX and X above in this by-law.

Provided also that the 'punching shear' stress in a footing (or similar construction) of reinforced concrete which complies with the requirements of these by-laws shall be not more than twice the permissible shearing stress shown as appropriate for each designation of concrete in tables IX and X above in this by-law.

100. The tensile and compressive stresses in steel reinforcement of reinforced concrete shall be calculated, and, subject to the requirements of by-law 101, such stresses shall not exceed those shown as appropriate for each designation of stress in Table XI following.

TABLE XI.

Designation of Stress in Steel Reinforcement.	Maximum Permissible Stress, in Pounds per Square Inch.
Tension in helical reinforcement in a column . . .	13,500
Tension other than in helical reinforcement in a column .	18,000
Longitudinal compression in a beam where the compressive resistance of the concrete is not taken into account.	18,000
Longitudinal compression, direct or due to bending where the compressive resistance of the concrete is taken into account.	The calculated compressive stress in the surrounding concrete multiplied by the modular ratio.

101. The maximum permissible stresses in a reinforced concrete column or part thereof having a ratio of effective column length to least radius of gyration not exceeding 50 shall be as specified in by-laws 99 and 100.

The maximum permissible stresses in a reinforced concrete column or part thereof having a ratio of effective column length to least radius of gyration between 50 and 120 as shown in Table XII following shall not exceed those which result from the multiplication of the appropriate maximum permissible stresses specified in by-laws 99 and 100 by the coefficient shown as appropriate for each ratio of effective column length to least radius of gyration in Table XII following :—

TABLE XII.

Ratio of Effective Column Length to Least Radius of Gyration.	Coefficient.
50	1·0
60	0·9
70	0·8
80	0·7
90	0·6
100	0·5
110	0·4
120	0·3

For any ratio of effective column length to least radius of gyration between 50 and 120 not shown in Table XII above, the appropriate coefficient shall be determined on the basis that the coefficient varies in proportion with the ratio of effective column length to least radius of gyration between those shown as consecutive in Table XII above.

A reinforced concrete column or part thereof shall not have a ratio of effective column length to least radius of gyration more than 120.

102. The effective column length to be assumed in determining the working load per square inch in accordance with by-laws 99, 100 and 101 shall be as follows :—

	Type of Column.	Effective Column Length.
Columns of one storey.	Properly restrained at both ends in position and direction.	0·75 of the actual column length.
	Properly restrained at both ends in position but not in direction.	Actual column length.
	Properly restrained at one end in position and direction and imperfectly restrained in both position and direction at the other end.	A value intermediate between the actual column length and twice that length depending upon the efficiency of the imperfect restraint.

	Type of Column.	Effective Column Length.
Columns continuing through two or more storeys.	Properly restrained at both ends in position and direction.	0·75 of the distance from floor level to floor level.
	Properly restrained at both ends in position and imperfectly restrained in direction at one or both ends.	A value intermediate between 0·75 and 1·00 of the distance from floor level to floor (or roof) level, depending upon the efficiency of the directional restraint.
	Properly restrained at one end in position and direction and imperfectly restrained in both position and direction at the other end.	A value intermediate between the distance from floor level to floor (or roof) level and twice that distance, depending upon the efficiency of the imperfect restraint.

The effective column length values given above are in respect of typical cases only and embody the general principles which shall be employed in assessing, to the satisfaction of the district surveyor, the appropriate value for any particular column.

103. The maximum permissible stresses in reinforced concrete and in its reinforcement may exceed these specified in by-laws 99 and 100 respectively by not more than 33⅓ per cent., provided such excess is solely due to stresses induced by wind loading, and provided that such excess shall not apply to secondary floor beams, nor to the stresses in roof construction above the topmost floor level in a building.

104. A reinforced concrete column shall have longitudinal steel reinforcement, and the cross-sectional area of such reinforcement shall not be less than 0·8 per cent., nor more than 8 per cent., of the gross cross-sectional area of the column required to transmit all the loading in accordance with these by-laws.

A reinforced concrete column having helical reinforcement shall have also at least six bars of longitudinal reinforcement within such helical reinforcement. Such longitudinal bars shall be in contact with such helical reinforcement and equidistant around its inner circumference.

At a splice in a longitudinal reinforcement, the spliced bars shall overlap longitudinally through a distance not less than 24 times the diameter of the upper bar, or a sufficient distance to develop the force in the bar by bond, whichever is the lesser.

105. A reinforced concrete column shall have transverse or helical reinforcement so disposed as to provide all necessary restraint against buckling of the longitudinal reinforcement; and the ends of such transverse reinforcement shall be anchored properly.

The diameter of such transverse reinforcement shall not be less than one-fourth of an inch.

The pitch of such transverse reinforcement shall not be more than the least of the three following distances:—

(1) the least lateral dimension of such column;

(2) twelve times the diameter of the smallest longitudinal reinforcement in such column;

(3) 12 inches.

Helical reinforcement shall be of regular formation, with the turns of the helix spaced evenly; and its ends shall be anchored properly. The pitch of the helical turns shall be not more than 3 inches nor more than one-sixth of the core-diameter of such column; and such pitch shall be not less than 1 inch nor less than three times the diameter of the steel forming such helix.

106. The diameter of a steel reinforcement in reinforced concrete shall be not more than 2 inches.

The diameter of a longitudinal steel reinforcement in a reinforced concrete column shall be not less than ½ inch.

The diameter of a main steel reinforcement in a reinforced concrete beam or slab shall be not less than ¼ inch.

The diameter of a steel reinforcement in reinforced concrete other than a longitudinal reinforcement in a column or a main reinforcement in a beam or slab, and the diameter of steel forming a tie, helix, stirrup or the like, shall be not less than $\frac{3}{16}$ inch.

The diameter of steel forming a mesh-reinforcement for the purpose of resisting tension in reinforced concrete shall be not less than $\frac{1}{8}$ inch.

107. The distance between two steel reinforcements in reinforced concrete shall be not less than the greatest of the three following distances :—

(a) the diameter of either bar if their diameters are equal ;

(b) the diameter of the larger bar if their diameters be unequal ;

(c) ¼ inch more than the greatest size of the coarse aggregate comprised in such concrete.

Provided that the vertical distance between two horizontal main steel reinforcements, or the corresponding distance at right angles to two inclined main steel reinforcements, shall be not less than ½ inch, except at a splice and except where one of such reinforcements is transverse to the other.

The pitch of distributing bars in a reinforced concrete solid slab shall be not more than four times the effective depth of such slab.

A mesh-reinforcement shall be of such dimensions, shape, proportions and arrangement as will afford proper combination of such reinforcement with such concrete.

108. Where the concrete alone is not sufficient to resist, in accordance with by-law 99 the shearing action in reinforced concrete, the whole of such shearing action shall be provided for by the tensile resistance of shear reinforcement acting in proper conjunction with the compressive resistance of the concrete ; but the magnitude of such shearing action to be so provided for shall not exceed *four* times that which the concrete alone could resist in accordance with by-law 99.

109. A stirrup in reinforced concrete shall pass round, or be secured adequately otherwise to, the appropriate tensile reinforcement ; and such stirrup shall have both its ends anchored properly.

110. A reinforced concrete solid slab spanning in one direction shall have distributing bars at right angles to the main tensile reinforcement of such slab ; and the aggregate cross-sectional area of such distributing bars shall not be less than one-tenth of the aggregate cross-sectional area of such main tensile reinforcement associated therewith.

111. Where the compressive resistance of concrete in beams is taken into account, the compression reinforcement where it is required shall be effectively anchored over the distance where it is required at points not further apart centre to centre than twelve times the diameter of the anchored bar.

Where the compressive resistance in concrete is not taken into account the compressive reinforcement shall be effectively anchored laterally and vertically over the distance where it is required at points not further apart centre to centre than eight times the diameter of the anchored bar. The subsidiary reinforcement used for this purpose shall pass round or be hooked over both the compression and tensile reinforcement.

112. Hooks and other anchorages of reinforcement in reinforced concrete shall be of such form, dimensions and arrangement as will ensure their adequacy without overstressing the concrete or any other material.

113. Where reinforced concrete is used in the construction of a building wherein the loads and stresses are not transmitted through each storey to the foundations wholly by a skeleton framework of reinforced concrete nor partly by a skeleton framework of reinforced concrete and partly by a party wall or party walls, the standard of stability shall, to the satisfaction of the district surveyor, be not inferior to that required for compliance with by-laws 99 to 112 inclusive.

114. Reinforcement in reinforced concrete shall not be connected by welding, except in accordance with conditions prescribed by the Council in each particular case.

Provided that round or square bars not more than 0·4 inch diameter and transverse to each other forming a mesh reinforcement in a solid slab may be connected by electrically fusing the metal of such rods at their points of contact if such fusion be executed at the works where such mesh is fabricated and if the district surveyor be satisfied as to its suitability.

115. Reinforced concrete subjected to bending actions in a building shall possess adequate stiffness to prevent such deflection or deformation as might, in the opinion of the district surveyor, affect adversely the stability of such building or of any part thereof.

116. Reinforced concrete subjected to compression in a building shall possess adequate stiffness, or be provided with adequate restraint, to prevent such lateral flexure as might, in the opinion of the district surveyor, affect adversely the stability of such building or of any part thereof.

117. The fabrication and erection of reinforced concrete shall be such as will ensure that the assumptions upon which the stresses in such concrete and in its reinforcement have been calculated shall be fulfilled adequately at all times in the building of which such reinforced concrete forms part.

118. Where the district surveyor finds substantial reason for doubt as to the sufficiency or suitability of the reinforced concrete for its purposes under these by-laws, the builder shall make such test or tests on such concrete as the district surveyor may require ; and if such testing proves, in the opinion of the district surveyor, that such concrete is insufficient or unsuitable for its purposes under these by-laws, such concrete shall be removed and replaced with reinforced concrete which complies with these by-laws.

LONDON COUNTY COUNCIL

London Building Acts 1930 & 1935

Building By-Laws—Computation of Stresses

. . . .

LOADING.

. . . .

3. For ordinary construction, the weight of reinforced concrete may be assumed as 144 lb. per cubic foot.

. . . .

REINFORCED CONCRETE.

5. *Basic assumptions.*—Design of reinforced concrete to resist bending should be based upon the assumptions :—

(i) that both steel and concrete are elastic within the range of the permissible stresses ;
(ii) that all tensile stresses are taken by the reinforcement ; and
(iii) that plane sections remain plane.

Stresses due to shrinkage or expansion of the concrete need not be calculated.

6. *Moment of inertia.*—In the absence of conditions rendering such a course unjustifiable, the moment of inertia may be calculated on :—

(i) the entire concrete section, ignoring the reinforcement ;
or (ii) the entire concrete section, including the reinforcement on the basis of the modular ratio;
or (iii) the compression area of the concrete section, combined with the reinforcement on the basis of the modular ratio.

These methods should not be changed or combined in a design. One method having been adopted, that method should be applied throughout.

. . . .

Beams and Slabs.

8. *Effective span.*—The effective span of a beam or slab should be taken as either :—

(i) the distance between the centres of support ;
or (ii) the clear distance between supports plus the effective depth of the beam or slab.

9. *Lateral stiffness.*—The ratio of length between adequate lateral restraints of a beam to the breadth of its compression flange should not exceed :—

$$20\left\{3 - 2\left(\frac{\text{calculated compressive stress}}{\text{permissible compressive stress}}\right)\right\}$$

10. T-*beams and* L-*beams.*—In T-beams the breadth of the flange assumed as taking compression should not exceed the least of the following :—

(i) one-third of the effective span of the T-beam ;
(ii) the distance between the centres of the ribs of the T-beams ;
(iii) the breadth of the rib plus twelve times the thickness of the slab.

In L-beams, the breadth of the flange assumed as taking compression should not exceed the least of the following :—

(i) one-sixth of the effective span of the L-beam ;
(ii) the breadth of the rib plus one-half of the clear distance between ribs ;
(iii) the breadth of the rib plus four times the thickness of the slab.

When a part of a slab is considered as the flange of a T-beam or L-beam, the reinforcement in the slab transverse to the beam should cross the full width of the flange ; and where the slab is assumed to be spanning independently in the same direction as the beam, such reinforcement should be near the top surface of the slab.

11. Bending moments in beams and slabs should be calculated for the effective span and all loading thereon.

The bending moments to be provided for at a cross-section of a continuous beam or slab should be the maximum positive and negative moments at such cross-section for the following arrangements of superimposed loading :—

(i) alternate spans loaded and all other spans unloaded ;
(ii) any two adjacent spans loaded and all other spans unloaded.

Nevertheless, provided the maximum positive moments so obtained in any two adjacent spans are increased by an amount not exceeding 15 per cent. of the maximum intermediate support moment, the latter may be reduced by the same numerical amount and the positive moments elsewhere in the span increased accordingly.

The computation of bending moments in flat slabs is dealt with later in this memorandum.

12. *Beams and slabs spanning in one direction.*—Provided the fabrication and erection of the reinforced concrete be such as will ensure that the assumptions upon which the stresses in such concrete and in its reinforcement have been calculated will be fulfilled adequately at all times in the building of which such reinforced concrete forms part, the bending moments in beams and slabs spanning in one direction may be calculated on one of the following assumptions:—

(i) Beams may be designed as members of a continuous framework, with monolithic connection between beams and columns, and the bending moments calculated taking into account the resistance of the columns to bending; or

(ii) Beams and slabs may be designed as continuous over supports and capable of free rotation about them; or

(iii) The total bending moments in all cases of uniformly distributed or other symmetrica loading over a number of approximately equal spans may be assumed to have the following relations to those induced in a freely supported span similarly loaded:—

Near Middle of End Span.	At Support next to End Support.	At Middle of Interior Spans.	At Other Interior Supports.
$+\frac{8}{10}$	$-\frac{8}{10}$	$+\frac{8}{12}$	$-\frac{8}{12}$

Two spans may be considered as approximately equal when they do not differ by more than 15 per cent. of the longer.

13. *Slabs spanning in two directions at right angles*—

(i) *General.*—To estimate the bending moments in a solid slab spanning in two directions at right angles, the slab may be assumed to act as perfectly elastic thin plate, Poisson's Ratio being assumed equal to zero.

(ii) *Slabs simply supported on four sides.*—The bending moments at the centre of a rectangular solid slab spanning in two directions at right angles, with loading uniformly distributed, and simply supported on four sides, may be assumed to have the values given by the following equations (1) and (2):—

$$M_x = Z_x\left(\frac{wl_x^2}{8}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$M_y = Z_y\left(\frac{wl_y^2}{8}\right) \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where M_x and M_y are the bending moments on strips of unit width and spans l_x and l_y respectively;

w is the total load per unit area;

l_y is the length of the longer side (see fig. 1);

l_x is the length of the shorter side (see fig. 1);

and Z_x and Z_y are coefficients shown in Table A following.

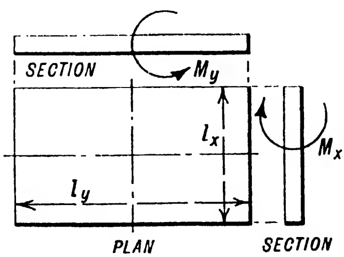


FIG 1.

TABLE A.

l_y/l_x	1·0	1·1	1·2	1·3	1·4	1·5	1·75	2·0	2·5	3·0
Z_x .	0·500	0·594	0·675	0·741	0·794	0·835	0·904	0·941	0·975	0·988
Z_y .	0·500	0·406	0·325	0·259	0·206	0·165	0·096	0·059	0·032	0·022

(iii) *Slabs fixed at or continuous over four sides.*—The negative bending moments at the supports of a rectangular solid slab spanning in two directions at right angles, with loading uniformly distributed, and fixed at or continuous over four sides, may be assumed to have the values given by equations (1) and (2) above, taking the values for Z_x and Z_y shown in Table B following; and the positive bending moments near mid-span may be assumed to have values not less than 80 per cent. of those given by equations (1) and (2) above, taking the values for Z_x and Z_y shown in Table B following :—

TABLE B.

l_y/l_x	1·0	1·1	1·2	1·3	1·4	1·5	1·75	2·0	2·5	3·0
Z_x .	0·295	0·358	0·419	0·477	0·532	0·581	0·681	0·757	0·869	0·940
Z_y .	0·295	0·237	0·191	0·154	0·127	0·107	0·071	0·051	0·032	0·022

14. *Resistance to shear*—

(*a*) *General.*—(i) The shear stress s at any cross-section in a reinforced concrete beam or slab should be calculated from the following equation (3) :—

$$s = \frac{S}{ba} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where S is the total shearing force across the section;

b is the breadth of a rectangular beam or the breadth of the rib of a T-beam;

and a is the arm of the resistance moment.

(ii) Where at any cross-section the shear stress, as calculated from equation (3) above, exceeds the permissible shear stress for the concrete, the whole shearing force should be provided for by the tensile resistance of the shear reinforcement acting in proper combination with the compression in the concrete. Moreover, even with the whole shearing force so provided for, the shearing stress as calculated from equation (3) above should not exceed four times the permissible shear stress for the concrete alone.

(*b*) *Shear reinforcement.*—(i) Tensile reinforcement which is inclined and carried through a depth of beam equal to the arm of the resistance moment may be considered as shear reinforcement provided it is anchored sufficiently.

(ii) Where two or more types of shear reinforcement are used in conjunction the total shearing resistance of the beam may be assumed to be the sum of the shearing resistances computed for each type separately.

(iii) The spacing of stirrups should not exceed a distance equal to the arm of the resistance moment. The resistance to shear 'S' should then be calculated from the following equation (4) :—

$$S = \frac{t_w A_w a}{p} \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where t_w is the permissible tensile stress in the shear reinforcement;

A_w is the cross-sectional area of the stirrup;

a is the arm of the resistance moment;

and p is the pitch or spacing of stirrups.

(iv) The resistance to shear at any section of a beam reinforced with inclined bars may be calculated on the assumption that the inclined bars form the tension members of one or more single systems of lattice girders in which the concrete forms the compression members. The shear resistance at any vertical section should then be taken as the sum of the vertical components of the tension and compression forces cut by the section.

15. *Bond and anchorage.*—(i) A plain bar in tension should extend from any section for a distance not less than n times the diameter of the bar, where :—

$$n = \frac{\text{the tensile stress in the bar}}{\text{four times the permissible bond stress}}$$

In no case should n be taken at less than 12. That is, an anchorage of at least 12 bar diameters should be provided beyond the section at which tension commences theoretically.

(ii) When reinforcement in the form of plain bars is used to resist stresses induced by bending, the bond stress s_b due to a variation in tensile stress, calculated from the following equation (5) should not at any point exceed twice the appropriate permissible bond stress.

$$s_b = \frac{S}{ao} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where S is the total shear across the section less the resistance to shear provided by the inclined bars.

a is the arm of the resistance moment;

and o is the sum of the perimeters of the bars in the tensile reinforcement.

In members of other than uniform depth the effect of the change in depth on the bond stress should also be taken into account.

(iii) A hook at the end of a bar should be of the form indicated in fig. 2, and its inner diameter should be not less than four times the diameter of the bar; except that when the hook fits over a main reinforcing or other adequate anchor bar the diameter of the hook may be equal to the diameter of such main reinforcing or anchor bar. A bend in a reinforcing bar may be assumed to increase the bond strength of the bar by an amount equivalent to that of a length of bar equal to four times the diameter of the bar for each 45 degrees through which the bar is bent; provided that:—

(*a*) the radius of the bend be not less than twice the diameter of the bar;

(*b*) the length of the straight part of the bar beyond the end of the curve be at least four times the diameter of the bar; and

(*c*) whatever be the angle through which the bar is bent, the assumed increase in the bond strength shall not be taken as more than that of a length of bar equal to sixteen times the diameter of the bar.

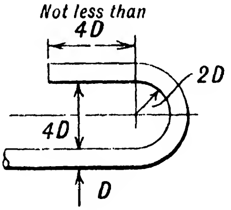


FIG. 2.

(iv) When an attachment is used in place of a hook, the permissible stress in the concrete may be increased to three times the value permitted for the concrete in direct compression, provided the end anchorage is of such form as to prevent local failure of the concrete.

(v) Notwithstanding any of the above notes, in the case of secondary reinforcement such as stirrups and binding, complete bond length and anchorage may be deemed to have been provided when a bend in the bar through an angle of at least 90 degrees passes round a bar of at least its own diameter and the bar is continued beyond the end of the curve for a length of at least eight diameters.

Columns.

16. *Short columns with lateral ties.*—The axial load P permissible on a short column reinforced with longitudinal bars and lateral ties should not exceed that given by the following equation (6):—

$$P = c(A_c + mA) \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where c is the permissible stress for the concrete ;

A_c is the cross-sectional area of concrete excluding any finishing material applied after the casting of the column ;

m is the modular ratio (15).

and A is the cross-sectional area of the longitudinal steel.

17. *Short columns with helical reinforcement.*—Where helical reinforcement is used, the axial load P permissible on a short column should not exceed that given by equations (6) or (7), whichever is the greater :—

$$P = c(A_k + mA) + 2t_b A_b \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where A_k is the cross-sectional area of concrete in the core ;

t_b is the permissible stress in the helical reinforcement ;

and A_b is the equivalent area of helical reinforcement (volume of helix per unit length of the column).

The sum of the loads contributed by the concrete in the core and by the helix should not exceed $0 \cdot 5uA_c$, where u is the crushing strength of the concrete required from the works test.

When, in a column having helical reinforcement, the permissible load is based on the core area, the radius of gyration also should be based on the core area of the column.

18. *Long columns.*—The permissible working load for a long column should not exceed that calculated as above for a short column, multiplied by the appropriate buckling coefficient.

19. *Bending in columns.*—Bending moments in internal columns supporting an approximately symmetrical arrangement of beams and loading need not be calculated.

Bending moments in external columns and in internal columns supporting an arrangement of beams and loading not approximately symmetrical should be calculated and provided for.

Unless more exact methods are preferred, the bending moments in external (and similarly loaded) columns may be estimated from the following equations (8), (9), (10) and (11) :—

Moment at foot of upper column . $M_e\left(\frac{K_u}{K_l + K_u + 0 \cdot 5K_b}\right)$ for a frame of one bay . . (8)

Moment at foot of upper column . $M_e\left(\frac{K_u}{K_l + K_u + K_b}\right)$ for frames of two or more bays . (9)

Moment at head of lower column . $M_e\left(\frac{K_l}{K_l + K_u + 0 \cdot 5K_b}\right)$ for a frame of one bay . . . (10)

Moment at head of lower column . $M_e\left(\frac{K_l}{K_l + K_u + K_b}\right)$ for frames of two or more bays . (11)

where M_e is the bending moment at the end of the beam framing into the column, assuming fixity at the connection ;

K_u is the stiffness of the upper column ;

K_l is the stiffness of the lower column ;

and K_b is the stiffness of the beam.

The equations for the moment at the head of the lower column may be used for columns in a topmost storey by taking K_u as zero.

For the purposes of these equations, the 'stiffness' of a member may be taken as the quotient obtained by dividing the moment of inertia of a cross section by the length of the member, provided the member be of constant cross-section throughout its length.

20. *Combined axial and bending stresses.*—The maxima stresses on longitudinal reinforcement and concrete due to combination of direct load and bending action should not exceed the permissible stresses for bending, multiplied by the appropriate buckling coefficient.

Code of Practice for the Use of Reinforced Concrete in Buildings.

The Codes of Practice Committee, formed under the ægis of the Ministry of Works, has recently issued a British Standard Code of Practice, C.P. 114 (1948), for the structural use of normal reinforced concrete in buildings. For concretes made with approved aggregates (B.S. 882), this code recommends even higher permissible stresses than those allowed for quality. A concrete in the L.C.C. By-laws. Where the aggregates do not comply with B.S. 882, the permissible stresses are based on the actual cube strengths in preliminary and works tests. The code assumes a modular ratio of 15.

The Memorandum on Bridge Design and Construction No. 577, issued by the Ministry of War Transport, like the D.S.I.R. Code of Practice which preceded it, assumes a variable modular ratio equal to 40,000 divided by three times the permissible concrete bending stress, expressed in pounds per square inch.

SYSTEMS OF REINFORCED CONCRETE CONSTRUCTION.

Although there exists a considerable number of so-called independent systems of reinforced concrete construction, the principles underlying the same are in every way similar, and it is only in the manner in which these principles are carried out that any difference arises. The several systems can be readily divided into groups, and the following notes will be sufficient here to indicate wherein one differs from another :—

Systems which make use of Plain Steel Rod Reinforcing Bars.

In the Hennebique system, the main bars are set parallel to the lower or tension face of the beam, some being, where necessary, bent or cranked upwards towards the upper face (as where the beam is continuous over a support). In addition to these bars, flat stirrups are hooked at intervals over the lower bars, and carried upwards towards the upper part of the beam; these bind the concrete and steels securely together, and assist largely in resisting the shearing stresses. For columns or piles, the rods are grouped symmetrically about the centre of the pile, and are tied together at intervals by means of straps or wire lacing.

The Coignet system is in many respects very similar to the Hennebique, except that generally a number of smaller rods are used, these being in turn bent upwards as they approach the ends of the beam. A continuous top bar is also provided, to which the bent-up members are securely tied or laced. Any additional stirrups or binding ties which are necessary are formed of small section round bars, varying from $\frac{3}{16}$ in. to $\frac{1}{4}$ in. diameter; these are laced or tied between the lower or main tension bars and the upper bar.

In the Considère system, the portions of any structure under compression are provided with a close-set spiral of steel wire or small-section steel rod, which, by surrounding the concrete and preventing its displacement, enables the enclosed portion to resist safely very high compressive stresses.

A number of other systems in which plain bars are used have recently appeared, the difference between them being in the manner in which the several bars are connected together, and the method of providing for shearing stresses, and for connecting the shear bars to the main bars.

Reinforced Concrete or Fire-Resisting Floors.

Numerous systems of these are on the market, but they can be grouped together into types, of which the following may be noted :—

(1) A series of hollow or solid blocks of fireclay, concrete, or other material are laid at short distances apart, and the space between same filled up with concrete or cement grouting, with steel rods embedded in same. In this way a series of reinforced concrete beams are formed surrounding the blocks. Where the area to be covered is approximately square, or the proportion of length to breadth does not much exceed 2 to 3, these reinforced concrete beams can be formed on all four sides of the blocks; but where hollow blocks are used, the open ends must be closed up, or provision made to prevent the concrete finding access into the hollows which are intended to lighten the weight of the floor. Of this type may be mentioned the 'Kleine,' the 'Fram,' Diespeker's 'Bigspan,' etc.

(2) Hollow blocks, usually of arched formation, and of a width or span of from 2 to 3 feet, or over, are laid between and supported by lines of rolled steel beams, or of reinforced concrete beams, these latter being made either at works, and delivered and erected complete, or formed *in situ* during the construction of the floor. A level floor surface is obtained by a light filling of breeze concrete over the blocks. Where steel beams or finished reinforced concrete beams are used, the floor can be erected without the use of centering. The dispensing with large masses of wet concrete minimises all trouble due to dripping from the new floor, and allows it to be used immediately after it has been erected.

A modification of this floor consists in the use of half-arch shaped blocks, which are set into the webs of steel beams, and brought together at the centre where they are lightly grouted at their meeting edges. A levelling-up covering of breeze concrete completes the floor. This type is represented by the 'Fram' arched floor, the 'Carlisle' tubular arch floor, etc.

(3) Concrete tubes of various sections and of considerable lengths may be formed, and reinforced where necessary with steel rods or wire. By placing these close together, it is only necessary to grout in between the joints in the several tubes to complete the floor ready for any top finish; the advantage of a dry laid floor is thus secured. The 'Siegwart,' etc., provide this type of floor.

(4) Floors may be formed of solid concrete slabbing reinforced with plain or special section bars as required, and supported by rolled-steel beams or reinforced concrete beams. Such types are supplied by any reinforced concrete contractor.

(5) Floors provided with a continuous mesh reinforcement :—The floors provided by the Expanded Metal Company and by the Lock Woven Mesh Company may be taken as examples of this class.

Other types of floors are to be met with, each possessing some particular qualification or arrangement peculiar to itself.

DESIGNING REINFORCED CONCRETE STRUCTURES.

Concrete is very strong under compression, but weak under tension. Good average concrete will safely resist a compressive stress of from 600 to 1,000 lbs. per square inch of section; its tensile strength does not exceed about one-tenth of this value, and may altogether disappear through the formation of minute cracks.

Concrete and steel possess practically the same coefficient of expansion, and therefore expand and contract together under changes of temperature, without developing internal stresses.

Concrete adheres strongly to a clean surface of steel; the average safe adhesion can generally be taken as about 100 lbs. per super. inch. It is thus possible to form a practical and economical combination of concrete and steel, the former material being utilised to resist the compressive stresses, with the necessary steel bars embedded at the points where tensile stresses occur. In certain cases, where the concrete alone cannot provide all the necessary compressive resistance, steel rods are added to assist the concrete; but from an economical point of view this should be avoided whenever possible, for the following reason :—

Steel possesses, on an average, about fifteen times the elasticity of concrete; where steel bars are embedded in concrete to assist in resisting compression, it follows that when the concrete around the bars has been stressed up to its limit, the steel has been compressed to fifteen times that of the concrete only, and is still far from its limit. Notwithstanding this want of economy, it is frequently necessary to utilise steel in this way, as, for instance, in the case of columns, arches, etc., and in beams when sufficient depth, or area of concrete in the compression flange, cannot be obtained.

Steel is very perfectly protected from deterioration by a covering of concrete, and many instances could be quoted to show that a covering of less than 1 inch of good concrete is sufficient to preserve steel almost indefinitely.

In many cases reinforced concrete forms an almost ideal method of construction, particularly where great strength and durability are essential. The strength of the concrete increases with age, and there is no cost of upkeep, as in the case of steelwork, which requires regular inspection, painting, etc.

All structures can have their several members resolved into three types or into combinations of these three types—namely, beams or girders, struts or stanchions, and ties. Beams or girders support loads which act at right angles to their length, and tend to cause bending or deflection of the member. In struts or stanchions the loads act in the direction of the length, and tend to compress the members; in long and slim compression members, however, stresses similar to those in beams may also arise, through bending or side-yielding when the load exceeds a particular value. In ties the load acts in the direction of their length, as in struts, but here the tendency is to produce tension or stretching in the member.

Composite members may be represented by struts or ties which are not placed vertically, and consequently tend to bend by their own weight, and to this latter extent act as beams. Arches also are generally subjected to bending stresses in addition to that of compression.

Beams or Girders.

The bending moment and shearing force values are calculated exactly as in the case of ordinary beams, and reference may be made to pages 216 and 217 for information on this point. Owing to the composite nature of a reinforced concrete beam, however, the determination of the internal stresses, or of the scantlings of any beam necessary to limit the tension and compression stresses to some specific maximum, are distinctly more complex than in the case of beams formed of a homogeneous material.

The following assumptions, which are practically correct within the usual limits of safe stress and ordinary conditions, are generally admitted, and form the basis on which the formulæ are built up :—

(*a*) The coefficient of elasticity in compression of good concrete is constant throughout the working limit of stress, and for an average 1 : 2 : 4 mixture may be taken as $\frac{1}{15}$ that of the steel reinforcement. For any distance from the neutral axis, therefore, the stress per square inch on the steel is fifteen times that on the concrete.

(*b*) The tensional resistance of concrete is ignored.

(*c*) The stress on the concrete varies uniformly from zero at the neutral axis to a maximum at outer edge ; the stress on the area of the steel is constant.

(*d*) The bond, or adhesion, between the concrete and the steel is sufficient to transmit any stress arising between these two materials.

Considering, in the first instance, a beam formed of some homogeneous material, supported at each end, and loaded so that it tends to deflect, the compressive stress at the upper edge of the beam will be exactly equal to the tensile stress at the lower edge, and these values will gradually diminish towards the centre of the depth, where they will altogether disappear ; this plane of no stress is called the neutral axis.

Assuming that the stresses vary uniformly, fig. 3 may be taken to represent the stresses in any section of the beam, the horizontal distance between the vertical line and the inclined line at any point above the neutral axis will be proportionate to the compressive stresses at the given point ; similarly, the horizontal lengths below the neutral axis will be proportionate to the tensile stress.

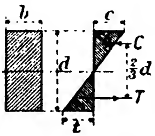


FIG. 3.

The total resistance to compression or extension would be represented by the area of either triangle. The resultant of either of these stresses would act through the centre of gravity of these triangles, or at a distance equal to $\frac{1}{3}$ the perpendicular depth from the base, giving the distance between the two resultants, or the lever arm of the resistance couples, as equal to $\frac{2}{3}d$.

The corresponding diagram for a simple reinforced concrete beam would be that of fig. 4. Here the compressive resistance, which occurs above the neutral axis, is represented, as before, by a triangle ; while the tensile resistance, being that of the steel reinforcement only, is represented by a line (the tensile strength of the concrete is ignored). The position of the resultant of the compressive resistance would pass through the centre of gravity of the triangle, or at $\frac{1}{3}n$ from the upper surface ; the lever arm in this case would equal $d - \frac{n}{3}$.

Where double reinforcement is adopted—that is to say, where a certain amount of steel is introduced into the upper or compression area of the beam to assist the concrete (fig. 5)—the total compressive resistance would, in addition to that of the concrete, include the compressive resistance of the steel ; the combined resultant of these two resistances would equal the sum of the two, and would act at some point between the two independent resistances, the exact position depending on the relative values of these two resistances.

In the case of T beams, or those in which the upper or compression flange is formed in part of the flooring slabs with which the beam proper is combined, two cases arise : the one in which the distance from the neutral axis to top of beam (n) is *not greater* than the thickness of the floor slab (d_s), and the other in which this value (n) is *greater* than the thickness of the slab (d_s). Since the compressive stress is assumed as being resisted by the concrete alone, the first case naturally falls under the same condition as that of a simple beam (fig. 4), except that the breadth of the beam must now be taken, not as the breadth of the reinforced beam only (b_r), but must include a certain portion of the slab at each side of the main beam, or b_s.

The difference between the first and second cases referred to will be obvious by a comparison between figs. 6 and 7. In the former the compression resistance is represented by a triangle, with a resultant acting $\frac{1}{3}n$ from the upper surface; while in the latter case the compressive resistance is represented by the area of a trapezoid, the point of action being through the centre of gravity of this figure. It should be noted that in this latter case the compressive value of the concrete in the main beam between the bottom of slab and the neutral axis is ignored, as its value is very slight, and the formulæ are much simplified by doing this.

Two extensions of the above two cases are shown in figs. 8 and 9, where double reinforcement is adopted ; the treatment in these two cases is modified generally, as previously noted when referring to figs. 4 and 5 for rectangular beams.

The following series of formulæ for the several types of single and double reinforced beams just referred to will enable the scantlings for any such members to be readily calculated or the designs for these checked with the minimum of trouble.

Rectangular Beams with Single Reinforcement.

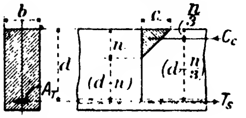


FIG. 4.

$$n=\frac{1}{1+\frac{t}{cm}} \quad . \quad . \quad . \quad (1)$$

$$r=\frac{1}{\frac{2t}{c}\left(1+\frac{t}{cm}\right)} \quad . \quad . \quad . \quad (2)$$

$$t=c\frac{(1-n)m}{n} \quad . \quad . \quad . \quad (3)$$

$$c=t\frac{n}{(1-n)m} \quad . \quad . \quad . \quad (4)$$

$$A_T=\frac{B}{td\left(1-\frac{n}{3}\right)} \quad . \quad . \quad . \quad (5)$$

$$n=\sqrt{(mr)^2+2mr}-mr \quad . \quad . \quad (6)$$

$$R=bd^2\left[tr\left(1-\frac{n}{3}\right)\right] \quad . \quad . \quad (7)$$

$$R=bd^2\left[\tfrac{1}{2}cn\left(1-\frac{n}{3}\right)\right] \quad . \quad . \quad (8)$$

SYMBOLS.

n = Ratio of distance of neutral axis from compression edge of beam to effective depth.
r = Ratio of area of steel to area of concrete $= A_T/bd$
t = Tensile stress intensity on steel.
c = Compressive stress intensity on concrete.
m = Modulus ratio $= E_s/E_c$.
A_T = Area of tensile reinforcement.
B = Bending moment of external forces.
R = Resistance moment of internal stresses.
b = Breadth of beam.
d = Effective depth of beam.

DESIGNING.—Decide values for m, t, c, and trial dimensions for b and d. Calculate n by (1) and r by (2); then value of A_T by (5). R is got from (7) or (8). If any other value of r is used than is given by (2), calculate n by (6) and R by (7) and (8), the lesser of these two values being adopted.

CHECKING A DESIGN.—Calculate r $(=A_T/bd)$, and with this determine n from (6); test t and c by (3) and (4). If t exceeds the safe stress, get R from (7); if c exceeds the safe stress, get R from (8), in each case using the maximum safe stress for t or c.

Note.—Where more than one bar is used to form the reinforcement, measure d to the centre of gravity of the group.

Usual Dimensions.—d = one-twelfth to one-sixteenth of span; b = one-half to two-thirds d.

Rectangular Beams with Double Reinforcement.

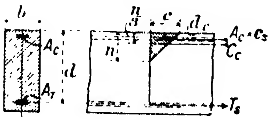


FIG. 5.

$$c_s=\frac{(n-d_c)}{n}m\times c \quad . \quad . \quad . \quad (9)$$

$$c_s=\frac{(n-d_c)}{(1-n)}t \quad . \quad . \quad . \quad (10)$$

$$A_c=\frac{B-\left[\tfrac{1}{2}cnbd^2\left(1-\frac{n}{3}\right)\right]}{c_s d(1-d_c)} \quad . \quad (11)$$

$$A_T=\frac{B-\left[A_c c_s d\left(\frac{n}{3}-d_c\right)\right]}{dt\left(1-\frac{n}{3}\right)} \quad (12)$$

$$n=\sqrt{m^2(r_t+r_c)^2+2m(r_t+r_c d_c)}-m(r_t+r_c) \quad (13)$$

$$R=bd^2\left[tr_t\left(1-\frac{n}{3}\right)+c_s r_c\left(\frac{n}{3}-d_c\right)\right] \quad (14)$$

$$R=bd^2\left[\tfrac{1}{2}cn\left(1-\frac{n}{3}\right)+c_s r_c(1-d_c)\right] \quad (15)$$

ADDITIONAL SYMBOLS.

c_s = Compressive stress intensity on steel.
d_c = Ratio of depth to centre of compression reinforcement to effective depth.
A_c = Area of compression reinforcement.
r_t = Ratio of area of steel in tension to area bd.
r_c = Ratio of area of steel in compression to area bd.

DESIGNING.—Calculate n by (1), then c_s by (9) or (10); the area of Compression Steel is got from (11), and the area of Tension Steel from (12). The Resistance Moment is got from (14) or (15).

CHECKING A DESIGN.—Calculate r and r_c from scantlings given, and n from (13). c_s is got from (9) and (10), the smaller value being adopted. R is obtained from (14) and (15), the former assuming the concrete, and the latter the steel as stressed to their respective safe limits; the least value of R from (14) and (15) must consequently be taken as the maximum safe R.

T Beams with Single Reinforcement.

Class I.—n not greater than d_s.

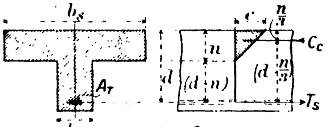


FIG. 6.

$$n = \frac{1}{1+\frac{t}{cm}} \quad . \quad . \quad . \quad . \quad . \quad (16)$$

$$r = \frac{b_s}{\frac{2t}{c}\left(1+\frac{t}{cm}\right)b_r} \quad . \quad . \quad . \quad . \quad (17)$$

$$t = c\frac{(1-n)m}{n} \quad . \quad . \quad . \quad . \quad (18)$$

$$c = t\frac{n}{(1-n)m} \quad . \quad . \quad . \quad . \quad (19)$$

$$A_T = \frac{B}{td\left(1-\frac{n}{3}\right)} \quad . \quad . \quad . \quad . \quad . \quad (20)$$

$$n = \sqrt{\left(mr\frac{b_r}{b_s}\right)^2 + 2mr\frac{b_r}{b_s}} - mr\frac{b_r}{b_s} \quad . \quad (21)$$

$$R = b_r d^2\left[tr\left(1-\frac{n}{3}\right)\right] \quad . \quad . \quad . \quad (22)$$

$$R = b_s d^2\left[\tfrac{1}{2}cn\left(1-\frac{n}{3}\right)\right] \quad . \quad . \quad . \quad (23)$$

SYMBOLS.

n = Ratio of distance of neutral axis from compression edge of beam to effective depth.
r = Ratio of area of steel to area ($b_r d$).
t = Tensile stress intensity on steel.
c = Compressive stress intensity on concrete.
m = Modulus ratio = E_s/E_c.
A_T = Area of tensile reinforcement.
B = Bending moment of external forces.
R = Resistance moment of internal stresses.
b_s = Breadth of T slab.
b_r = Breadth of rib.
d = Effective depth of beam.

DESIGNING.—Decide values for m, t, c, and trial dimensions for b_r, b_s, and d. Calculate n by (16) and r by (17), then value of A_T by (20). R is got from (22) or (23). If any other value of r is used than is given by (17), calculate n by (21) and R by (22) and (23), the lesser of these two values being adopted.

CHECKING A DESIGN.—Calculate r ($=A_T/b_r d$), and with this determine n from (21). Test t and c by (18) and (19). If t exceeds the safe stress, get R from (22), if c exceeds the safe stress get R from (23), in each case using the maximum safe stress for c or t.

T Beams with Single Reinforcement.

Class II.—n greater than d_s

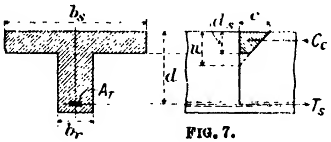


FIG. 7.

$$r = \frac{nd_s\left(1-\frac{d_s}{2n}\right)b_s}{(1-n)mb_r} \quad . \quad . \quad . \quad . \quad . \quad (24)$$

$$A' = \frac{B}{td\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)} \quad . \quad . \quad . \quad . \quad (25)$$

$$n = \frac{d_s^2\frac{b_s}{b_r}+2mr}{\frac{b_s}{b_r}+2mr} \quad . \quad . \quad . \quad . \quad . \quad (26)$$

$$R = bd^2\left[\left(1-\frac{d_s}{2n}\right)cd_s\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)\right] (27)$$

$$R = b_r d^2\left[tr\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)\right] \quad . \quad . \quad (28)$$

ADDITIONAL SYMBOL.

d_s = Ratio of depth or thickness of slab to effective depth of beam.

DESIGNING.—Decide values for m, t, c, and trial dimensions for b_r, b_s, and d. Calculate n by (16) and r by (24), then value of A_T by (25). R is got from (27) or (28). If any other value of r is used than is given by (24), calculate n by (26), and R by (27) and (28), the lesser of these two values being adopted.

CHECKING A DESIGN.—Calculate r ($=A_T/b_r d$), and with this determine n from (26). Test t and c by (18) and (19). If t exceeds the safe stress get R from (28), if c exceeds the safe stress get R from (27), in each case using the maximum safe stress for c or t.

T Beams with Double Reinforcement.

Class I.—n not greater than d_s.

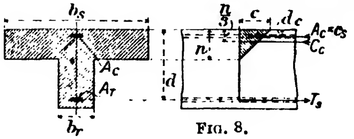


FIG. 8.

$$n=\frac{1}{1+\frac{t}{cm}} \quad . \quad . \quad . \quad . \quad (29)$$

$$t=c\frac{1-n)m}{n} \quad . \quad . \quad . \quad (30)$$

$$c=t\frac{n}{(1-n)m} \quad . \quad . \quad . \quad . \quad . \quad (31)$$

$$c_s=\frac{(n-d_c)}{n}m\times c \quad . \quad . \quad . \quad . \quad (32)$$

$$c_s=\frac{(n-d_c)}{(1-n)}t \quad . \quad . \quad . \quad . \quad . \quad (33)$$

$$A_c=\frac{B-\left[\frac{1}{2}cnb_sd^2\left(1-\frac{n}{3}\right)\right]}{c_sd(1-d_c)} \quad . \quad . \quad (34)$$

$$A_t=\frac{B-\left[A_cc_sd\left(\frac{n}{3}-d_c\right)\right]}{td\left(1-\frac{n}{3}\right)} \quad . \quad . \quad (35)$$

$$n=\sqrt{\left[\frac{m(r_t+r_c)b_r}{b_s}\right]^2+\left[\frac{2m(r_t+r_cd_c)b_r}{b_s}\right]}-\left[\frac{m(r_t+r_c)b_r}{b_s}\right] \quad . \quad (36)$$

$$R=b_rd^2\left[tr_t\left(1-\frac{n}{3}\right)+c_sr_c\left(\frac{n}{3}-d_c\right)\right] \quad . \quad (37)$$

$$R=d^2\left[\frac{1}{2}cnb_s\left(1-\frac{n}{3}\right)+c_sr_cb_r(1-d_c)\right] \quad (38)$$

SYMBOLS.

n=Ratio of distance of neutral axis from compression edge of beam to effective depth.
t=Tensile stress intensity on steel.
c=Compressive stress intensity on concret
c_s=Compressive stress intensity on steel.
m=Modulus ratio=E_s/E_c.
A_t=Area of tensile reinforcement.
A_c=Area of compression reinforcement.
b_s=Breadth of T slab.
b_r=Breadth of rib.
d=Effective depth of beam.
d_c=Ratio of depth to centre of compression reinforcement to effective depth of beam.
r_t=Ratio of area of steel in tension to area b_rd.
r_c=Ratio of area of steel in compression to area b_rd.
B=Bending moment of external forces.
R=Resistance moment of internal stresses.

DESIGNING.—Calculate n by (29), then c_s by (32) or (33), the area of Compression Steel is got from (34), and the area of Tension Steel from (35). The Resistance Moment is got from (37) or (38).

CHECKING A DESIGN.—Calculate r_t and r_c from scantlings given, and n from (36). c_s is got from (32) and (33), the smaller value being adopted. R is obtained from (37) and (38), the former assuming the Steel and the latter the Concrete as stressed to their respective safe limits. The least value of R from (37) and (38) must consequently be taken as the maximum safe R.

T Beams with Double Reinforcement.

Class II.—n greater than d_s.

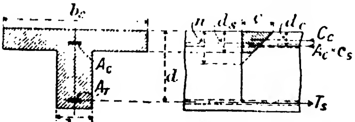


FIG. 9.

$$A_c=\frac{B-\left[\left(1-\frac{d_s}{2n}\right)cdb_sd^2\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)\right]}{c_sd(1-d_c)} \quad (39)$$

$$A_t=\frac{B-\left[A_cc_sd\left(\frac{3nd_s-2d_s^2}{6n-3d_s}-d_c\right)\right]}{td\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)} \quad . \quad . \quad (40)$$

$$n=\frac{d_s^2b_s+2m(r_t+r_cd_c)b_r}{2d_sb_s+2m(r_t+r)b} \quad . \quad . \quad . \quad (41)$$

$$R=d^2\left[\left(1-\frac{d_s}{2n}\right)cd\,b_s\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)+c_sr_c(1-d_s)r\right] \quad . \quad (42)$$

$$R=b_rd^2\left[tr_t\left(1-\frac{3nd_s-2d_s^2}{6n-3d_s}\right)+c_sr_c\left(\frac{3nd_s-2d_s^2}{6n-3d_s}-d_c\right)\right] \quad . \quad (43)$$

ADDITIONAL SYMBOL.

d_s=Ratio of depth or thickness of slab to effective depth of beam.

DESIGNING.—Calculate n from (29), then c_s by (32) or (33), the area of Compression Steel is got from (39) and the area of Tension Steel from (40). The Resistance Moment is got from (42) or (43).

CHECKING A DESIGN.—Calculate r_t and r_c from scantlings given, and n from (41), c_s is got from (32) and (33), the smaller value being adopted. R is obtained from (42) and (43), the former assuming the Concrete and the latter the Steel as stressed to their respective safe limits. The least value of R from (42) and (43) must consequently be taken as the maximum safe R.

Note.—In many cases of double reinforcement the compression reinforcement can be placed at the centre of gravity of the compression area; the formula for R could then be somewhat simplified.

Shearing Stresses and Diagonal Tension in Beams.

These stresses and the methods of resisting them require careful consideration in all cases. At any point in a beam, the horizontal and vertical shearing stresses are of equal value, and these stresses may be looked upon as the connecting link between successive increments of tension in the reinforcement and corresponding increment of compression in the concrete; also, in the case of double reinforcement, in connecting up the increments of tension and compression in the two forms of reinforcement. The tendency of the longitudinal shearing stress is to destroy the adhesion between the concrete and steel, and it is necessary to calculate this value to ensure that no slipping or failure takes place from this cause.

If S represents the total vertical shear, which is equal to the rate of change of the bending moment from point to point along the beam; s_b the shearing stress between concrete and steel; O the periphery of reinforcement, whether one bar or a number of bars; the shearing stress round the bars per unit surface of reinforcement is approximately equal to:—

$$s_b = \frac{S}{O \times 0{\cdot}86d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (44),$$

and this same stress per unit length may be written—

$$s_b O = \frac{S}{0{\cdot}86d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (45).$$

The fraction 0·86*d* in the above equations represents the approximate average length of the lever arm, or distance between the centre of tension in the steel and the centre of compression in the concrete; and these equations may therefore be used to give practical values for rectangular or T beams, either with single or double reinforcement.

If *s* represents the unit horizontal shearing stress in the concrete, the shearing stress per unit length on any width *b* of a beam is as follows:—

$$s \times b = \frac{S}{0{\cdot}86d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (46).$$

This last formula not only gives the value of the horizontal shearing stress, but also that of the vertical shearing stress at the same point, since both these stresses are always of equal value.

Failure by *diagonal tension* is the cause of numerous mishaps, which are frequently attributed to shear. The maximum diagonal tension in a reinforced concrete beam is largely governed by the amount of the horizontal tension in the concrete as well as that of the shearing stress at the same point. From practical experiments and data it would appear that, with average concrete, beams with no web reinforcement are liable to fail by diagonal shear when the vertical shearing stress intensity, at point of failure, averages about 120 lbs. per square inch. Accepting a factor of safety of 4, the shearing stress intensity given by formula (**46**) should not exceed 30 lbs. per square inch. With the same factor of safety, the shearing stress calculated by this formula in beams with web reinforcement may be taken at 120 lbs. per square inch.

Web Reinforcement.

This generally consists either of vertical or diagonal stirrups, alone or in conjunction with inclined bars, these latter generally consisting of a portion of the main reinforcing bars bent upwards as required.

The sectional area of steel to withstand the total shearing force at any vertical section, when the shear bar is inclined at an angle ϕ with the horizontal, should be as follows:—

$$\text{Sectional area required} = \frac{S}{t \times \sin\phi} \quad . \quad . \quad . \quad . \quad (47),$$

t being the tensile strength per square inch of the steel.

The resistance to shear and to diagonal tension in a T beam is practically similar to that for rectangular beams, if the breadth in the former be taken as the breadth of the rib (b_r), this value being compared with the breadth (b) of the rectangular beam.

The breadth of rib in T beams (b_r) will usually be found sufficient to resist any horizontal shear when of a size to give sufficient practical space for embedding the steel reinforcing bars, but to allow for any excessive shear, and for diagonal tension, it is well to keep this breadth of ample size.

Inverted Beams.

Where a beam is continuous over one or more intermediate supports, or where a single span beam is securely fixed or built in at its ends, the effect on the continuous beam over the supports or at the ends of the single span beam is to produce reverse bending, and consequently an inversion of the stresses; the upper portion now being in tension, while the lower portion is in compression. To meet this change in stress, it is usual to bend up a number of the lower reinforcing bars as they approach the ends, and for partial fixing this may be sufficient, but it will generally be found that additional reverse stress bars (sometimes called 'continuity bars') must be added to provide the full area of reinforcement necessary at the upper portion of the beam.

In the case of inverted T beams, it will usually be found that the neutral axis falls outside the slab, and it may therefore be calculated as for an ordinary rectangular beam, of **a breadth (b) equal to the breadth of rib (b_r).**

Deflection of Reinforced Concrete Beams.

An ingenious method of calculating the approximate deflection of a reinforced concrete beam or slab was suggested by Mr. Eli White in the 'Engineering Record' for November 9, 1907. The following explanations are based on this method:—

The deflection of a reinforced concrete beam being proportionate to the elongation of the steel tensile reinforcement, this deflection will be similar in amount, for similar tensile stress intensities, to that of a symmetrical beam, whose flanges are each of an area equal to the tensile reinforcement in the reinforced concrete beam, and are placed at a distance apart equal to twice that of

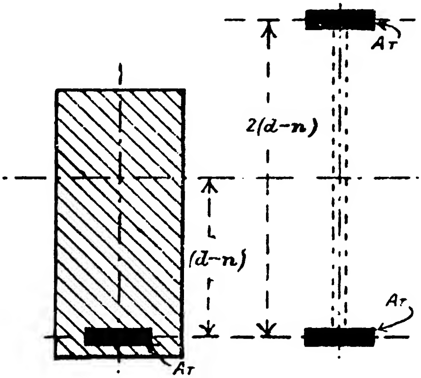


FIG. 10.

the distance between the tensile steel and the neutral axis in the reinforced concrete beam. Fig. 10 indicates to the left a sketch section of a reinforced concrete beam, with tensile reinforcement equal to A_T, and the sketch to the right indicates the section of an imaginary symmetrical steel beam, whose two flanges are each equal in area to A_T and are placed at $2(d-n)$ apart. If t = tensile stress intensity on steel, the

$$\text{Resistance Moment of this symmetrical beam} = 2(d-n) \times A_T \times t. \quad . \quad (48).$$

When any beam is carrying a load, it is obvious that the resistance moment must be equal to the bending moment of the external force acting on the beam.

For this symmetrical beam, the

$$\text{Approximate Moment of Inertia value} = 2A_T \times (d-n)^2 \quad . \quad . \quad . \quad . \quad (49).$$

and taking E at 30,000,000 lbs., the deflection formulæ given in the tables on pages 216 and 217 can be applied to calculate the approximate deflection in any single or double reinforced concrete beam, the correct value for $(d-n)$ being used in the above formulæ.

It should be noted that in the above formulæ the tensile resistance of the concrete has been entirely neglected, and that the formulæ for deflection on pages 216 and 217 are those for beams of uniform section (reinforced concrete beams only more or less approach this condition); for these reasons the calculated deflection will generally be distinctly higher than the actual values, which may at times be found as much as 10 to 15 per cent. less than given by these calculations.

The tables on pages 216 and 217 may prove of service in determining *graphically* the bending moment and shearing stress values for various types of loading; and since these diagrams show clearly the *change of values* they can frequently be used in arranging the main reinforcement and the points where these bars may be bent up to resist shear; independent shear members can also be set out by the same means.

PIPES, CIRCULAR RESERVOIRS OR BINS, ETC.

SYMBOLS USED IN THE FOLLOWING FORMULÆ.

A_h = Sectional area of hoop reinforcement.
C = Direct compressive stress on the shell.
b_r = Breadth of ring under consideration, in inches.
d = Diameter of pipe or reservoir, in inches.
t = Tensile stress in lbs. on steel reinforcement.
p_1 = Pressure in lbs. per sq. in. on shell.
H_w = Head of water, in feet.

When under Internal Pressure.

(Concrete neglected.)

Consider any circumferential strip or ring, then :—

$$A_h = \frac{p_1 \times d \times b_r}{2t} \quad . \quad . \qquad (50)$$

The total area thus found is divided up into any suitable number of hoops or rings.

Where water pressure has to be resisted :—

$$A_h = \frac{0 \cdot 216 H_w \times d \times b_r}{t} \quad . \quad . \quad . \qquad (51)$$

In addition to the hoop or spiral reinforcement determined above, longitudinal bars are necessary to strengthen the concrete shell between each hoop to resist the pressure they have to sustain; the shell here acts as a fixed ended beam or slab of a span equal to the distance between the hoops, and the scantlings calculated by the formulæ previously given for such a type of beam.

Practical considerations generally decide the thickness of the concrete shell, which is usually 1¼ to 2 ins. thick for pipes up to 9 or 10 ins. diameter, and seldom exceeds 3 to 3½ ins. even for large diameter pipes. In vertical reservoirs or bins the thickness towards the base must also be such as will safely carry the weight of the shell.

NOTE.—The required reinforcements for circular bins or silos may be determined by calculating 'p' from formulæ (54) or (55) on next page, for two levels say 12 ins. apart; the difference of the two values thus got will give the pressure per lin. foot and per one foot in height between these levels. Dividing this value by 12×12 (144) will give the average pressure per sq. in., and formula (50) above will at once decide the necessary hoop reinforcement.

When under External Pressure.

(Concrete and hooping both effective.)

$$C = \frac{p_1 d}{2} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (52)$$

The safe load which the shell will sustain depends on the proportion of the steel hooping, and is obtained from formulæ (6), page 396. A few trials with varying thicknesses and proportions, of reinforcements may be required. The minimum practical thickness of shell is that necessary to give complete protection to the steel rods, and to permit of safe transport and handling.

Longitudinal reinforcement is provided under the same conditions as in the last case.

STORAGE BINS OR SILOS.

SYMBOLS USED IN THE FOLLOWING FORMULÆ.

h = Depth of bin in feet, or height of material above any point.
b = Breadth of rectangular bin, in feet.
l = Length of rectangular bin, in feet.
d = Diameter of circular bin, in feet.
p = Pressure against sides of bin in lbs. per lin. foot of horizontal perimeter.
w = Weight per cubic foot of material in bin.
μ = Coefficient of friction within the material stored.
μ_1 = Coefficient of friction between material and sides of bin.
(For values of the last three items, see table below.)

Bins can be conveniently divided into two classes, as follows :—

First.—Shallow Bins, where the plane of rupture (or the plane along which the material would tend to slide if the sides yielded) passes out through the material itself before it meets the opposite side of the bin, and

Second.—Deep Bins, where the plane of rupture meets the opposite side of the bin within the mass of the contained material.

The particular depth forming the dividing point between the above two classes is obtained by the following formulæ.

$$h = b \times \left[\mu + \sqrt{\mu \frac{1 + \mu^2}{\mu + \mu_1}}\right] \quad . \quad . \quad . \quad . \quad . \quad . \qquad (53)$$

All depths less than the above value would indicate shallow bins, all greater depths would be classed as deep bins.

For square bins:—

$$\textit{First Class.}\text{—}p = \tfrac{1}{2}wh^2\left[\frac{1}{\sqrt{\mu(\mu+\mu_1)}+\sqrt{1+\mu^2}}\right]^2 \quad . \quad . \quad . \quad . \quad (54)$$

$$\textit{Second Class.}\text{—}p = \tfrac{1}{2}wb^2\left[\frac{\sqrt{\frac{2h}{b}(\mu+\mu_1)+(1-\mu\mu_1)}-\sqrt{1+\mu^2}}{\mu+\mu_1}\right]^2 \quad . \quad (55)$$

The total pressure on bottom of bin =

(Total weight of material) − ($\mu_1 \times p \times$ perimeter of bin in feet) . (56)

The maximum pressure on the bottom of a bin occurs when

$$h = \frac{b}{2}\left[\frac{1+\mu^2}{\mu+\mu_1}\left(\frac{4\mu_1}{3\mu_1-\mu}\right)^2 - \frac{1-\mu\mu_1}{\mu+\mu_1}\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad (57)$$

For Rectangular Bins the values given by formulæ (54) and (55) will be different for the sides and the ends, since in the latter case the value *l* would replace that of *b*.

For Circular Bins substitute *d* for *b* in formulæ (53), (55), and (57).

These formulæ are not applicable to the inclined sides of Hopper bins, nor to the conical or spherical bottoms of shallow bins.

Nature of Grain, etc.	Weight per Cubic Foot in Pounds.	Coefficients of Friction.	
		Grain on Grain.	Grain on Cement.
Wheat	53	0·466	0·444
Barley	43	0·509	0·452
Oats	32	0·532	0·466
Maize	48	0·521	0·423
Beans	50	0·613	0·442
Peas	54	0·477	0·296
Tares	53	0·554	0·394
Linseed	45	0·456	0·414
Cement	100	0·554	0·316
Sand	90	0·674	0·577
Bituminous Coal	50	0·700	0·700
Anthracite Coal (small)	52	0·510	0·510
Ashes	40	0·839	0·839
Coke	28	0·839	0·839

RETAINING WALLS.

Symbols used in the following Formulæ and Diagrams.

h = Height of wall in feet.
w = Weight of a cub. foot of earth in lbs.
e = Angle of surface of earth with horizontal; + when *above*. − when *below* horizontal.
ϕ = Angle of repose of earth.

(*a*) Back of wall vertical, and surface of earth inclined at an angle *e*.

$$\textit{Resultant Pressure} = \frac{wh^2}{20}\left[\text{Cos.}\,e\,\frac{\text{Cos.}\,e - \sqrt{\text{Cos.}^2 e - \text{Cos.}^2 \phi}}{\text{Cos.}\,e + \sqrt{\text{Cos.}^2 e - \text{Cos.}^2 \phi}}\right] \quad . \quad . \quad (58)$$

The *direction* of the resultant pressure is parallel to earth surface, and acts at $\frac{1}{3}h$ from base of wall.

(*b*) Back of wall vertical, and surface of earth horizontal,

$$\textit{Resultant pressure} = \frac{wh^2}{2} \times \frac{1-\sin\phi}{1+\sin\phi} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (59)$$

(*c*) Water pressure.

$$\textit{Resultant pressure} = 31 \cdot 25h^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (60)$$

The direction of resultant pressure in (*b*) and (*c*) is horizontal, and acts at a point $\frac{1}{3}h$ from base.

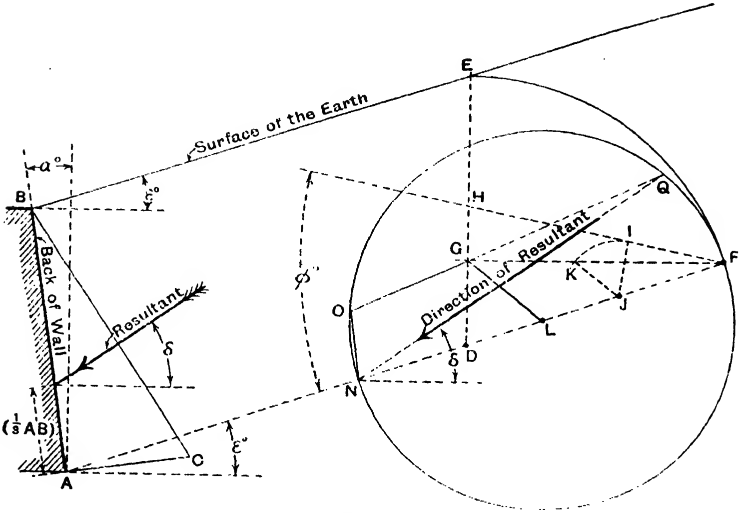


FIG. 11.

To construct this Diagram :—

Draw A F parallel to B E, and at any point D lay off D F = vertical height D E.

Draw F G horizontal, and set off F H, making angle D F H = ϕ.

From any point J in D F describe arc I K tangent to F H, and draw G L parallel to K J.

With L as centre and radius L F, describe circle A D in N.

From N draw N O parallel to A B, cutting circle in O, and join O G.

At A draw A C = O G, and normal to A B.

The area of the triangle A B C multiplied by w will give the resultant pressure on the wall.

The point of action of the resultant is at $\frac{1}{3}$ A B from A.

To find the direction of the resultant pressure, produce O G to Q and join Q N, the direction of the line Q N is that of the resultant pressure

DOMES AND CONICAL COVERINGS.

The following graphic method of determining the stresses in domes is only correct for thin coverings of true spherical or conical section, and formed of a material of uniform weight per unit of surface, and capable of resisting both tension and compression. The majority of reinforced concrete domes approximate to these conditions.

For a detailed explanation of the following statements and formulæ, &c., a reference may be made to a Paper by Mr. W. Dunn, Assoc.M.Inst.C.E., F.R.I.B.A., published in the 'Transactions of the Royal Institute of British Architects,' May 21, 1904.

To construct Figs. 12 and 13. Divide centre line of dome (fig. 12) into any convenient number of parts, and draw the horizontal lines to shell, as 1A, 2B, &c. Draw load line 0–12 (fig. 13) and from O draw radial lines parallel to tangents on the curve at points A, B, &c., till they meet the corresponding horizontal lines; from these meeting points verticals are drawn to the adjacent horizontal lines to complete the zigzag lines shown. If correctly set out, the outer meeting points will lie on a regular curve as shown.

Select a scale such that the length 0–12 corresponds with the total weight of the dome; then the several heights between any two horizontal lines can be scaled off direct to give the weight of the portion between these two lines. To same scale, the radial lengths from O (fig. 13) to any point A, B, C, &c., indicate the total pressure on any horizontal section; the horizontal lengths A1, B2, &c., to same scale, give the total horizontal or radial thrust; the shorter full lines drawn from A, B, &c., give the variation in the radial thrust between the adjoining section lines.

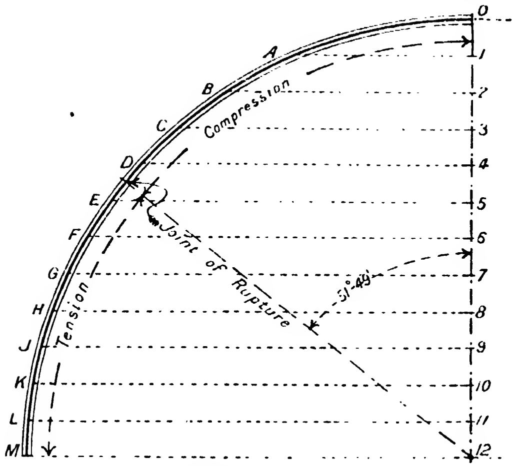


FIG. 12.

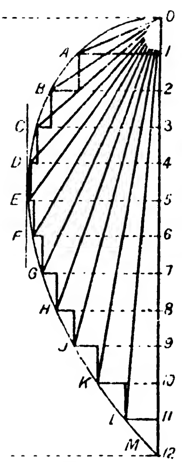


FIG. 13.

To determine the hoop stress, or that acting at the vertical face of any section, the radial thrust must be divided by 6·28 $\left(= \dfrac{\text{Radius}}{\text{Circumference}}\right)$, or by choosing a scale such that the height

0–12 = $\dfrac{1}{6 \cdot 28}$ × total weight of dome, the horizontal lengths to this scale will give the radial thrust direct.

The pressure per lineal unit on any horizontal section is equal to the length of the radial line from O to the particular section measured by the modified scale just mentioned, divided by the radius length at this section.

The position of the joint of rupture is at an angle of about 51¾° with the vertical, or where the tangent to the curve in fig. 12 is parallel to the loadline 0–12; this is also the point of no hoop stress.

The addition of a lantern, or other load, at apex of dome will raise the point O in load line, and also the joint of rupture; the formation of a central opening will depress the point O to level of edge of opening, and lower the joint of rupture.

This graphic method will similarly apply to conical coverings.

ARCHES.

The designing of concrete arched bridges as a rule should be undertaken only by experts, as the accurate determination of the stresses, and the economical arrangement of concrete and steel to resist those stresses, calls for a large amount of skill and experience. The following diagrams and brief explanations will serve to indicate the manner in which such arches can be designed, or the strength of existing arches calculated.

Arched bridges may be divided into three general types :—

1. That in which a continuous arch is provided, without joints or hinges, and connected rigidly to the abutments.

2. Where the arch is hinged or jointed at each abutment, but is continuous between these two points.

3. Where the arch is hinged or jointed not only at the abutments, but also at the centre.

The existence of a hinge or joint implies that any stress in the arch ring must pass through the centre of the joint, and it is necessary to keep this in view when drawing the equilibrium polygon, and to arrange that this polygon passes through the centre of all joints.

Analytical methods of determining the stresses in two-pin and in three-pin arches are described on pages 533 and 534. An analytical method of determining the line of thrust in a parabolic arch with fixed ends is given on page 535, and may be applied to a flat segmental arch which does not sensibly differ from a parabola.

Calculation of Stresses in Arches with Fixed Ends.

To make the solution complete, the derivation, from generally accepted principles, of the conditions to be satisfied, will be briefly indicated. In fig. 14 ds is an extremely short length of an arch

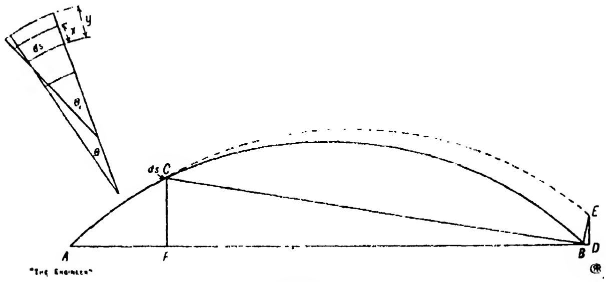


FIG. 14.

with fixed ends at A and B. Throughout this short length the bending moment may be supposed to be constant and equal to M. If the angle subtended by this length before bending is θ and due to the bending moment M is changed to θ_1, the length of a fibre distant x from the neutral axis will have changed from $ds + \theta x$ to $ds + \theta_1 x$, the difference being

$x(\theta - \theta_1)$ or $\frac{x(\theta - \theta_1)}{ds + \theta x}$ per unit of length. The maximum stress f, due to the bending moment M,

s $\frac{My}{I}$, where y is the distance of the extreme fibre from the neutral axis and I is the moment of inertia of the section. The stress on the fibre under consideration is

$$\frac{x}{y} \times \frac{M}{I} = \frac{Mx}{I}$$

Therefore, E, the modulus of elasticity, is:

$$\frac{Mx}{I} \div \frac{x(\theta - \theta_1)}{ds + \theta x}$$

$$\text{and } \theta - \theta_1 = \frac{M(ds + \theta x)}{EI} = \frac{Mds}{EI}$$

θx being negligible compared with ds.

If only this short length of the arch is subjected to bending and the end towards A is supposed to remain fixed, the effect of the bending will be to deflect the end B to a position E, the angle BOE being equal to $\theta - \theta_1$, and therefore equal to $\frac{Mds}{EI}$. As this angle is extremely small, the angle OBE is practically a right angle, and the triangle CFB is similar to the triangle BDE.

Therefore
$$\frac{CF}{OB} = \frac{BD}{BE}$$

or
$$BD = \frac{BE}{OB} \times CF$$
$$= (\theta - \theta_1)CF$$
$$= \frac{Mds}{EI} . CF \qquad . \quad . \quad . \quad . \quad . \quad (61)$$

Also
$$\frac{FB}{OB} = \frac{DE}{BE}$$

or
$$DE = \frac{BE}{OB} \times FB$$
$$= \frac{Mds.}{EI} FB \qquad . \quad . \quad . \quad . \quad (62)$$

In the fixed end arch with rigid abutments the ends are incapable of lateral or vertical movement, and therefore the sum of the quantities BD given by equation (61) is equal to zero, and the sum of the quantities DE given by equation (62) is also equal to zero. The bending of the short section ds is represented by $\theta - \theta_1 = \frac{M ds}{EI}$, and as the tangents at the ends of a fixed end arch remain unchanged, however the arch may be deformed due to bending, the sum of these quantities is likewise equal to zero.

These three conditions will now be applied to determine the resultant thrust line in the arch indicated in fig. 15. As space will not allow of a larger scale being used, the centre line of the arch

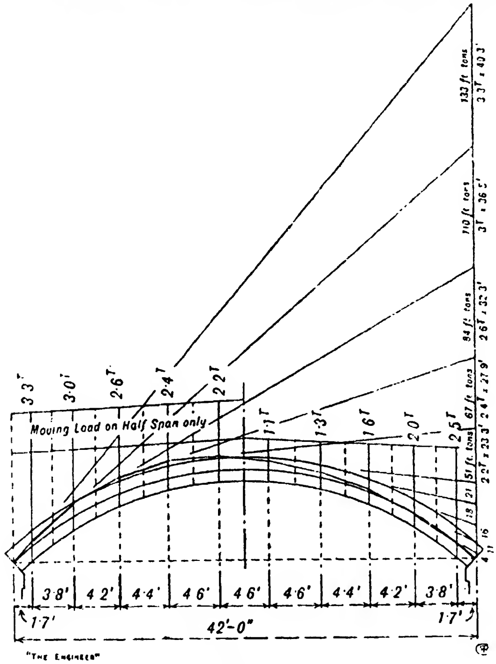


FIG. 15.

has been divided into ten equal parts, but in actual practice it would be preferable to draw the arch to a scale of not less than 4 ft. or 5 ft. to the inch, and to divide the centre line into twenty equal parts. The loads shown on the diagram which correspond to 1 ft. width of the arch, are assumed to be applied at the centre of each of the ten divisions. As the arch is of uniform thickness, the only variable quantity in the expression $\frac{M ds}{EI}$ is the bending moment M, which is equal to the horizontal thrust H multiplied by the vertical distance between the centre line of the arch and the resultant thrust line. By calculating the moment of each load about the right abutment, and plotting these moments consecutively from the springing upwards, as shown in fig. 15, an equilibrium polygon can be drawn, which intersects at the centre of each springing; but as the ends of the arch are not hinged but fixed, the resultant thrust line may intersect the verticals drawn through the centres of the springings above or below these centres. Let these vertical distances above or below the centres of the springings be y at the left abutment and y_1 at the right abutment, distances above the level of the springing being considered positive and distances below this level negative. The ordinates of the resultant thrust line are obtained by dividing the ordinates of the equilibrium polygon, measured from the base line which closes the polygon, by the horizontal thrust H. The respective ordinates of the equilibrium polygon are not altered whether the closure line at the base be horizontal or inclined, and as the calculation depends on small differences between the ordinates of the resultant thrust line and the ordinates of the centre line of the arch, the ordinates of the equilibrium polygon have been calculated for each division by taking moments of all the forces to the left of that division, having previously determined the vertical reaction at the left abutment for a two-hinged arch or simple beam subjected to the same loading, by dividing the sum of the moments of all the loads with reference to the right abutment by the span, *i.e.*, 515 foot-tons ÷ 42 ft. = 12·25 tons.

The expressions which follow represent the vertical distances between the resultant thrust line and the centre line of the arch for each of the ten divisions commencing from the left, and the sum of these expressions is zero.

$$y - \frac{1\cdot7}{42}(y - y_1) + \frac{20\cdot8}{\mathrm{H}} - 1\cdot55$$
$$y - \frac{5\cdot5}{42}(y - y_1) + \frac{54\cdot8}{\mathrm{H}} - 4\cdot23$$
$$y - \frac{9\cdot7}{42}(y - y_1) \times \frac{79\cdot8}{\mathrm{H}} - 6\cdot30$$
$$y - \frac{14\cdot1}{42}(y - y_1) + \frac{94\cdot5}{\mathrm{H}} - 7\cdot70$$
$$y - \frac{18\cdot7}{42}(y - y_1) + \frac{99\cdot0}{\mathrm{H}} - 8\cdot41$$
$$y - \frac{23\cdot3}{42}(y - y_1) + \frac{93\cdot0}{\mathrm{H}} - 8\cdot41$$
$$y - \frac{27\cdot9}{42}(y - y_1) + \frac{82\cdot2}{\mathrm{H}} - 7\cdot70$$
$$y - \frac{32\cdot3}{42}(y - y_1) + \frac{66\cdot2}{\mathrm{H}} - 6\cdot30$$
$$y - \frac{36\cdot5}{42}(y - y_1) + \frac{44\cdot1}{\mathrm{H}} - 4\cdot23$$
$$y - \frac{40\cdot3}{42}(y - y_1) + \frac{16\cdot6}{\mathrm{H}} - 1\cdot55$$

$$10y - 5(y - y_1) + \frac{651\cdot0}{\mathrm{H}} - 56\cdot38 = 0$$

or
$$y + y_1 + \frac{130\cdot2}{\mathrm{H}} = 11\cdot28 \quad . \quad . \quad . \quad . \quad (63)$$

The sum of these expressions, each multiplied by the respective distance from the left abutment, is also equal to zero and the results are as follows :—

$$1\cdot7y - \frac{1\cdot7^2}{42}(y - y_1) + \frac{35}{\mathrm{H}} - 2\cdot6$$
$$5\cdot5y - \frac{5\cdot5^2}{42}(y - y_1) + \frac{301}{\mathrm{H}} - 23\cdot3$$
$$9\cdot7y - \frac{9\cdot7^2}{42}(y - y_1) + \frac{774}{\mathrm{H}} - 61\cdot1$$
$$14\cdot1y - \frac{14\cdot1^2}{42}(y - y_1) + \frac{1332}{\mathrm{H}} - 108\cdot6$$
$$18\cdot7y - \frac{18\cdot7^2}{42}(y - y_1) + \frac{1851}{\mathrm{H}} - 157\cdot2$$
$$23\cdot3y - \frac{23\cdot3^2}{42}(y - y_1) + \frac{2167}{\mathrm{H}} - 196\cdot0$$
$$27\cdot9y - \frac{27\cdot9^2}{42}(y - y_1) + \frac{2294}{\mathrm{H}} - 214\cdot8$$
$$32\cdot3y - \frac{32\cdot3^2}{42}(y - y_1) + \frac{2138}{\mathrm{H}} - 203\cdot5$$
$$36\cdot5y - \frac{36\cdot5^2}{42}(y - y_1) + \frac{1610}{\mathrm{H}} - 154\cdot4$$
$$40\cdot3y - \frac{40\cdot3^2}{42}(y - y_1) + \frac{669}{\mathrm{H}} - 62\cdot4$$

$$210y - \frac{5996}{42}(y - y_1) + \frac{13{,}711}{\mathrm{H}} - 1183\cdot9 = 0$$

or
$$67\cdot2y + 142\cdot8y_1 + \frac{13{,}171}{\mathrm{H}} = 1183\cdot9 \quad . \quad . \quad . \quad (64)$$

The sum of the numerical coefficients in the first column is equal to five times the span. The values of the squares in the second column are taken from tables and the products figuring in the third and fourth columns are worked out using four-figure logarithms. As the same numbers occur two or three times in the various calculations, by working systematically the apparently laborious operations are very much simplified.

The sum of the original expressions, each multiplied by the respective ordinate of the centre line of the arch, is also equal to zero and results as follows :—

$$\begin{array}{l}
1\cdot55y - \frac{2\cdot6}{42}(y - y_1) + \frac{32}{H} - 2\cdot4 \\
4\cdot23y - \frac{23\cdot3}{42}(y - y_1) + \frac{232}{H} - 17\cdot9 \\
6\cdot30y - \frac{61\cdot1}{42}(y - y_1) + \frac{503}{H} - 39\cdot7 \\
7\cdot70y - \frac{108\cdot6}{42}(y - y_1) + \frac{728}{H} - 59\cdot3 \\
8\cdot41y - \frac{157\cdot2}{42}(y - y_1) + \frac{833}{H} - 70\cdot7 \\
8\cdot41y - \frac{196\cdot0}{42}(y - y_1) + \frac{782}{H} - 70\cdot7 \\
7\cdot70y - \frac{214\cdot8}{42}(y - y_1) + \frac{633}{H} - 59\cdot3 \\
6\cdot30y - \frac{203\cdot5}{42}(y - y_1) + \frac{417}{H} - 39\cdot7 \\
4\cdot23y - \frac{154\cdot4}{42}(y - y_1) + \frac{187}{H} - 17\cdot9 \\
1\cdot55y - \frac{62\cdot4}{42}(y - y_1) + \frac{25}{H} - 2\cdot4
\end{array}$$

$$56\cdot38y - 28\cdot19(y - y_1) + \frac{4372}{H} - 380\cdot0 = 0,$$

or

$$28\cdot19y + 28\cdot19y_1 + \frac{4372}{H} = 380 \quad . \quad . \quad . \quad . \quad (65)$$

These three equations—(63), (64) and (65) determine the values :

$$H = 11\cdot32 \text{ tons}$$
$$y = -0\cdot72 \text{ ft.}$$
$$y_1 = 0\cdot50 \text{ ft.}$$

The resultant thrust line can now be drawn following the procedure indicated in fig. 16. The value of $y = -0\cdot72$ ft. is set off below the centre of the left springing, and the value of $y_1 = 0\cdot50$ ft. above the centre of the right springing. From this point and vertically upwards, the values of the moments of each of the loads about the right abutment, divided by the horizontal thrust, are consecutively set off. This can be done graphically as follows :—The sum of the moments of all the loads with reference to the right abutment is 515 foot-tons, which, divided by the horizontal thrust, 11·32 tons, gives 45·4 ft. This distance PQ is set up vertically on the right abutment to the same scale as the drawing of the arch, starting from the point determined by $y_1 = 0\cdot50$ ft. At Q a horizontal line QR is drawn. The value 515 foot-tons is set off to a convenient scale on a diagonal line PR, and on this line the respective moments of each of the loads are plotted. By drawing horizontal lines through the points thus determined, the distance PQ is divided up into parts representing each moment divided by the horizontal thrust. By using the same construction as that employed in fig. 15, the true resultant thrust line is now completed

The absolute maximum deviations from the centre line of the arch occur at the abutments, and the maximum values within the span occur at divisions 4 and 7. Calculating the amount of the deviation at these divisions by substituting the values of y, y_1 and H in the corresponding expressions in the first summation, gives the values $+ 0\cdot34$ ft. at division 4, and $- 0\cdot35$ ft. at division 7. From these values the accuracy of the graph can be checked and the stresses in the corresponding sections of the arch truly calculated.

At division 7, for instance, the bending moment is 11·32 tons × − 0·35 ft. = − 3·96 foot-tons. The direction of the resultant thrust on the left abutment is clearly indicated in fig. 16, and as the horizontal thrust is uniform throughout the arch, the vertical component of the reaction at the left abutment is evidently

$$\frac{45\cdot4 \text{ ft.} + 0\cdot5 \text{ ft.} + 0\cdot72 \text{ ft.}}{42 \text{ ft.}} \times 11\cdot32^{T} = 12\cdot57 \text{ tons}$$

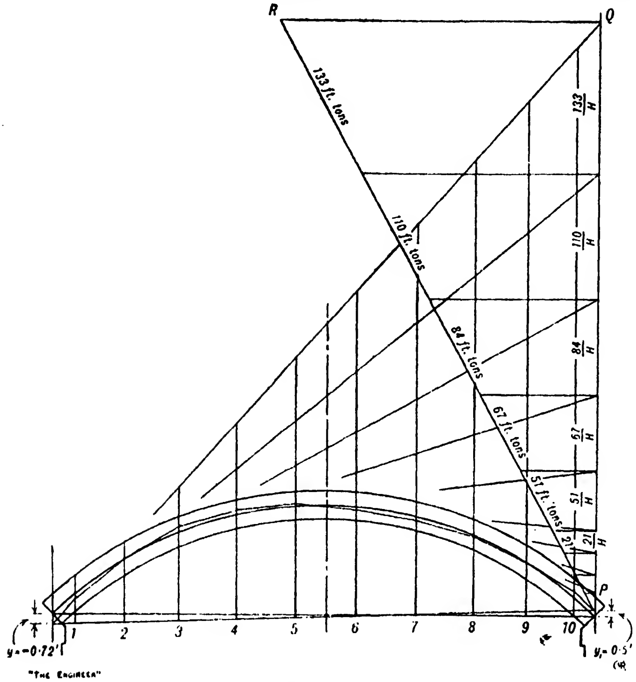


FIG. 16.

The total load is 22 tons and, therefore, the vertical component of the reaction at the right abutment is 9·43 tons. The vertical shear at division 7 is

$$(9\cdot 43^{T} - 2\cdot 5^{T} - 2^{T} - 1\cdot 6^{T} - \tfrac{1}{2} \times 1\cdot 3^{T}) = 2\cdot 68 \text{ tons},$$

and the resultant thrust at this division of the arch is therefore equal to

$$\sqrt{11\cdot 32^2 + 2\cdot 68^2} = 11\cdot 63 \text{ tons}.$$

The section of the arch at division 7 is thus subjected to direct compression of 11·63 tons, and to a bending moment of 3·96 foot-tons, the tension due to this bending being at the extrados of the arch.

The calculation described is for an arch of uniform section. If the section is not uniform, each of the expressions giving the vertical distances between the resultant thrust line and the centre line of the arch must be divided by the moment of inertia of the corresponding section before summing them, and these corrected expressions must also be used in proceeding to the second and third summations.

Specification for Reinforced Concrete Work.

The following sketch specification may be of assistance. It can be modified to any required extent to meet special conditions.

Specification for Reinforced Concrete Work to be executed in erecting New Premises at ………… for Messrs. …………

(Date) ……………

……………………………………

Engineer.

……………………………………

Architect.

General Description and Conditions.—The Structure will be erected in strict accordance with the Engineer's Drawings Nos. ...... and ...... and to any further Detail Drawings to be afterwards provided.

The Structure will consist of (*give general description*). The Foundations shall be taken out to (*the clay, ballast?*). The Maximum Load at this point must not exceed ...... tons per square foot. The average depth of foundations may be assumed as ...... below datum level.

The whole of the work included in this Contract shall be constructed of the best materials and workmanship, and shall be carried out to the entire approval of the Consulting Engineer, and be completed to his satisfaction.

No variations or alterations from Specification or Drawings shall be permitted unless same are approved in writing by the Consulting Engineer.

Immediately on acceptance of his tender, the Contractor must provide a complete and detailed Bill of Quantities (priced in ink) for this work. Should any variations or alterations occur during the carrying out of the work, the variations shall be measured up and priced according to the Bill of Quantities, and any necessary additions or deductions thereafter made from the quoted lump sum price. Any materials or labour required during the progress of the work which are not indicated in the Bill of Quantities shall be paid for at Rates or Prices to be agreed with the Engineer.

The Contractor will be responsible for meeting all requirements of the London County Council, or other Local Authorities, with regard to the mode of construction or the carrying out of same, and shall at once notify the Consulting Engineer should such requirements necessitate any modification in his designs or details.

The Contractor will be held responsible for setting out the Works, and for amending any errors due to inaccuracy, or other cause, at his own cost.

The Contractor must provide for these Works all labour, tools, tackle, boards, horsing, centering, scaffolding, cartage, water, etc.

The Contractor must provide and maintain proper sheds, latrines, and other accommodation for the use of workmen, and proper sheds and shelters for the protection of materials.

The Contractor will be held responsible for all losses, accidents, and damages that may happen to the Works, adjoining properties, or persons during the progress, and caused by the execution of the Works.

The Contractor will be bound to complete all the Works contained in this Specification, etc., within .... months of the date of signing the Contract or obtaining possession of site, under the penalty of .... per week or part of a week as liquidated and agreed damages, except the delay in the opinion of the Engineer arises from causes beyond the control of the Contractors, in which case the Engineer shall grant an extension of time in writing.

The Contractor must allow for making good at his own expense any defect or defects which may become noticeable in the Works during a period of twelve months after certified completion.

Samples.—Samples of all materials must be submitted to the Consulting Engineer, and approved by him in writing before same are used. All materials rejected must be removed immediately from the vicinity of the Works.

Cement.—The Cement used must be the best quality of British Portland Cement, of an approved make, and guaranteed to conform to the Specification of the British Standards Institution for slow-setting cement. Before being used all Cement must be tested, as required by the Engineer, at the Contractor's expense. The Cement should be delivered to the Site in sealed bags bearing the maker's name and the weight of the cement contained.

Cement shall be stored on the Site in such a manner as to permit of easy access for proper inspection and identification. The building used for same must be raised from the ground, and properly weatherproofed to protect the cement.

Sand.—The Sand used shall be clean, sharp, and coarse, composed of grains of various sizes up to particles which shall pass a ¼-inch square mesh, but of which at least 75 per cent. shall pass a ⅛-inch square mesh. It shall be free from all ligneous, organic, or earthy matter.

Aggregate.—This must be composed of Thames ballast; screened, broken stone of a hard close-grained quality; or gravel, cleaned free from organic matter, and angular, varied in size as much as possible, and to pass through a ¾-inch ring, but not through a ¼-inch ring.

Proportions of Concrete.—(*Here specify the composition of concrete, which for usual requirements, may be* 1 *part cement*, 2 *sand*, 4 *aggregate. Foundations are frequently made of a weaker mixture, say* 1 : 3 : 6.) These proportions are approximately correct, but in all cases sufficient sand must be added to fill the interstices. The proportioning should be done in an approved manner, as by means of a bottomless box, to ensure each of the materials being accurately measured.

Mixing of Concrete.—The concrete must be thoroughly well mixed in a suitable machine, sufficient clean water being added to convert the whole into a semi-liquid mass, which will quiver under the blows of the rammer. The water for each batch of concrete must be measured in a small tank, so as to ensure the same amount being used each time. When consent is given to mix any of the concrete by hand, it shall be done as follows:—The necessary quantity of the three materials—coarse aggregate, sand, and cement—shall be measured separately, 5 per cent. more cement being added in addition to the above specified amount. The materials should first be thoroughly mixed dry, then the necessary amount of water added, and the mixing continued till the concrete shows a uniform colour. As far as possible, the mixture shall always be brought to the same degree of wetness. A competent foreman must be in constant attendance at the mixer, to give his approval of every batch which leaves the machine.

Allowable Working Stresses.—(*Here specify maximum stresses desired.*)

Steel Reinforcement.—All steel used should have a tensile strength of not less than 60,000 lbs. per square inch of section, and an elastic limit of not less than 50 per cent., nor more than 60 per cent. of the ultimate tensile strength, with an elongation of not less than 20 per cent. in a length of 8 ins. Any bar must stand bending cold round a bar of its own diameter, through an angle of 180°, and should close down upon itself without fracture on the outside of the bent portion. All welding will be prohibited, unless where absolutely necessary, and must be done only with the sanction of the Consulting Engineer.

All bars shall be free from scale, flaws, or other imperfections, and from oil, but a slight coating of rust is not objectionable. Wherever cinder concrete is used, the bars before being placed in position should receive a coating of neat cement grouting.

Any temporarily protruding bars which shall be exposed to the weather for any period must be protected from rusting by a thin coat of cement grout.

Centering.—All centering must be of dressed timber with close joints, and must be true to line, and sufficiently braced and strutted to be rigid and free from measurable deflection when carrying the dead load of the concrete which is to be placed upon it. A slight camber of ½ in. per 10 ft. of span should be given to all beams, &c. It must also be without appreciable deflection when men are working upon it, and barrows, &c., are being wheeled over it. The joints must be tight, so as to prevent leakage of the liquid cement. The centering shall be arranged so that the sides of the columns can first be removed, then that at the side of beams, and below floor slabs, and finally that below the beams themselves. The time of removal of these several portions shall be arranged with the Consulting Engineer.

Placing of Reinforcement.—All bars must be accurately placed in the exact position shown on the detail drawings, and great care must be taken that they are not displaced during the process of packing the concrete around them. Where, for unavoidable reasons, it is necessary to stop the concreting work, care must be taken to place in position all the bars which will be embedded in the portion of the concrete work which is being executed.

Placing of Concrete.—Before any concrete is put in position a careful inspection of the centering must be made, to ascertain that no dirt, shavings, loose stones, etc., have been allowed to remain in or about the centering, and the woodwork should be well watered to avoid any danger of its sucking up too much moisture from the concrete.

All concrete shall be placed in its final position in the work as soon as possible after mixing. In no case must more than half-an-hour be allowed to elapse before this is done. The concrete must be placed gently into position, in layers not exceeding 6 inches thick, and must not be tipped or dropped from a height. It must then be thoroughly rammed into position, great care being taken to see that the steel reinforcement is thoroughly surrounded by the liquid cement, and that no voids or cavities are left. This can be ensured only by repeated ramming with a suitable tool.

In hot weather the concrete shall be continually moistened for at least a week after placing in position.

In beams and slabs, if any difficulty is found in getting the concrete to pass through the bars which have been placed in position in the bottom of the timber centering, the concrete at the bottom of the beam should be made of finer and more liquid consistency than that used elsewhere, so that it may freely pass between the bars.

Cessation and Recommencement of Work.—Stoppage of work in laying concrete in slabs, beams, etc., will be allowed only at the centre of the span of the slab, or a beam, and the joint with the new concrete should be a vertical plane at right angles to the direction of the span of beam or slab. No joint must be made towards the ends of beams or slabs, or close to a concentrated load.

Wherever a stop has been made in concreting, before recommencing the work, the surface of the existing concrete must be carefully washed with a stiff brush to remove loose particles and dust, and a thick grout of neat cement must be poured over it before the concrete is rammed against it.

Frosty Weather.—No concreting work should be done when the temperature is near or below freezing point. If a frost comes on shortly after any concreting work has been completed, this should be very carefully covered over with sawdust, straw, or sacking, and the centering must be allowed to remain up for at least as long a time after the complete disappearance of the frost as it would have been left in position if no frost had occurred.

Finish.—All concrete roofs and gutters must be laid with the necessary fall, and finished for asphalt, all as may be directed.

All floors shall be finished. (*Here state whether cement, granolithic, preparation for wood blocking, or other covering is desired.*)

The top surface of all outside walls to be weathered.

Tests.—The Maker's Certificate must be provided, when required, with each lot of steel reinforcing bars supplied, but in addition to this the Consulting Engineer shall choose whatever test pieces he may desire from the material delivered, and shall have same tested by Messrs. Kirkaldy, or other firm, and should the result not comply with Specification the right is reserved to condemn all bars included in same consignment.

The cement shall be tested as required, and must comply with the Specification. Cubes of concrete from the material used shall be made as and when directed, and these shall be tested for crushing strength by Messrs. Kirkaldy, or other firm.

The Consulting Engineer shall select at least one bay in each floor, or the equivalent, and these bays shall be tested to 1½ times the safe superimposed load, by means of bricks, cement bags, or in such other manner as the Engineer may decide. Should the deflection under this test exceed $\frac{1}{380}$th part of the span, or should the beam or floor slab fail to regain its normal position after the removal of the load, or show any indication whatever of weakness, the building shall be considered unsafe, and the contractor shall carry out at his own expense such additional work as the Engineer may consider necessary to make the building secure.

The cost of all such testing shall be borne by the Contractor.

Practical Notes.

Wood Strips.

Particularly long lengths of wood for nailing strips in concrete beams should be avoided; they have been known to absorb the moisture from the concrete, and cause serious cracks in the beams. Short plugs are the safest, and where a nailing strip must be used flush with beams, a moulded impress of the wood strip should be made and the strip then removed, and only fixed when the concrete beam is thoroughly dry, it being then secured by means of the plugs. (*Horobin.*)

Durability.

Reinforced concrete will last as long as plain concrete in any situation provided that certain special precautions are taken during its construction. The precautions to be taken are as follows: The materials (cement, sand, and stone) must be of good quality. They must be most carefully and thoroughly mixed and scientifically proportioned, so as to be practically water-proof and air-proof. The mixture must be fairly wet, and must be well punned into position so as to minimise voids. The aggregate should be as non-porous as possible, and any aggregate which is known to have a chemical action on steel should be avoided. The aggregate should all pass through a ¾ in. mesh. The concrete covering should in no case be less than ½ in., and it is suggested that if round or square bars be used the covering should not be less than the diameter of the bar. In structures exposed to the action of water or damp air the thickness of covering should be increased at least 50 per cent., or the size of the aggregate should be reduced so as to ensure a dense skin. In the case of structures exposed to very severe conditions the concrete might be covered with some impervious coating as an extra precaution. The reinforcement should be so arranged that there shall be sufficient space between one piece and its neighbour to allow the concrete to pass and to completely surround every part of the steel. All steel should be firmly supported during the ramming of the concrete, so as to avoid displacement. It should not be oiled or painted, and thick rust should be scraped and brushed off before placing. The scantling of the various members of the structure should be sufficient to prevent excessive deflection. If electric mains are laid down very great care must be taken that no current is allowed to pass through the reinforced concrete. Fresh water should be used in mixing, and aggregates charged with salt should be washed. These recommendations have regard only to the prevention of corrosion of steel and not to fire-resistance or any other property of reinforced concrete. (*Committee of the Concrete Institute.*)

Causes of Failures in Concrete Structures.

Owing to the failure of several reinforced concrete floors in the United States within ten or a dozen years of their construction, Prof. H. J. M. Creighton, of Swarthmore College, examined a large number of reinforced concrete structures in which cracks were developing, and found that in every case the cracks ran along the reinforcing rods, due to the deteriorating action of salt and brine on the concrete. Solutions of the chlorides reacted with the lime and the silicates in the concrete, and penetrating to the iron of the reinforcement converted it into oxide and hydrate, which occupy more space than the metal and force the concrete apart. It was therefore necessary to waterproof reinforced concrete structures which will be in contact with brine, to cease to use in the concrete beach gravel which has not been thoroughly washed with fresh water, and never to add salt to the concrete to prevent it freezing during building operations in cold weather.

Cracking.

According to a recent report, the cracking of reinforced concrete structures is markedly prevalent in the Philippine Islands. A study of this trouble has shown that not a single structure showing rusted steel has been free from salt, the percentage of which varies considerably. In view of this, engineers in the Philippines have been advised that not only is the use of salt water dangerous in concrete structures, but that beach sand and beach gravel should be employed only after having been thoroughly washed with fresh water. The foregoing, and many other similar pieces of evidence, indicate that salt and brine exert a deteriorating action on concrete.

Specification Notes.

Specifications governing concrete construction generally require that 1–2–4 concrete be used in the best class of reinforced concrete construction. In other reinforced concrete work, and in some mass concrete work, 1–3–5 concrete is specified, while in other mass work, such as dams, gravity retaining walls, etc., 1–3–6 concrete is generally required.

TABLE GIVING THE SECTIONAL AREAS OF **Round and Square Rods** AT VARIOUS DISTANCES APART, IN 12-INCH WIDTH OF SLAB.

Diameter in Ins.	Circumference in Ins.	Area in Sq. Ins.	Distance in Inches from Centre to Centre of **Round** Rods.										
			2½	3	4	5	6	7	8	9	10	11	12
¼	0·785	0·049	0·24	0·20	0·15	0·12	0·10	0·08	0·07	0·06	0·06	0·05	0·05
5/16	0·982	0·077	0·37	0·31	0·23	0·18	0·15	0·13	0·11	0·10	0·09	0·08	0·08
⅜	1·178	0·110	0·53	0·44	0·33	0·26	0·22	0·19	0·16	0·15	0·13	0·12	0·11
7/16	1·374	0·150	0·72	0·60	0·45	0·36	0·30	0·25	0·22	0·20	0·18	0·16	0·15
½	1·571	0·196	0·94	0·78	0·59	0·47	0·39	0·33	0·29	0·26	0·23	0·21	0·20
9/16	1·767	0·248	1·19	0·99	0·74	0·59	0·50	0·42	0·37	0·33	0·30	0·27	0·25
⅝	1·964	0·307	1·47	1·23	0·92	0·74	0·61	0·52	0·46	0·41	0·37	0·33	0·31
11/16	2·160	0·371	1·78	1·48	1·11	0·89	0·74	0·63	0·56	0·49	0·44	0·40	0·37
¾	2·356	0·442	2·12	1·77	1·32	1·06	0·88	0·75	0·66	0·59	0·53	0·48	0·44
13/16	2·552	0·518	2·49	2·07	1·55	1·24	1·04	0·88	0·78	0·69	0·62	0·56	0·52
⅞	2·743	0·601	2·89	2·40	1·80	1·44	1·20	1·02	0·90	0·80	0·72	0·65	0·60
15/16	2·945	0·690	3·31	2·76	2·07	1·66	1·38	1·17	1·03	0·92	0·83	0·75	0·69
1	3·142	0·785	3·77	3·14	2·36	1·88	1·57	1·33	1·18	1·05	0·94	0·86	0·78
1 1/16	3·338	0·887	—	3·55	2·66	2·13	1·77	1·51	1·33	1·18	1·06	0·97	0·89
1⅛	3·534	0·994	—	3·98	2·97	2·39	1·99	1·69	1·49	1·32	1·19	1·08	0·99
1 3/16	3·731	1·107	—	—	3·32	2·66	2·21	1·88	1·66	1·48	1·33	1·21	1·11
1¼	3·927	1·227	—	—	3·68	2·94	2·45	2·09	1·84	1·64	1·47	1·34	1·23
1 5/16	4·123	1·353	—	—	—	3·25	2·71	2·30	2·03	1·80	1·62	1·47	1·35
1⅜	4·320	1·485	—	—	—	3·56	2·97	2·52	2·23	1·98	1·78	1·62	1·48
1 7/16	4·516	1·623	—	—	—	—	3·25	2·76	2·43	2·16	1·95	1·77	1·62
1½	4·712	1·767	—	—	—	—	3·53	3·00	2·65	2·36	2·12	1·93	1·77
Side in Ins.	**Perimeter in Ins.**	**Area in Sq. Ins.**	**Distance in Inches from Centre to Centre of Square Rods.**										
			2½	3	4	5	6	7	8	9	10	11	12
¼	1·00	0·063	0·30	0·25	0·19	0·15	0·13	0·11	0·09	0·08	0·07	0·07	0·06
5/16	1·25	0·098	0·47	0·39	0·29	0·23	0·20	0·17	0·15	0·13	0·12	0·11	0·10
⅜	1·50	0·141	0·68	0·56	0·42	0·34	0·28	0·24	0·21	0·19	0·17	0·15	0·14
7/16	1·75	0·192	0·92	0·77	0·58	0·46	0·38	0·33	0·29	0·26	0·23	0·21	0·19
½	2·00	0·250	1·20	1·00	0·75	0·60	0·50	0·43	0·37	0·33	0·30	0·27	0·25
9/16	2·25	0·317	1·52	1·27	0·95	0·76	0·63	0·54	0·47	0·42	0·38	0·34	0·32
⅝	2·50	0·392	1·88	1·57	1·77	0·94	0·78	0·67	0·59	0·52	0·47	0·43	0·39
11/16	2·75	0·474	2·27	1·90	1·42	1·14	0·95	0·81	0·71	0·63	0·57	0·52	0·47
¾	3·00	0·564	2·71	2·26	1·69	1·35	1·13	0·96	0·84	0·75	0·68	0·61	0·56
13/16	3·25	0·661	3·17	2·64	1·98	1·59	1·32	1·13	0·99	0·88	0·79	0·72	0·66
⅞	3·50	0·767	3·68	3·07	2·30	1·84	1·53	1·31	1·15	1·02	0·92	0·84	0·77
15/16	3·75	0·881	4·23	3·52	2·64	2·11	1·76	1·51	1·32	1·17	1·03	0·96	0·88
1	4·00	1·000	4·80	4·00	3·00	2·40	2·00	1·71	1·50	1·33	1·20	1·09	1·00
1⅛	4·50	1·266	6·08	5·06	3·80	3·04	2·53	2·16	1·90	1·69	1·52	1·38	1·27
1¼	5·00	1·575	7·56	6·30	4·72	3·78	3·15	2·69	2·36	2·10	1·89	1·72	1·57
1⅜	5·50	1·905	9·14	7·62	5·71	4·57	3·81	3·26	2·86	2·54	2·29	2·08	1·90
1½	6·00	2·253	10·81	9·01	6·76	5·41	4·51	3·85	3·38	3·00	2·70	2·45	2·25

SECTION XIII

EARTHWORK — EXCAVATION — FOUNDATIONS — PILING — DAMS—SCAFFOLDING—STAGINGS—SHORING—TIMBERING EXCAVATIONS—RETAINING-WALLS—CHIMNEYS—STONE AND BRICK ARCHES—PIERS (pp. 421-473)

(Contributed by C. G. Kent, M.I.C.E., and J. C. Morris B.Sc., A.M.I.C.E.)

SECTION XIII

EARTHWORK — EXCAVATION — FOUNDATIONS — PILING — DAMS — SCAFFOLDING — STAGINGS — SHORING — TIMBERING EXCAVATIONS — RETAINING-WALLS — CHIMNEYS—STONE AND BRICK ARCHES—PIERS.

EARTHWORK.

To determine the Volume of Earth in a Cutting or Embankment.

First Method (By Simson's Rule).

Divide the length of the cutting or embankment into an even number of equal parts, not less than four, and take the areas of the transverse sections at the points of division. Number all the cross sections consecutively. Add together the areas of the extreme sections, twice the sum of the areas of the intermediate sections bearing odd numbers and four times the sum of the sections bearing even numbers and multiply by one-third of the common distance between the sections. The product will be the volume.

Second Method.

This, although somewhat less accurate than the First Method, is frequently used.

Divide the length of the cutting or embankment into any number of equal parts and take the areas of the transverse sections. Add together half the sum of the areas of the extreme sections and the areas of all the intermediate sections, and multiply by the common distance between the sections.

Third Method.

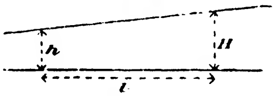


FIG. 1.

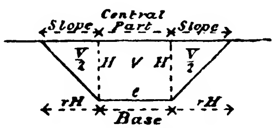


FIG. 2.

H, h = heights of section, in feet, at each end of a length, l, in feet (fig. 1).
rH = breadth of each slope at one end (fig. 2).
rh = breadth of each slope at other end.
(r = ratio of breadth of each slope to height = $\frac{\text{breadth}}{\text{height}}$).
e = base.
$\frac{A}{2}$ = area of each slope at one end.
$\frac{a}{2}$ = area of each slope at other end.
V = cubic contents of both slopes = $\frac{1}{3}(A + \sqrt{Aa} + a) \times l = \frac{1}{3}r(H^2 + Hh + h^2) \times l$.
V = cubic contents of centre = $\frac{1}{2}(H + h)e \times l$.
Q = cubic contents of length, l, in feet = $V + V$.

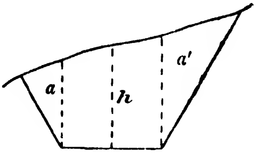
Fourth Method.

If the transverse section is on an incline, instead of horizontal, as in figs. 3 and 4, then, if A, A′, a, and a' are the respective areas of the ends of the slopes,

$$V = \tfrac{1}{6}(A + \sqrt{Aa} + a) \times l + \tfrac{1}{6}(A' + \sqrt{A'a'} + a') \times l. \qquad V = \tfrac{1}{2}(H + h)e \times l.$$

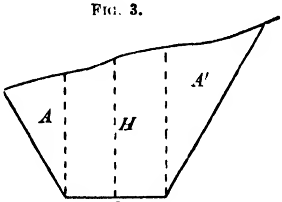
The longitudinal section of the line of railway, road, or canal, having been drawn, and the gradients laid off, the quantities of earthwork may be calculated as follows :—

If the ground is tolerably even, record the heights (or depths in cuttings) on the section at every 100 feet, or some other convenient distance. The sectional areas at each of these recorded heights (or depths) may then be calculated.



c

FIG. 3.



e

FIG. 4.

Let
A = area of section, s = s to 1 slope,
h = height or depth of section,
F = width at formation-level,

all expressed in feet ; then, if the slopes on both sides are the same, as is usually the case,

$$A = h(sh + F).$$

By this formula the areas of the first three sections of a series increasing in height regularly are calculated, and from them the remainder are at once written down by the method of second differences, as follows :

Assume heights to be 1, 2, 3, 4, 5 feet,

$$F = 10, \quad s = 2 \text{ to } 1.$$

Then

	Difference.	Second Difference.
A for 1 ft. = 12		
„ 2 „ = 28	16	4
„ 3 „ = 48	20	
„ 4 „ = 72..	$\frac{4}{24}$	—
„ 5 „ = 100..	$\frac{4}{28}$	—

and so on.

If the cross-slope of the ground is slight, and the areas of the sections do not differ very widely, the quantity of earthwork may be found by taking the mean of the two areas and multiplying by the distance between them, thus :

A′ = area of first section ; A″ = area of second section ; d = distance apart.

$$\text{Cubic contents} = d\,\frac{A' + A''}{2} \text{ feet.}$$

Where the surface of the ground is uneven, the cross slope considerable or the areas of the sections widely different, the best method of finding the volume is to obtain the areas of all the cross sections by means of a planimeter or otherwise and use these areas to obtain the volume by either the First or Second Method.

Earthwork Tables.*

(*H. R. Kempe.*)

The tables on pages 426 to 431 are for the purpose of facilitating the calculation of the volumes of earthwork.

V = cubic contents, in *yards*, of both slopes.
V = cubic contents, in *yards*, of centre part.

V is calculated from the formula

$$V = \tfrac{1}{3}r\,(H^2 + Hh + h^2) \times l,$$

r being = 1, l = 1 chain = 66 feet, and H and h being in *feet* ; so that

$$V = \tfrac{1}{3}(H^2 + Hh + h^2)\,66 \div 27 = \cdot814815\,(H^2 + Hh + h^2) \text{ cubic yards.}$$

V is calculated from the formula

$$V = \tfrac{1}{2}(H + h)e \times l,$$

e being = 1 foot ; so that

$$V = \tfrac{1}{2}(H + h)\,66 \div 27 = 1\cdot22222\,(H + h) \text{ cubic yards.}$$

For any slope other than 1 to 1 the values of V in Table I. must be multiplied by r, that is by $\dfrac{\text{breadth of slope}}{\text{height of slope}}$. Table II. gives the values of V and V for slopes of 1 to $1\tfrac{1}{2}$.

* Revised 1920.

Example.—The length of a cutting is 3 chains, the width 10 feet, the $\frac{\text{breadth of slope}}{\text{height of slope}} = 2$, the height at one end 20 feet, and at the other end 15 feet; what is the cubic content of the cutting?

From table (page 426), $V = 753{\cdot}70$, $\mathsf{V} = 42{\cdot}78$: then,

$$\text{total cubic content} = (753{\cdot}70 \times 2 + 42{\cdot}78 \times 10)3 = 5805{\cdot}60 \text{ cubic yards.}$$

Excavating Machinery.

Excavating machinery may be divided into the following types:—

(1) General purpose navvies.
(2) Trench diggers.
(3) Scrapers, bulldozers, etc.
(4) Power-driven hand tools.

Type (1) is illustrated in Figs. 5 to 9 inclusive, which show the machine in use as (5) a shovel, (6) a back-acting shovel, (7) a skimmer, (8) a drag line and (9) a grab and crane. Machines of this type can be adapted for almost any class of excavation provided timbering does not prevent their use.

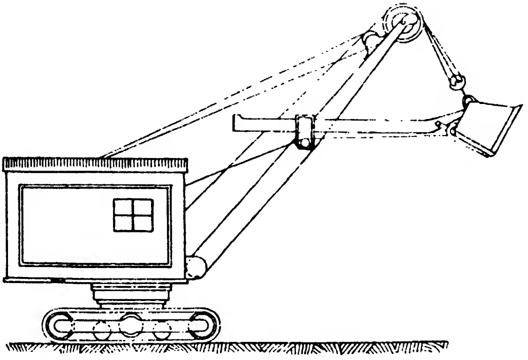


Fig. 5.—Shovel.

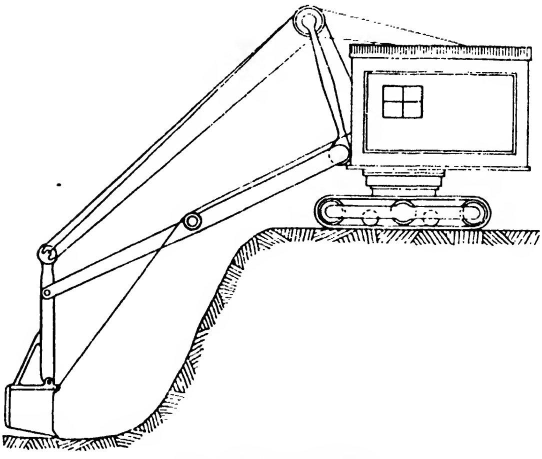


Fig. 6.—Back-Acting Shovel.

Type (2) consists of machines mounted on creeper tracks, which work on the same principle as continuous bucket dredgers. They are suitable for small trenches in ground which will stand for a short time unsupported and which does not contain many mains or service pipes.

Type (3) consists of various units which are pushed or pulled by tractors with creeper tracks. The ground to be excavated may, if very hard, be loosened by some form of plough or ripper drawn

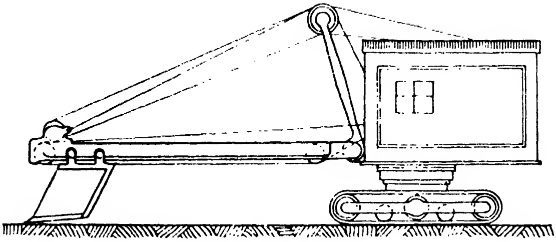


FIG. 7.—Skimmer.

by a tractor. It is then scooped up by a scraper which is, in effect, a shovel mounted on wheels the front end being depressed while scooping and raised for transporting. In some cases a sliding door retains the contents of the scraper which has at the back a plate capable of sliding forward to expel the contents when necessary. The scraper has large balloon tyres to enable it to travel on soft ground.

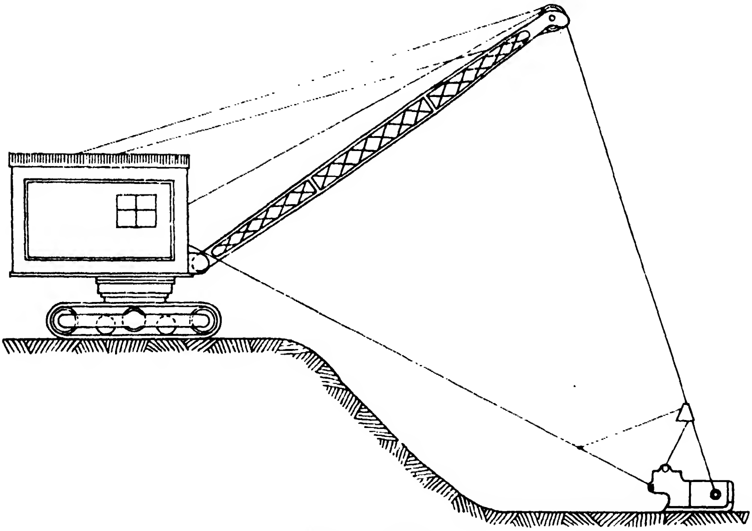


FIG. 8.—Dragline.

After the material has been tipped the surface can be levelled off by means of a bulldozer which is attached to the front of a tractor and consists of a steel plate at right angles to the line of travel of the tractor. This machine removes the material from mounds and deposits it in hollows. A similar machine in which the steel plate is fixed at a smaller angle to the line of travel of the tractor is known as an angledozer and is useful in some cases for moving material sideways to the direction of travel of the machine.

The machines of this type are suitable for levelling and grading large areas in cases in which the excavation is shallow and the material has to be deposited on or near the site. They are largely used in the construction of aerodromes.

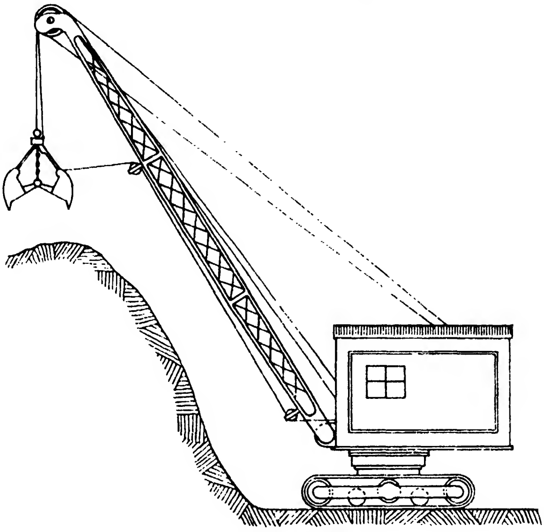


FIG. 9.—Grab and Crane.

Type (4) includes picks, drills, clay shovels, concrete breakers, rammers, etc., driven by compressed air, electricity or internal combustion. They are used where rock, concrete, tough clay or other hard material has to be broken up or where lack of space or other conditions render the use of the previous types impracticable.

Bulking and Shrinkage of Materials.

In estimating the relative amounts of excavation and filling required allowance must be made for the difference in volume of the material before excavation and after being tipped and consolidated. The volume after excavation will also have a bearing on the cost of transportation. Rock, after being broken up will increase in volume very considerably, possibly to the extent of 50 or 60 per cent. on account of the voids formed, but even after being consolidated and partially crushed it will never return to its original volume. On the other hand, loose soil bulks little when excavated, but if placed in an embankment it becomes compressed by the superincumbent weight and may shrink to the extent of 12 or 15 per cent. The relative volumes of other materials may be stated as depending generally upon the pressures to which they are subjected before being excavated and after being tipped and also upon their wetness or otherwise in both cases.

Transporting Earth.

The methods in use for transporting earth comprise barrow runs, horses and carts, motor lorries, tractors and trailers, and rail transport.

Motor lorries with double tyred rear wheels will, except in wet weather, run over ordinary ground without special tracks, but on soft ground tracks formed of old railway sleepers laid transversely are generally used.

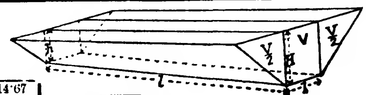
Petrol tipping wagons have become very popular in recent years and are hopper wagons mounted on large balloon tyres. They can move at good speeds over soft ground.

Rail transport includes hand-propelled bogies, horse wagons and trucks drawn by locomotives or by ropes worked by stationary engines.

Gradients on transport ways increase cost if excessive. The following limits should not be exceeded except under special conditions: for barrows, 1 in 30; for horses and carts, 1 in 40; for motor transport, 1 in 30; temporary railway track, 1 in 60.

TABLE I. FOR CALCULATING EARTHWORK.

Slopes 1 to 1. Base = 1 foot. l = 66 feet = 1 chain. H and h in feet. $\overline{V}$ and V in cubic yards.



H	h = 0 $\overline{V}$	0 V	1 $\overline{V}$	1 V	2 $\overline{V}$	2 V	3 $\overline{V}$	3 V	4 $\overline{V}$	4 V
1	·81	1·22	2·44	2·44						
2	3·26	2·44	5·70	3·67	9·78	4·89				
3	7·33	3·67	10·59	4·89	15·48	6·11	22·00	7·38		
4	13·04	4·89	17·11	6·11	22·81	7·33	30·15	8·56	39·11	9·78
5	20·37	6·11	25·26	7·33	31·78	8·56	39·92	9·78	49·70	11·00
6	29·33	7·33	35·04	8·56	42·37	9·78	51·33	11·00	61·92	12·22
7	39·93	8.56	46·45	9·78	54·59	11·00	64·87	12·22	75·78	13·44
8	52·15	9·78	59·48	11·00	68·44	12·22	79·04	13·44	91·26	14·67
9	66·00	11·00	74·15	12·22	83·93	13·44	95·33	14·67	108·37	15·89
10	81·48	12·22	90·44	13·44	101·04	14·67	113·26	15·89	127·11	17·11
11	98·59	13·44	108·37	14·67	119·78	15·89	132·81	17·11	147·48	18·33
12	117·33	14·67	127·93	15·89	140·15	17·11	154·00	18·33	169·48	19·56
13	137·70	15·89	149·11	17·11	162·15	18·33	176·81	19·56	193·11	20·78
14	159·70	17·11	171·93	18·33	185·78	19·56	201·26	20·78	218·37	22·00
15	183·33	18·33	196·37	19·56	211·04	20·78	227·33	22·00	245·26	23·22
16	208·59	19·56	222·44	20·78	237·93	22·00	255·04	23·22	273·78	24·44
17	235·48	20·78	250·15	22·00	266·44	23·22	284·37	24·44	303·92	25·67
18	264·00	22·00	279·48	23·22	296·59	24·44	315·33	25·67	335·70	26·89
19	294·15	23·22	310·45	24·44	328·37	25·67	347·92	26·89	369·11	28·11
20	325·93	24·44	343·04	25·67	361·78	26·89	382·15	28·11	404·15	29·33
21	359·33	25·67	377·26	26·89	396·81	28·11	418·00	29·33	440·81	30·56
22	394·37	26·89	413·11	28·11	433·48	29·33	455·48	30·56	479·11	31·78
23	431·04	28·11	450·59	29·33	471·78	30·56	494·59	31·78	519·01	33·00
24	469·33	29·33	489·70	30·56	511·70	31·78	535·33	33·00	560·59	34·22
25	509·26	30·56	530·45	31·78	553·26	33·00	577·70	34·22	603·78	35·44
26	550·81	31·78	572·81	33·00	596·44	34·22	621·70	35·44	648·59	36·67
27	594·00	33·00	616·81	34·22	641·26	35·44	667·33	36·67	695·04	37·89
28	638·81	34·22	662·45	35·44	687·70	36·67	714·59	37·89	743·11	39·11
29	685·26	35·44	709·70	36·67	735·78	37·89	763·48	39·11	792·81	40·33
30	733·33	36·67	758·59	37·89	785·48	39·11	814·00	40·33	841·15	41·56
31	783·04	37·89	809·11	39·11	836·81	40·33	866·15	41·56	897·11	42·78
32	834·37	39·11	861·26	40·33	889·78	41·56	919·92	42·78	951·70	44·00
33	887·33	40·33	915·04	41·56	944·37	42·78	975·33	44·00	1007·92	45·22
34	941·93	41·56	970·45	42·78	1000·59	44·00	1032·37	45·22	1065·78	46·44
35	998·15	42·78	1027·48	44·00	1058·44	45·22	1091·04	46·44	1125·26	47·67
36	1056·00	44·00	1086·15	45·22	1117·93	46·44	1151·33	47·67	1186·37	48·89
37	1115·48	45·22	1146·45	46·44	1179·04	47·67	1213·26	48·89	1249·11	50·11
38	1176·59	46·44	1208·37	47·67	1241·78	48·89	1276·81	50·11	1313·48	51·33
39	1239·33	47·67	1271·93	48·89	1306·15	50·11	1342·00	51·33	1379·48	52·56
40	1303·70	48·89	1337·11	50·11	1372·15	51·33	1408·81	52·56	1447·11	53·78

H	h = 5 $\overline{V}$	5 V	6 $\overline{V}$	6 V	7 $\overline{V}$	7 V	8 $\overline{V}$	8 V	9 $\overline{V}$	9 V
5	61·11	12·22								
6	74·15	13·44	88·00	14·67						
7	88·81	14·67	103·48	15·89	119·78	17·11				
8	105·11	15·89	120·59	17·11	137·70	18·33	156·44	19·56		
9	123·04	17·11	139·33	18·33	157·25	19·56	176·81	20·78	198·00	22·00
10	142·59	18·33	159·70	19·56	178·44	20·78	198·81	22·00	220·81	23·22
11	163·78	19·56	181·70	20·78	201·26	22·00	222·44	23·22	245·26	24·44
12	186·59	20·78	205·33	22·00	225·70	23·22	247·70	24·44	271·33	25·67
13	211·04	22·00	230·59	23·22	251·78	24·44	274·59	25·67	299·04	26·89
14	237·11	23·22	257·48	24·44	279·48	25·67	303·11	26·89	328·37	28·11
15	264·81	24·44	286·00	25·67	308·81	26·89	333·26	28·11	359·33	29·33
16	294·15	25·67	316·15	26·89	339·78	28·11	365·03	29·33	391·92	30·56
17	325·11	26·89	347·92	28·11	372·37	29·33	398·44	30·56	426·15	31·78
18	357·70	28·11	381·33	29·33	406·59	30·56	433·48	31·78	462·00	33·00
19	391·92	29·33	416·37	30·56	442·44	31·78	470·15	33·00	499·48	34·22
20	427·78	30·56	453·04	31·78	479·92	33·00	508·44	34·22	538·59	35·44
21	465·26	31·78	491·33	33·00	519·04	34·22	548·37	35·44	579·33	36·67
22	504·37	33·00	531·26	34·22	559·78	35·44	589·92	36·67	621·70	37·89
23	545·11	34·22	572·81	35·44	602·15	36·67	633·11	37·89	665·70	39·11
24	587·48	35·44	616·00	36·67	646·15	37·89	677·92	39·11	711·33	40·33
25	631·48	36·67	660·81	37·89	691·78	39·11	724·37	40·33	758·59	41·56
26	677·11	37·89	707·26	39·11	739·04	40·33	772·44	41·56	807·48	42·78
27	724·37	39·11	755·33	40·33	787·92	41·56	822·15	42·78	858·00	44·00
28	773·26	40·33	805·04	41·56	838·44	42·78	873·48	44·00	910·15	45·22
29	823·78	41·56	856·37	42·78	890·59	44·00	926·44	45·22	963·92	46·44
30	875·92	42·78	909·33	44·00	944·37	45·22	981·04	46·44	1019·33	47·67
31	929·70	44·00	963·92	45·22	999·78	46·44	1037·26	47·67	1076·37	48·89
32	985·11	45·22	1020·15	46·44	1056·82	47·67	1095·11	48·89	1135·04	50·11
33	1042·15	46·44	1078·00	47·67	1115·48	48·89	1154·59	50·11	1195·33	51·33
34	1100·81	47·67	1137·48	48·89	1175·78	50·11	1215·70	51·33	1257·26	52·56
35	1161·11	48·89	1198·59	50·11	1237·70	51·33	1278·44	52·56	1320·81	53·78
36	1223·04	50·11	1261·33	51·33	1301·26	52·56	1342·82	53·78	1386·00	55·00
37	1286·59	51·33	1325·70	52·56	1366·44	53·78	1408·82	55·00	1452·81	56·22
38	1357·78	52·56	1391·70	53·78	1433·26	55·00	1476·44	56·22	1521·26	57·44
39	1418·59	53·78	1459·33	55·00	1501·70	56·22	1545·70	57·44	1591·33	58·67
40	1487·04	55·00	1528·59	56·22	1571·78	57·44	1616·59	58·67	1663·03	59·89

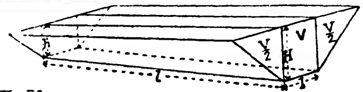
TABLE I. FOR CALCULATING EARTHWORK—(*continued*).

Slopes 1 to 1. Base = 1 foot. l = 66 feet = 1 chain. H and h in feet. V̄ and V in cubic yards.

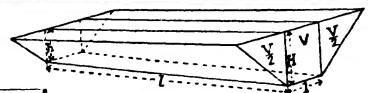
H	h = 10 V̄	10 V	11 V̄	11 V	12 V̄	12 V	13 V̄	13 V	14 V̄	14 V	15 V̄	15 V	16 V̄	16 V	17 V̄	17 V	18 V̄	18 V	19 V̄	19 V
10	244·44	24·44																		
11	269·70	25·67	295·78	26·89																
12	296·59	26·89	323·48	28·11	352·00	29·33														
13	325·11	28·11	352·81	29·33	382·15	30·56	413·11	31·78												
14	355·26	29·33	383·78	30·56	413·92	31·78	445·70	33·00	479·11	34·22										
15	387·04	30·56	410·37	31·78	447·33	33·00	479·92	34·22	514·15	35·44	550·00	36·67								
16	420·44	31·78	450·59	33·00	482·37	34·22	515·78	35·44	550·82	36·67	587·48	37·89	625·78	39·11						
17	455·48	33·00	486·44	34·22	519·04	35·44	553·26	36·67	589·11	37·89	626·59	39·11	665·70	40·33	706·44	41·56				
18	492·15	34·22	523·92	35·44	557·33	36·67	592·37	37·89	620·04	39·11	667·33	40·33	707·26	41·56	748·81	42·78	792·00	44·00		
19	530·44	35·44	563·04	36·67	597·26	37·89	633·11	39·11	670·59	40·33	709·70	41·56	750·44	42·78	792·81	44·00	836·81	45·22	882·44	46·44
20	570·37	36·67	603·78	37·89	638·81	39·11	675·48	40·33	713·78	41·56	753·70	42·78	795·26	44·00	838·44	45·22	883·26	46·44	920·70	47·67
21	611·92	37·89	646·15	39·11	682·00	40·33	719·48	41·56	758·59	42·78	799·33	44·00	841·70	45·22	885·70	46·44	931·33	47·67	978·59	48·89
22	655·11	39·11	690·15	40·33	726·81	41·56	765·11	42·78	805·04	44·00	846·59	45·22	889·78	46·44	934·59	47·67	981·04	48·89	1029·11	50·11
23	699·92	40·33	735·78	41·56	773·26	42·78	812·37	44·00	853·11	45·22	895·48	46·44	939·48	47·67	985·11	48·89	1032·37	50·11	1081·26	51·33
24	746·37	41·56	783·04	42·78	821·33	44·00	861·26	45·22	902·81	46·44	946·00	47·67	990·81	48·89	1037·26	50·11	1085·33	51·33	1135·04	52·56
25	794·44	42·78	831·92	44·00	871·04	45·22	911·78	46·44	954·15	47·67	998·15	48·89	1043·78	50·11	1091·04	51·33	1139·92	52·56	1190·44	53·78
26	844·15	44·00	882·44	45·22	922·37	46·44	963·92	47·67	1007·11	48·89	1051·92	50·11	1098·37	51·33	1146·44	52·56	1196·15	53·78	1247·48	55·00
27	895·48	45·22	934·59	46·44	975·33	47·67	1017·70	48·89	1061·70	50·11	1107·33	51·33	1154·59	52·56	1203·48	53·78	1254·00	55·00	1300·15	56·22
28	948·44	46·44	988·37	47·67	1029·92	48·89	1073·11	50·11	1117·92	51·33	1164·37	52·56	1212·44	53·78	1262·15	55·00	1313·48	56·22	1366·44	57·44
29	1003·04	47·67	1043·78	48·89	1086·15	50·11	1130·15	51·33	1175·78	52·56	1223·04	53·78	1271·92	55·00	1322·44	56·22	1374·59	57·44	1428·37	58·67
30	1059·26	48·89	1100·82	50·11	1144·00	51·33	1188·82	52·56	1235·26	53·78	1283·33	55·00	1333·04	56·22	1384·37	57·44	1437·33	58·67	1491·92	59·89
31	1117·11	50·11	1159·48	51·33	1203·48	52·56	1249·11	53·78	1296·37	55·00	1345·26	56·22	1395·78	57·44	1447·92	58·67	1501·70	59·89	1557·11	61·11
32	1176·59	51·33	1219·78	52·56	1264·59	53·78	1311·04	55·00	1359·11	56·22	1408·82	57·44	1460·15	58·67	1513·11	59·89	1567·70	61·11	1623·92	62·33
33	1237·70	52·56	1281·70	53·78	1327·33	55·00	1374·59	56·22	1423·48	57·44	1474·00	58·67	1526·15	59·89	1579·92	61·11	1635·33	62·33	1692·37	63·56
34	1300·44	53·78	1345·26	55·00	1391·70	56·22	1439·78	57·44	1489·48	58·67	1540·82	59·89	1593·78	61·11	1648·37	62·33	1704·59	63·56	1762·44	64·78
35	1364·82	55·00	1410·44	56·22	1457·70	57·44	1506·59	58·67	1557·11	59·89	1609·26	61·11	1663·04	62·33	1718·44	63·56	1775·48	64·78	1834·15	66·00
36	1430·82	56·22	1477·26	57·44	1525·33	58·67	1575·04	59·89	1626·37	61·11	1679·33	62·33	1733·92	63·56	1790·15	64·78	1848·00	66·00	1907·48	67·22
37	1498·44	57·44	1545·70	58·67	1594·59	59·89	1645·11	61·11	1697·26	62·33	1751·04	63·56	1806·44	64·78	1863·48	66·00	1922·15	67·22	1982·44	68·44
38	1567·70	58·67	1615·78	59·89	1665·48	61·11	1716·82	62·33	1769·78	63·56	1824·37	64·78	1880·59	66·00	1938·44	67·22	1997·92	68·44	2059·04	69·67
39	1638·59	59·89	1687·48	61·11	1738·00	62·33	1789·15	63·56	1843·92	64·78	1899·33	66·00	1956·37	67·22	2015·04	68·44	2075·83	69·67	2137·26	70·89
40	1711·11	61·11	1760·92	62·33	1812·15	63·56	1865·11	64·78	1919·70	66·00	1975·92	67·22	2033·78	68·44	2093·26	69·07	2154·37	70·89	2217·11	72·11

TABLE I. FOR CALCULATING EARTHWORK—(*continued*).

Slopes 1 to 1. Base = 1 foot. l = 66 feet = 1 chain. H and h in feet. V and V in cubic yards.



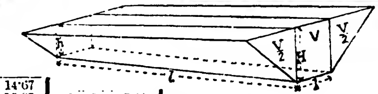
H	h = 20		21		22		23		24		25		26		27		28		29	
	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V
20	977·78	48·89																		
21	1027·48	50·11	1078·00	51·33																
22	1078·82	51·33	1130·15	52·56	1183·11	53·78														
23	1131·78	52·56	1183·92	53·78	1237·70	55·00	1293·11	56·22												
24	1186·37	53·78	1239·33	55·00	1293·92	56·22	1350·15	57·44	1408·00	58·67										
25	1242·59	55·00	1296·37	56·22	1351·78	57·44	1408·82	58·67	1467·48	59·89	1527·78	61·11								
26	1300·44	56·22	1355·04	57·44	1411·26	58·67	1469·11	59·89	1528·59	61·11	1589·70	62·33	1652·44	63·56						
27	1359·92	57·44	1415·33	58·67	1472·37	59·89	1531·04	61·11	1591·33	62·33	1653·26	63·56	1716·82	64·78	1782·00	66·00				
28	1421·04	58·67	1477·26	59·89	1535·11	61·11	1594·59	62·33	1655·70	63·56	1718·44	64·78	1782·82	66·00	1848·82	67·22	1916·44	68·44		
29	1488·78	59·89	1540·81	61·11	1599·48	62·33	1659·78	63·56	1721·70	64·78	1785·26	66·00	1850·44	67·22	1917·26	68·44	1985·70	69·67	2055·78	70·89
30	1548·15	61·11	1606·00	62·33	1665·48	63·56	1726·59	64·78	1789·33	66·00	1853·70	67·22	1919·70	68·44	1987·33	69·67	2056·59	70·89	2127·48	72·11
31	1614·15	62·33	1672·82	63·56	1733·11	64·78	1795·04	66·00	1858·59	67·22	1923·78	68·44	1990·59	69·67	2059·04	70·89	2129·11	72·11	2200·82	73·33
32	1681·78	63·56	1741·26	64·78	1802·37	66·00	1865·11	67·22	1929·48	68·44	1995·48	69·67	2063·11	70·89	2132·37	72·11	2203·26	73·33	2275·78	74·56
33	1751·04	64·78	1811·33	66·00	1873·26	67·22	1936·82	68·44	2002·00	69·67	2068·82	70·89	2137·26	72·11	2207·33	73·33	2279·04	74·56	2352·37	75·78
34	1821·92	66·00	1883·04	67·22	1945·78	68·44	2010·15	69·67	2076·15	70·89	2143·78	72·11	2213·04	73·33	2283·93	74·56	2356·44	75·78	2430·59	77·00
35	1894·44	67·22	1956·37	68·44	2019·92	69·67	2085·11	70·89	2151·93	72·11	2220·37	73·33	2290·44	74·56	2362·15	75·78	2435·48	77·00	2510·45	78·22
36	1968·59	68·44	2031·33	69·67	2095·70	70·89	2161·70	72·11	2229·33	73·33	2298·59	74·56	2369·48	75·78	2442·00	77·00	2516·15	78·22	2591·93	79·44
37	2044·37	69·67	2107·92	70·89	2173·11	72·11	2239·93	73·33	2308·37	74·56	2378·44	75·78	2450·15	77·00	2523·48	78·22	2598·45	79·44	2675·04	80·67
38	2121·78	70·89	2186·15	72·11	2252·15	73·33	2319·78	74·56	2389·04	75·78	2459·93	77·00	2532·45	78·22	2606·59	79·44	2682·37	80·67	2759·78	81·89
39	2200·82	72·11	2266·00	73·33	2332·82	74·56	2401·26	75·78	2471·33	77·00	2543·04	78·22	2616·37	79·44	2691·33	80·67	2767·93	81·89	2846·15	83·11
40	2281·48	73·33	2347·48	74·56	2415·11	75·78	2484·37	77·00	2555·26	78·22	2627·78	79·44	2701·93	80·67	2777·70	81·89	2855·11	83·11	2934·15	84·33



H	h = 30		31		32		33		34		35		36		37		38		39	
	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V	V
30	2200·00	73·33																		
31	2274·15	74·56	2349·11	75·78																
32	2349·93	75·78	2425·70	77·00	2503·11	78·22														
33	2427·33	77·00	2503·93	78·22	2582·15	79·44	2662·00	80·67												
34	2506·37	78·22	2583·78	79·44	2662·82	80·67	2743·48	81·89	2827·78	83·11										
35	2587·04	79·44	2665·26	80·67	2745·11	81·89	2826·59	83·11	2909·70	84·33	2994·45	85·56								
36	2669·33	80·67	2748·37	81·89	2829·04	83·11	2911·33	84·33	2995·26	85·56	3080·82	86·78	3168·00	88·00						
37	2753·26	81·89	2833·11	83·11	2914·59	84·33	2997·70	85·56	3082·45	86·78	3168·82	88·00	3256·82	89·22	3346·45	90·44				
38	2838·81	83·11	2919·48	84·33	3001·78	85·56	3085·70	86·78	3171·26	88·00	3258·45	89·22	3347·26	90·44	3437·70	91·67	3520·78	92·89		
39	2926·00	84·33	3007·48	85·56	3090·59	86·78	3175·33	88·00	3261·70	89·22	3349·70	90·44	3439·33	91·67	3530·59	92·89	3623·48	94·11	3718·00	95·33
40	3014·81	85·56	3097·11	86·78	3181·04	88·00	3266·59	89·22	3353·78	90·44	3442·59	91·67	3533·04	92·89	3625·11	94·11	3718·81	95·33	3814·15	96·56

TABLE II. FOR CALCULATING EARTHWORK.

Slopes $1\frac{1}{2}$ to 1. Base = 1 foot. l = 66 feet = 1 chain. H and h in feet. V and V in cubic yards.

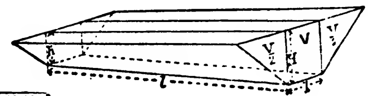


H	h = 0 V	0 V	1 V	1 V	2 V	2 V	3 V	3 V	4 V	4 V
1	1·22	1·22	3·67	2·44						
2	4·89	2·44	8·56	3·67	14·67	4·89				
3	11·00	3·67	15·89	4·89	23·22	6·11	33·00	7·33		
4	19·56	4·89	25·67	6·11	34·22	7·33	45·22	8·56	58·67	9·78
5	30·56	6·11	37·89	7·33	47·67	8·56	59·89	9·78	74·56	11·00
6	44·00	7·33	52·56	8·56	63·56	9·78	77·00	11·00	92·89	12·22
7	59·89	8·56	69·67	9·78	81·89	11·00	95·56	12·22	113·67	13·44
8	78·22	9·78	89·22	11·00	102·67	12·22	118·56	13·44	136·89	14·67
9	99·00	11·00	111·22	12·22	125·89	13·44	143·00	14·67	162·56	15·89
10	122·22	12·22	135·67	13·44	151·56	14·67	169·89	15·89	190·67	17·11
11	147·89	13·44	162·56	14·67	179·67	15·89	199·22	17·11	221·22	18·33
12	176·00	14·67	191·89	15·89	210·22	17·11	231·00	18·33	254·22	19·56
13	206·56	15·89	223·67	17·11	243·22	18·33	265·22	19·56	289·67	20·78
14	239·56	17·11	257·89	18·33	278·67	19·56	301·89	20·78	327·56	22·00
15	275·00	18·33	294·56	19·56	316·56	20·78	341·00	22·00	367·89	23·22
16	312·89	19·56	333·67	20·78	356·89	22·00	382·56	23·22	410·67	24·44
17	358·22	20·78	375·22	22·00	390·67	23·22	426·56	24·44	455·89	25·67
18	396·00	22·00	419·22	23·22	444·89	24·44	473·00	25·67	503·56	26·89
19	441·22	23·22	465·67	24·44	492·56	25·67	521·89	26·89	553·67	28·11
20	488·89	24·44	514·56	25·67	542·67	26·89	573·22	28·11	606·22	29·33
21	539·00	25·67	565·89	26·89	595·22	28·11	627·00	29·33	661·22	30·56
22	591·56	26·89	619·67	28·11	650·22	29·33	682·22	30·56	718·67	31·78
23	646·56	28·11	675·89	29·33	707·67	30·56	741·89	31·78	778·56	33·00
24	704·00	29·33	734·56	30·56	767·56	31·78	803·00	33·00	840·89	34·22
25	763·89	30·56	795·67	31·78	829·89	33·00	866·56	34·22	905·67	35·44
26	826·22	31·78	859·22	33·00	894·67	34·22	932·56	35·44	972·89	36·67
27	891·00	33·00	925·22	34·22	961·89	35·44	1001·00	36·67	1042·56	37·89
28	958·22	34·22	993·67	35·44	1031·56	36·67	1071·89	37·89	1114·67	39·11
29	1027·89	35·44	1064·56	36·67	1103·67	37·89	1145·22	39·11	1189·22	40·33
30	1100·00	36·67	1137·89	37·89	1178·22	39·11	1221·00	40·33	1266·22	41·56
31	1174·56	37·89	1213·67	39·11	1255·22	40·33	1299·22	41·56	1345·67	42·78
32	1251·56	39·11	1291·89	40·33	1334·67	41·56	1379·89	42·78	1427·55	44·00
33	1331·00	40·33	1372·56	41·56	1416·56	42·78	1463·00	44·00	1511·89	45·22
34	1412·89	41·56	1455·67	42·78	1500·89	44·00	1548·56	45·22	1598·67	46·44
35	1497·22	42·78	1541·22	44·00	1587·67	45·22	1636·56	46·44	1687·89	47·67
36	1584·00	44·00	1629·22	45·22	1676·89	46·14	1727·00	47·67	1779·56	48·89
37	1672·22	45·22	1719·67	46·44	1768·56	47·67	1819·89	48·89	1873·67	50·11
38	1764·89	46·44	1812·56	47·67	1862·67	48·89	1915·22	50·11	1970·22	51·33
39	1859·00	47·67	1907·89	48·89	1959·22	50·11	2013·00	51·33	2069·22	52·56
40	1955·55	48·89	2005·67	50·11	2058·22	51·33	2113·22	52·56	2170·67	53·78

H	h = 5 V	5 V	6 V	6 V	7 V	7 V	8 V	8 V	9 V	9 V
5	91·67	12·22								
6	111·22	13·44	132·00	14·67						
7	133·22	14·67	155·22	15·89	179·67	17·11				
8	157·67	15·89	180·89	17·11	206·56	18·33	234·67	19·56		
9	184·56	17·11	209·00	18·33	235·89	19·56	265·22	20·78	297·00	22·00
10	213·89	18·33	239·56	19·56	267·67	20·78	298·22	22·00	337·22	23·22
11	245·67	19·56	272·56	20·78	301·89	22·00	333·67	23·22	367·89	24·44
12	279·89	20·78	308·00	22·00	338·56	23·22	371·56	24·44	407·00	25·67
13	316·56	22·00	345·89	23·22	377·67	24·44	411·89	25·67	448·56	26·89
14	355·67	23·22	386·22	24·44	419·22	25·67	454·67	26·89	492·56	28·11
15	397·22	24·44	429·00	25·67	463.22	26·89	499·89	28·11	539·00	29·33
16	441·22	25·67	474·22	26·89	509·67	28·11	547·54	29·33	587·89	30·56
17	487·67	26·89	521·88	28·11	558·56	29·33	597·67	30·56	639·22	31·78
18	536·56	28·11	572·00	29·33	609·89	30·56	650·22	31·78	693·00	33·00
19	587·89	29·33	624·56	30·56	663·67	31·78	705·22	33·00	749·22	34·22
20	641·67	30·56	679·56	31·78	719·89	33·00	762·67	34·22	807·89	35·44
21	697·89	31·78	737·00	33·00	778·56	34·22	822·56	35·44	869·00	36·67
22	756·56	33·00	796·89	34·22	839·67	35·44	884·88	36·67	932·56	37·89
23	817·67	34·22	859·22	35·44	903·22	36·67	949·67	37·89	998·56	39·11
24	881·22	35·44	924·00	36·67	969·22	37·89	1016·88	39·11	1067·00	40·33
25	947·22	36·67	991·22	37·89	1037·67	39·11	1086·56	40·33	1137·89	41·56
26	1015·67	37·89	1060·89	39·11	1108·56	40·33	1158·67	41·56	1211·22	42·78
27	1086·56	39·11	1133·00	40·33	1181·88	41·56	1233·22	42·78	1287·00	44·00
28	1159·89	40·33	1207·56	41·56	1257·67	42·78	1310·22	44·00	1365·22	45·22
29	1235·67	41·56	1284·56	42·78	1335·89	44·00	1389·67	45·22	1445·89	46·44
30	1313·89	42·78	1364·00	44·00	1416·56	45·22	1471·56	46·44	1529·00	47·67
31	1394·56	44·00	1445·89	45·22	1499·67	46·44	1555·89	47·67	1614·56	48·89
32	1477·67	45·22	1530·22	46·44	1585·22	47·67	1642·67	48·89	1697·56	50·11
33	1563·22	46·44	1617·00	47·67	1673·22	48·89	1731·89	50·11	1793·00	51·33
34	1651·22	47·67	1701·22	48·89	1763·67	50·11	1823·56	51·33	1885·89	52·56
35	1741·67	48·89	1797·89	50·11	1856·58	51·33	1917·67	52·56	1981·22	53·78
36	1834·56	50·11	1892·00	51·33	1951·89	52·56	2014·22	53·78	2079·00	55·00
37	1929·89	51·33	1988·56	52·56	2049·67	53·78	2113·22	55·00	2179·22	56·22
38	2027·66	52·56	2087·56	53·78	2149·89	55·00	2214·67	56·22	2281·89	57·44
39	2127·89	53·78	2189·00	55·00	2252·56	56·22	2318·56	57·44	2387·00	58·67
40	2230·56	55·00	2292·89	56·22	2357·67	57·44	2424·89	58·67	2494·56	59·89

TABLE II. FOR CALCULATING EARTHWORK—(*continued*).

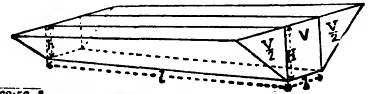
Slopes $1\frac{1}{2}$ to 1. Base=1 foot. l=66 feet=1 chain. H and h in feet. V and V in cubic yards.



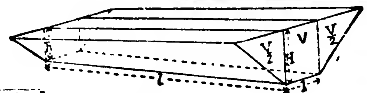
H	h = 10 V	10 V	11 V	11 V	12 V	12 V	13 V	13 V	14 V	14 V	15 V	15 V	16 V	16 V	17 V	17 V	18 V	18 V	19 V	19 V
10	366·67	24·44																		
11	404·56	25·67	443·67	26·89																
12	444·89	26·89	485·22	28·11	528·00	29·33														
13	487·67	28·11	529·22	29·33	574·22	30·56	619·67	31·78												
14	532·89	29·33	575·67	30·56	620·89	31·78	668·56	33·00	718·67	34·22										
15	580·56	30·56	624·56	31·78	671·00	33·00	719·89	34·22	771·22	35·44	825·00	36·67								
16	630·67	31·78	675·89	33·00	723·56	34·22	773·67	35·44	826·22	36·67	881·22	37·89	968·67	39·11						
17	683·22	33·00	729·67	34·22	778·56	35·44	829·89	36·67	883·67	37·89	939·88	39·11	998·56	40·33	1059·67	41·56				
18	738·22	34·22	785·89	35·44	835·00	36·67	888·56	37·89	943·56	39·11	1001·00	40·33	1060·89	41·56	1123·22	42·78	1188·00	44·00		
19	795·67	35·44	844·56	36·67	893·89	37·89	949·67	39·11	1005·89	40·33	1064·56	41·56	1125·67	42·78	1189·22	44·00	1255·22	45·22	1323·67	46·44
20	855·56	36·67	905·67	37·89	958·22	39·11	1013·22	40·33	1070·67	41·56	1130·56	42·78	1192·89	44·00	1257·67	45·22	1324·89	46·44	1494·56	47·67
21	917·89	37·89	969·22	39·11	1023·00	40·33	1079·22	41·56	1137·89	42·78	1199·00	44·00	1262·56	45·22	1328·56	46·44	1397·00	47·67	1567·89	48·89
22	982·67	39·11	1035·22	40·33	1090·22	41·56	1147·67	42·78	1207·56	44·00	1269·89	45·22	1334·67	46·44	1401·89	47·67	1471·56	48·89	1643·67	50·11
23	1049·89	40·33	1103·67	41·56	1159·89	42·78	1218·56	44·00	1279·67	45·22	1343·22	46·44	1409·22	47·67	1477·67	48·89	1548·56	50·11	1721·89	51·33
24	1119·56	41·56	1174·56	42·78	1232·00	44·00	1291·89	45·22	1354·22	46·44	1419·00	47·67	1486·22	48·89	1555·89	50·11	1628·00	51·33	1802·56	52·56
25	1191·67	42·78	1247·89	44·00	1306·56	45·22	1367·67	46·44	1431·22	47·67	1497·22	48·89	1565·67	50·11	1636·56	51·33	1709·89	52·56	1885·67	53·78
26	1263·22	44·00	1323·67	45·22	1383·56	46·44	1445·89	47·67	1510·67	48·89	1577·89	50·11	1647·56	51·33	1719·67	52·56	1794·22	53·78	1971·22	55·00
27	1343·22	45·22	1401·89	46·44	1463·00	47·67	1525·56	48·89	1592·56	50·11	1661·00	51·33	1731·89	52·56	1805·22	53·78	1881·00	55·00	2059·22	56·22
28	1422·67	46·44	1482·56	47·67	1544·89	48·89	1609·67	50·11	1676·89	51·33	1746·56	52·56	1818·67	53·78	1893·22	55·00	1970·22	56·22	2149·67	57·44
29	1504·56	47·67	1565·67	48·89	1629·22	50·11	1695·22	51·33	1763·67	52·56	1834·56	53·78	1907·89	55·00	1983·67	56·22	2061·89	57·44	2242·36	58·67
30	1588·89	48·89	1651·22	50·11	1716·00	51·33	1783·22	52·56	1852·89	53·78	1925·00	55·00	1999·56	56·22	2076·56	57·44	2156·00	58·67	2337·89	59·89
31	1675·67	50·11	1739·22	51·33	1805·22	52·56	1873·67	53·78	1944·56	55·00	2017·89	56·22	2093·67	57·44	2171·89	58·61	2252·56	59·89	2435·67	61·11
32	1764·89	51·33	1829·67	52·56	1896·89	53·78	1966·56	55·00	2038·67	56·22	2113·22	57·44	2190·22	58·67	2269·67	59·89	2351·56	61·11	2535·89	62·33
33	1856·56	52·56	1922·56	53·78	1991·00	55·00	2061·89	56·22	2135·22	57·44	2211·00	58·67	2289·22	59·89	2369·89	61·11	2453·00	62·33	2638·56	63·56
34	1950·67	53·78	2017·89	55·00	2087·56	56·22	2159·67	57·44	2234·22	58·67	2311·22	59·89	2390·67	61·11	2472·56	62·33	2556·89	63·56	2743·67	64·78
35	2047·22	55·00	2115·67	56·22	2186·56	57·44	2259·89	58·67	2335·67	59·89	2413·89	61·11	2494·56	62·33	2577·67	63·56	2663·22	64·78	2851·22	66·00
36	2146·22	56·22	2215·89	57·44	2288·00	58·67	2362·56	59·89	2442·00	61·11	2519·00	62·33	2600·98	63·56	2685·22	64·78	2772·00	66·00	2961·22	67·22
37	2247·67	57·44	2318·56	58·67	2391·89	59·89	2467·67	61·11	2545·89	62·33	2626·56	63·56	2709·67	64·78	2795·22	66·00	2883·22	67·22	3073·67	68·44
38	2351·56	58·67	2423·67	59·89	2498·22	61·11	2575·22	62·33	2654·66	63·56	2736·56	64·78	2820·89	66·00	2907·67	67·22	2996·89	68·44	3189·56	69·67
39	2457·89	59·89	2531·22	61·11	2607·00	62·33	2685·22	63·56	2765·89	64·78	2849·00	66·00	2934·56	67·22	3022·56	68·44	3113·00	69·67	3305·89	70·89
40	2566·67	61·11	2641·22	62·33	2718·22	63·56	2797·67	64·78	2879·56	66·00	2963·89	67·22	3050·67	68·44	3189·89	69·67	3231·56	70·89	3425·67	72·11

TABLE II. FOR CALCULATING EARTHWORK—(*continued*).

Slopes $1\frac{1}{2}$ to 1. Base=1 foot. l=66 feet=1 chain. H and h in feet. V and **V** in cubic yards.



H	h = 20 V	20 **V**	21 V	21 **V**	22 V	22 **V**	23 V	23 **V**	24 V	24 **V**	25 V	25 **V**	26 V	26 **V**	27 V	27 **V**	28 V	28 **V**	29 V	29 **V**
20	1466·67	48·89																		
21	1541·22	50·11	1617·00	51·33																
22	1618·22	51·33	1695·22	52·56	1774·67	53·78														
23	1697·67	52·56	1775·89	53·78	1856·56	55·00	1939·67	56·22												
24	1779·56	53·78	1859·00	55·00	1940·89	56·22	2025·22	57·44	2112·00	58·67										
25	1863·89	55·00	1944·56	56·22	2027·67	57·44	2113·22	58·67	2201·22	59·89	2291·67	61·11								
26	1950·67	56·22	2032·56	57·44	2116·89	58·67	2203·67	59·89	2292·89	61·11	2384·56	62·33	2478·67	63·56						
27	2039·89	57·44	2123·00	58·67	2208·56	59·89	2296·56	61·11	2387·00	62·33	2479·89	63·56	2575·22	64·78	2673·00	66·00				
28	2131·56	58·67	2215·89	59·89	2302·67	61·11	2391·89	62·33	2483·56	63·56	2577·67	64·78	2674·22	66·00	2778·22	67·22	2874·66	68·44		
29	2225·67	59·89	2311·22	61·11	2399·22	62·33	2489·67	63·56	2582·56	64·78	2677·89	66·00	2775·67	67·22	2875·88	68·44	2978·55	69·67	3088·66	70·89
30	2322·22	61·11	2409·00	62·33	2498·22	63·56	2589·89	64·78	2684·00	66·00	2780·56	67·22	2879·56	68·44	2980·99	69·67	3084·88	70·89	3191·22	72·11
31	2421·22	62·33	2509·22	63·56	2593·67	64·78	2692·56	66·00	2787·89	67·22	2885·67	68·44	2985·89	69·67	3088·55	70·89	3193·66	72·11	3301·22	73·33
32	2522·67	63·56	2611·89	64·78	2703·56	66·00	2797·67	67·22	2891·22	68·44	2993·22	69·67	3094·67	70·89	3198·55	72·11	3304·88	73·33	3413·66	74·56
33	2626·56	64·78	2717·00	66·00	2809·89	67·22	2905·22	68·44	3008·00	69·67	3103·22	70·89	3205·89	72·11	3310·99	73·33	3418·55	74·56	3528·55	75·78
34	2732·89	66·00	2824·56	67·22	2918·67	68·44	3015·22	69·67	3114·22	70·89	3215·67	72·11	3319·55	73·33	3425·88	74·56	3531·66	75·78	3645·88	77·00
35	2841·67	67·22	2931·56	68·44	3029·89	69·67	3127·67	70·89	3227·89	72·11	3330·55	73·33	3435·67	74·56	3543·22	75·78	3658·22	77·00	3765·66	78·22
36	2952·89	68·44	3047·00	69·67	3143·56	70·89	3242·55	72·11	3344·00	73·33	3447·89	74·56	3554·22	75·78	3662·99	77·00	3774·22	78·22	3887·88	79·44
37	3066·56	69·67	3161·89	70·89	3259·67	72·11	3359·89	73·33	3462·56	74·56	3567·66	75·78	3675·22	77·00	3785·22	78·22	3897·66	79·44	4012·55	80·67
38	3182·67	70·89	3269·22	72·11	3378·22	73·33	3479·67	74·56	3583·56	75·78	3689·88	77·00	3798·67	78·22	3909·88	79·44	4023·55	80·67	4139·66	81·89
39	3301·22	72·11	3399·00	73·33	3499·22	74·56	3600·89	75·78	3707·00	77·00	3814·55	78·22	3924·55	79·44	4066·99	80·67	4151·88	81·89	4269·21	83·11
40	3422·22	73·33	3521·22	74·56	3622·67	75·78	3725·55	77·00	3832·89	78·22	3941·66	79·44	4052·89	80·67	4166·55	81·89	4282·66	83·11	4401·21	84·33



H	h = 30 V	30 **V**	31 V	31 **V**	32 V	32 **V**	33 V	33 **V**	34 V	34 **V**	35 V	35 **V**	36 V	36 **V**	37 V	37 **V**	38 V	38 **V**	39 V	39 **V**
30	3300·00	73·33																		
31	3411·22	74·56	3523·67	75·78																
32	3524·89	75·78	3638·55	77·00	3754·67	78·22														
33	3640·99	77·00	3755·89	78·22	3873·22	79·44	3993·00	80·67												
34	3759·55	78·22	3875·67	79·44	3994·22	80·67	4115·22	81·89	4238·67	83·11										
35	3880·55	79·44	3997·89	80·67	4117·67	81·89	4239·89	83·11	4364·55	84·33	4491·67	85·56								
36	4003·96	80·67	4122·55	81·89	4243·55	83·11	4367·00	84·33	4492·89	85·56	4621·22	86·78	4752·00	88·00						
37	4129·[illegible]	81·89	4249·67	83·11	4371·89	84·33	4495·55	85·56	4623·67	86·78	4753·22	88·00	4885·22	89·22	5019·67	90·44				
38	4258·22	83·11	4379·22	84·33	4502·67	85·56	4628·55	86·78	4756·89	88·00	4887·67	89·22	5020·89	90·44	5156·55	91·67	5294·67	92·89		
39	4388·[illegible]	84·33	4511·22	85·56	4635·89	86·78	4763·00	88·00	4892·55	89·22	5024·55	90·44	5159·00	91·67	5295·89	92·89	5435·22	94·11	5577·00	95·33
40	4522·22	85·56	4645·67	86·78	4771·55	88·00	4899·89	89·22	5030·67	90·44	5163·89	91·67	5299·55	92·89	5437·67	94·11	5578·22	95·33	5721·22	96·56

Loads of Carts and Wagons.

An ordinary cart, $6' \times 3\frac{1}{2}' \times 2\frac{1}{2}'$ deep,	holds $1\frac{7}{8}$	cubic yards.
An earth wagon	,, 2	,,
,, ,, large . . .	,, 3	,,
A wheelbarrow, navvy . . .	,, $\frac{1}{10}$	,,
A 30-cwt. motor lorry . . .	,, 1	,,
A petrol tipping wagon . . .	,, 1 to 3	,,

Weight of a Cubic Yard of different Soils.

	cwts.		cwts.		cwts.		cwts.
Dry peat . .	$7\frac{1}{2}$	Common earth .	24	Common gravel .	27	Grey chalk . .	36
Wet peat . .	15	Sandy loam . .	24	Wet sand . . .	28	Sandstone . .	37
Top soil . .	20	Marl . . .	26	Gravelly clay .	30	Shale . . .	39
Dry sand . .	22	Clay . . .	27	Rough water gravel	34	Limestone . .	48

Barrow and Cart Runs.

The barrows used usually hold about one-tenth cubic yard, and are mostly run on planks, which should not be narrower than 11″, and 3″ thick, nor, as a rule, have a greater inclination than 1 in 12, unless aided by ropes or winding machinery. Box-horses (fig. 10) are the best supports on an incline, either singly or resting one on another; they are mostly from $1\frac{1}{2}'$ to 3′ square, made out of 11″ × 3″ planks. If a considerable height is required, trestles (fig. 11) are used, but boxhorses are stronger, and less likely to get out of repair.

A barrow run is 20 or 25 yards. Each foot of rise is considered equal to 6′ to 9′ additional run.

Wheeling is more economical than carting for distances under 100 yards.

An earth wagon holds as much as 20 or 30 wheelbarrows, and goes one-fifth faster—equal, therefore, to 24 or 36 barrows.

If loaded wagons have to be drawn up an ascent, and the temporary rails are in moderately good order, each foot of ascent may be considered equivalent to about 150 feet of additional horizontal distance.

BOXHORSE.

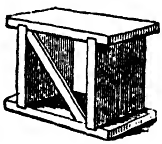


FIG. 10.

TRESTLE.

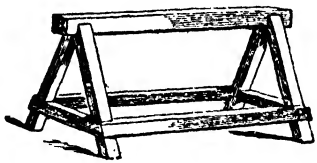


FIG. 11.

When the height exceeds 12 feet, a horse lift is better than wheeling.

For railway embankments and cuttings, locomotives are better than carting for distances over $1\frac{1}{2}$ miles.

Narrow Gauge Track and Tipping Wagons.

Whilst standard gauge track and large trucks are most suitable for railway work, narrow gauge track is often more serviceable for other engineering work for the following reasons:—

(1) It is more easily laid.

(2) It is more convenient for use in confined spaces where many changes of direction are needed.

(3) The trucks are of such a size as to be man-handled with ease when necessary.

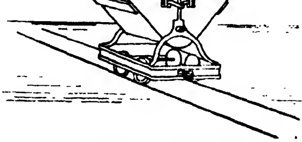


FIG. 12.—Tipping Wagon.

With light track the tipping hopper truck shown in fig. 12 is generally used. The gauge varies from 1 ft. 6 ins. to 2 ft. 6 ins. and the rails weigh from 16 to 20 lbs. a yard. The wagons hold from ½ yard to 3 yards and are of all metal construction.

Tip Wagons.

End Tip Wagons (figs. 13 and 14) will hold 2 yds. filled up to top of sides, as with gravel, 2½ yds. when heaped up with clay, &c., the most economical size. The lowness of sides is of great advantage in filling.

The sides, one end, and bottom of body are of elm 1½″ thick (the earth being confined at other end by movable tailboard), bound together with iron straps at angles, and with upright

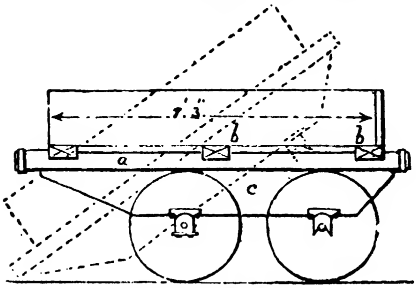


Fig. 13. Side Elevation.

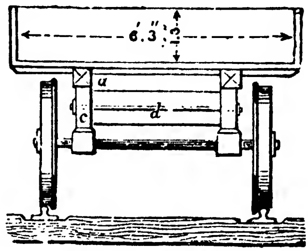


Fig. 14. Cross Section.

straps at sides and one end, all bolted through. The sole is the longitudinal piece, *a*, at each side, of elm, 5″ × 5″, upon which the bottom boards rest. At each end, and in middle of body, is a cross-piece of elm, *b*, 6″ × 3″, bolted down to the soles. The under soles, *c*, are of Memel, 12″ × 3″, between which from side to side are two distance-pieces, *d*, each 9″ × 3″, and alongside of each is a ¾″ bolt. The wheels are of cast iron, 2′ 6″ diam., with wrought-iron axles.

Weight of body of waggon, ¾ ton, of load, 3¼ tons; total load on axles, 4 tons. With well-greased axles, resistance will be about 10 lbs. per ton, but may be as high as 20. Resistance, 80 lbs. per waggon. A horse can draw 150 lbs. 2½ miles an hour for eight hours; a horse can therefore draw two waggons.

Temporary contractor's rails weigh from 25 lbs. to 30 lbs. per yard.

It is generally better to throw earth away than to lead it three miles.

Slopes of Excavations and Embankments.

Slopes of cuttings are often—

In gravel, 1 to 1 when not more than 20′ deep	**In clay, 2 to 1 when well drained**
„ 1½ „ 1 when greater depth	**„ 3 or 4 to 1 when wet**

Slopes of embankments are often—

In gravel, 1½ to 1 when not more than 30′ high	**In clay, 2 to 1 when not more than 40′ high**
„ 2 „ 1 when of greater height	**„ 2½ „ 1 when of greater height**

Chalk and rock will stand perpendicular, or with a slope of about ¼ base to 1 height.

In roads it is often better to make the sides of cuttings, especially on the south side, with greater slope than that they will stand at, to admit sun.

Earth in embankments should be laid in regular horizontal courses not more than 4′ thick. In high embankments they should be rather concave, to avoid slipping.

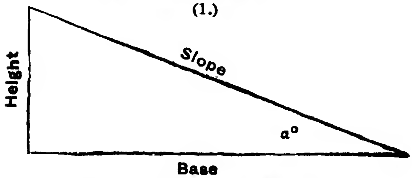
Form of Side Slopes.

The natural, strongest, and ultimate form of earth slopes is a concave curve, in which the flattest portion is at the bottom. This form is very rarely given to the slopes in constructing them; in fact, the reverse is often the case, the slopes being made convex, thus saving excavation for the contractor, and inviting slips.

In cuttings exceeding 10 feet in depth the formation of concave slopes will materially aid in preventing slips, and in any case it will reduce the amount of material which will eventually have to be removed when cleaning up. Straight or convex slopes will continue to slip until the natural form is attained.

Table of Angles and Lengths of Slopes.

(1.)



If Height = 1, then

$$\text{Base} = \cot a^\circ\ ;\ \text{Slope} = \sec a^\circ = \frac{1}{\cos a^\circ}.$$

Base and Slope in Terms of Angle.						Angle in Terms of Base and Slope.					
Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).	Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).	Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).	Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).
90°	·000	∞	45°	1·000	1·414	89° 25′	·01	98·223	17° 21′	3·2	1·048
89	·017	57·299	44	1·036	1·390	88 17	·03	33·381	16 52	3·3	1·045
88	·035	28·654	43	1·072	1·367	87 8	·05	19·995	16 23	3·4	1·042
87	·052	19·107	42	1·111	1·346	86	·07	14·336	15 57	3·5	1·040
86	·070	14·336	41	1·150	1·325	84 51	·09	11·140	15 31	3·6	1·038
85	·087	11·474	40	1·192	1·305	84 17	·1	10·039	15 8	3·7	1·036
84	·105	9·567	39	1·235	1·287	81 28	·15	6·739	14 45	3·8	1·034
83	·123	8·206	38	1·280	1·269	78 41	·2	5·096	14 23	3·9	1·032
82	·141	7·185	37	1·327	1·252	75 58	·25	4·124	14 2	4·0	1·031
81	·158	6·392	36	1·376	1·236	73 18	·3	3·480	13 42	4·1	1·030
80	·176	5·759	35	1·428	1·221	70 43	·35	3·028	13 24	4·2	1·028
79	·194	5·241	34	1·483	1·206	68 12	·4	2·693	13 6	4·3	1·027
78	·213	4·810	33	1·540	1·192	65 47	·45	2·438	12 49	4·4	1·026
77	·231	4·445	32	1·600	1·179	63 26	·5	2·236	12 32	4·5	1·024
76	·249	4·134	31	1·664	1·167	61 11	·55	2·075	12 16	4·6	1·023
75	·268	3·864	30	1·732	1·155	59 2	·6	1·943	12 1	4·7	1·022
74	·287	3·628	29	1·804	1·143	56 59	·65	1·835	11 46	4·8	1·021
73	·306	3·420	28	1·881	1·133	55 1	·7	1·744	11 32	4·9	1·021
72	·325	3·236	27	1·963	1·122	53 8	·75	1·667	11 19	5	1·020
71	·344	3·072	26	2·050	1·113	51 20	·8	1·601	10 19	5·5	1·016
70	·364	2·924	25	2·145	1·103	49 38	·85	1·544	9 28	6	1·014
69	·384	2·790	24	2·246	1·095	48 1	·9	1·495	8 45	6·5	1·012
68	·404	2·669	23	2·356	1·086	46 28	·95	1·452	8 8	7	1·010
67	·424	2·559	22	2·475	1·079	45 0	1	1·414	7 8	8	1·008
66	·445	2·459	21	2·605	1·071	42 16	1·1	1·351	6 20	9	1·006
65	·466	2·366	20	2·747	1·064	39 48	1·2	1·302	5 43	10	1·005
64	·488	2·281	19	2·904	1·058	37 34	1·3	1·262	5 12	11	1·004
63	·510	2·203	18	3·078	1·051	35 32	1·4	1·229	4 46	12	1·003
62	·532	2·130	17	3·271	1·046	33 41	1·5	1·202	4 24	13	1·003
61	·554	2·063	16	3·487	1·040	32	1·6	1·179	4 5	14	1·003
60	·577	2·000	15	3·732	1·036	30 28	1·7	1·160	3 49	15	1·002
59	·601	1·942	14	4·011	1·031	29 3	1·8	1·144	3 35	16	1·002
58	·625	1·887	13	4·331	1·026	27 46	1·9	1·130	3 22	17	1·002
57	·649	1·836	12	4·705	1·022	26 33	2	1·118	3 11	18	1·002
56	·675	1·788	11	5·145	1·019	25 28	2·1	1·108	3 1	19	1·001
55	·700	1·743	10	5·671	1·015	24 27	2·2	1·099	2 52	20	1·001
54	·727	1·701	9	6·314	1·012	23 30	2·3	1·090	2 18	25	1·001
53	·754	1·662	8	7·115	1·010	22 37	2·4	1·083	1 55	30	1·001
52	·781	1·624	7	8·144	1·008	21 48	2·5	1·077	1 38	35	1·000
51	·810	1·589	6	9·514	1·006	21 2	2·6	1·071	1 26	40	1·000
50	·839	1·556	5	11·430	1·004	20 19	2·7	1·066	1 9	50	1·000
49	·869	1·524	4	14·301	1·002	19 39	2·8	1·062	58	60	1·000
48	·900	1·494	3	19·081	1·001	19 2	2·9	1·058	49	70	1·000
47	·933	1·466	2	28·636	1·001	18 26	3	1·054	43	80	1·000
46	·966	1·440	1	57·290	1·000	17 53	3·1	1·051	38	90	1·000

Table of Angles and Lengths of Slopes.

(2.)

Base of Slope is given in terms of the Height; Length of Slope, in terms of the Base.

Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).	Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).	Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).	Angle.	Base of Slope (Height = 1).	Length of Slope (Base = 1).
° ′			° ′			° ′			° ′		
45 0	1	1·414	8 0	7·1	1·009	2 52	20	1·0012	0 43	80	1·0000
40 0	1·2	1·305	7 7	8	1·007	2 18	25	1·0008	0 40	86	„
38 40	1¼	1·280	7 0	8·1	1·007	2 12	26	1·0007	0 38	90	„
35 0	1·4	1·220	6 20	9	1·006	2 7	27	1·0007	0 36	95	„
33 41	1½	1·201	6 0	9·5	1·005	2 3	28	1·0006	0 34	101	„
30 0	1·7	1·154	5 42	10	1·005	2 0	28·6	1·0006	0 23	150	„
29 44	1¾	1·151	5 11	11	1·0041	1 59	29	1·0006	0 17	202	„
26 34	2	1·118	5 0	11·4	1·0038	1 54	30	1·0005	0 14	245	„
25 0	2·1	1·103	4 46	12	1·0034	1 38	35	1·0004	0 12	286	„
20 0	2·7	1·064	4 23	13	1·0029	1 26	40	1·0003	0 10	343	„
18 26	3	1·054	4 5	14	1·0025	1 16	45	1·0002	0 9	381	„
15 0	3·7	1·035	4 0	14·3	1·0024	1 9	50	1·0002	0 8	429	„
14 2	4	1·030	3 49	15	1·0022	1 2	55	1·0001	0 7	491	„
11 18	5	1·019	3 34	16	1·0019	1 0	57·3	1·0001	0 6	572	„
10 0	5·7	1·015	3 22	17	1·0017	0 57	60	1·0001	0 4	859	„
9 27	6	1·013	3 11	18	1·0015	0 53	65	1·0001	0 2	1718	„
9 0	6·3	1·012	3 1	19	1·0013	0 49	70	1·0001	0 1	3437	„
8 8	7	1·010	3 0	19	1·0013	0 46	75	1·0000	—	—	„

Angles of Repose of Soils.

Material.	Angle.	Ratio of Base of Slope to Height.
	°	
Clay, dry	29	1·8 to 1
„ damp, well drained	45	1 to 1
„ wet	16	3·5 to 1
Earth, dry	29	1·8 to 1
„ moist	45 to 49	1 to 1 to ·87 to 1
„ very wet	17	3·27 to 1
„ punned	66 to 74	·45 to 1 to ·28 to 1
Gravel, clean	48	·9 to 1
„ with sand	26	2 to 1
Sand, fine dry	37 to 31	1·3 to 1 to 1·6 to 1
„ wet	26	2 to 1
„ very wet	32	1·6 to 1
Shingle, loose	39	1·2 to 1
Peat	14 to 45	4 to 1 to 1 to 1

FOUNDATIONS.

In cases where there is any doubt as to the ground being of too soft a nature to bear the weight of the proposed structure, a trial pit should be sunk, or borings made with boring-rods; samples of the underlying strata may thus be brought up and examined.

Where a soft stratum overlies firm ground, the foundation should, if possible, be carried down to the firm ground; if this is not possible, piles may be driven to the solid ground and the building supported on them, or a number of piers of brickwork, masonry, or iron cylinders sunk.

Where ground is soft to a great depth, a wide trench filled with concrete may distribute the weight over a sufficiently large area to support it, or the trench may be filled with carefully rammed sand, prevented, by sheet piling or some other means, from escaping laterally. Another method is to drive a number of piles, which, by the friction of their sides, would bear the weight of the building.

Foundations on different Soils.

In any structure begun at different levels larger blocks should be used for the deeper parts, if the material admits of it, so as to reduce the number of mortar joints and the risks of unequal settlement.

With Sand or Gravel Foundation overlying Clay on a Slope, intercept water by drain on upper side of building.

Least Depth to escape Effects of Heat and Frost, from 3′ to 6′, according to climate. Not less than 3′ in England for ordinary soils, and 4′ for clay.

Rock.—Good foundation, but must be level; should be levelled, if necessary, in steps. The uneven parts should be filled up with large stones, firmly built with strong cement, or with concrete.

Gravel.—The best foundation.

Sand.—Good foundation, if dry and not liable to be washed away, but this easily occurs; drains with leaky joints may cause a subsidence, or any disturbance of the water-level in the stratum, whether by natural or artificial means, such as pumping operations connected with deep foundations, even at a great distance.

Clay.—Generally very treacherous and damp; the foundation must be deep.

Hard overlying Soft Ground.—If care is taken that the pressure per unit of area is not greater than the firm layer will bear, it may be wiser to build on it, sinking into it as little as possible.

Soft Ground overlying Hard Ground.—If the stratum of soft ground does not exceed 15′ to 20′ it will generally be cheaper to sink down to the firm ground; if not more than about 30′, drive piles or sink wells of masonry. If of indefinite depth, the platform must be supported by friction against the sides, and be therefore of considerable thickness.

Made Ground should never be trusted for the support of much weight, even though it may have lain undisturbed for years.

Chemical Treatment of Foundation Soils.—A method of increasing the bearing power of soft, sandy soils is said to have been employed with conspicuous success in connection with the construction of the Metropolitan Railway, Berlin. On a site composed of alluvial and diluvial sands, two solutions, one rich in silicic acid, the other containing soluble salts or acids, were successively injected under pressure into the soil. The first solution served solely to impregnate the soil, the second, by chemical reactions, causing solidification through the formation of silicates. The method is particularly applicable to soils containing acids and saline solutions, which exert a favourable influence on the chemical reactions involved. Its relative economic advantages have yet to be determined, the cost, principally attributable to the quantity of chemical material required, varying according to the site, and being comparatively lower with damp than with dry, sandy soils.

Loads which Foundations will Bear.

Safe Loads on Ordinary Foundations.

	Tons per sq. ft.		Tons per sq. ft.
1. Hard rock	12 upwards	8. Dry sand	2 to 3
2. Moderately hard rock (strength of engineering bricks)	9	9. ,, ,, with clay	2
3. Very soft rock	2	10. Wet or loose sand	1
4. Hard chalk	4	11. Hard clay	4
5. Soft chalk	1½	12. Ordinary clay	2
6. Marl and firm shale	6	13. Soft clay	1
7. Compact gravel	4	14. Alluvial soil	½ to 1

Intensity of pressure on a rock foundation should at no point exceed one-eighth pressure which would crush the rock.

Safe Loads on Materials.

	Tons per sq. ft.		Tons per sq. ft.
Portland cement concrete, 6, 3, 1 mix	10	Blue bricks in cement mortar, 3 to 1	12
,, ,, ,, 4, 2, 1 ,,	15	Granite	15
Common bricks in lime mortar	3½	Sandstone	12
,, ,, in cement mortar, 3 to 1	5	Limestone	9
London stock bricks in cement mortar, 3 to 1	8	Rubble in cement mortar, 3 to 1	4

Safe Load on Stone Walls and Columns.

	Ashlar Walls, single bedstones. Columns diameter = ⅛ height. Lbs. per sq. in.	Block in course. Columns, diameter = $\frac{1}{12}$ height. Lbs. per sq. in.
Granite	712	570
Hard stone	356	280
Medium	214	142

Safe Load on Brick Walls and Columns.

	Walls not less than 18 ins. Columns, diameter ⊲ $\frac{1}{6}$ height. Lbs. per sq. in.	Walls under 18 ins. Columns, diameter ⊲ $\frac{1}{6}$ height. Lbs. per sq. in.	Columns, diameter = $\frac{1}{6}$ to $\frac{1}{12}$ height. Lbs. per sq. in.
Brick in mortar	72	36	—
,, ,, cement	108	72	—
,, ,, Portland cement	142	108	44
Rubble mortar	58	—	—
,, cement	72	—	—
Pressed bricks in mortar	128	114	108
,, ,, ,, cement	172	142	114
,, ,, ,, Portland cement	100	—	—

Safe Loads on Floors.

London County Council Regulations.

In the area under the control of the London County Council floors must conform to the London Building Act (Amendment) Act, 1935, and the By-laws in connection therewith (see p. 318).

Piled Foundations.

There are two kinds of piled foundations:—

1. *When the soil is soft to a great depth*, area should be enclosed with sheet piling before main piles are driven, to consolidate the ground better. When the tops of the piles are connected by timber, cross-pieces (12 ins. × 9 ins. for 12-in. piles) running across breadth of foundation are first notched on to heads of piles, and longitudinals, or *string pieces*, 12 ins. × 9 ins., running parallel to length of foundation, over them. On these latter, 3-in. or 4-in. planking (better diagonal) is laid to carry masonry. In modern practice the piles are generally of reinforced concrete. After driving, the heads are broken away and the reinforcement bared. The beams and slabs are cast *in situ*, the reinforcement of the piles being incorporated with that of the beams to form a monolithic whole. The platform so formed may be part of the main structure or the latter may be built on it.

2. *When there is hard ground below.*—In this case, as each pile resists partly by resistance to buckling as a column, the cross-section should be greater in proportion to the depth. Maximum depth may be put at 30 ft. The number of piles must be proportional to weight carried.

TIMBER PILES.

Timber piles may be chosen from the following table according to the conditions under which they will be placed:—

(1) In sea-water where sea worms are to be resisted: Greenheart, jarrah, lignum vitae, Huon pine, African oak, teak, creosoted timbers.

(2) In the tropics where ants are to be resisted: Cedar, greenheart, ironwood, jarrah, lignum vitae, locust, Huon pine, pitch pine, teak.

(3) Where exposed to weather and alternately wet and dry: Cedar, cypress, greenheart, blue gum, hornbeam, ironwood, jarrah, larch, locust, oak, Huon pine, Kauri pine, northern pine, white poplar, teak.

(4) Where permanently wet: Beech, white cedar, Spanish chestnut, elm, larch, oak, Huon pine, Northern pine, American plane, teak.

(5) Durable only when kept dry: Ash, Norway fir, American red and yellow pine, grey and black poplar.

(6) Wet or dry, but not alternately: Beech, fir, mahogany, pitch pine, spruce.

Timber in sea-water is rapidly attacked by the wood borers *Teredo navalis* and *Limnoria terebrans*; creosoting will prevent their attack for a considerable time; it has also been suggested to cover piles with sheet copper or flat-headed copper nails. The head of the pile may be protected by a wrought-iron hoop, about 3 ins × 1 in., to prevent splitting when being driven. The lower end should be pointed, the length of the point being from 1½ to twice the diameter, and where the ground is hard, protected with an iron shoe. Piles protected by shoes should have a blunt end 4 to 6 ins. in diameter. The shoes should have a solid point; the base being properly fitted to the blunt end of the pile prevents splitting by bolts which secure the shoes.

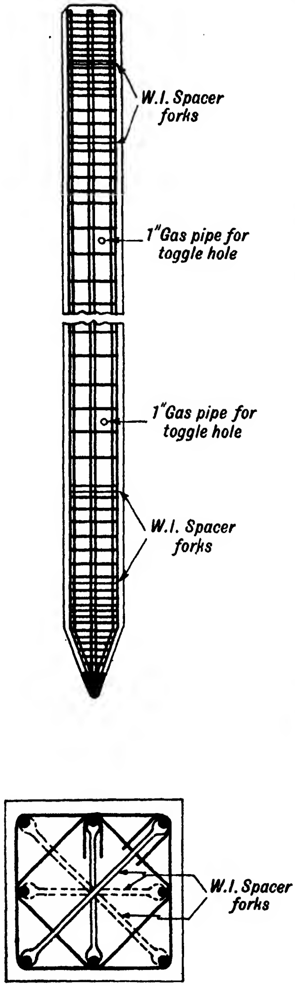


Fig. 15.

Concrete Piles.

(1) Pre-cast Piles.

Fig. 15 shows the commonest type of pile in general use, although other shapes of cross section may be used. In some cases the binding is helical at the same spacing as shown and often additonal longitudinal bars are placed along the centres of the sides, inside the binding, for the top 3 or 4 feet, with additional binding to prevent the bursting of the head of the pile. Toggle holes are left at convenient points for slinging and pitching. Piles should be matured for at least six weeks if made with ordinary Portland cement. If made with rapid hardening cement, however, they may be driven a week, or even less, after being cast. It is desirable to cast piles on the site to avoid damage in transit. A steel thimble-shaped helmet should be placed on the pile while it is being driven, to receive the blow of the monkey or hammer. A bag of sawdust, a pad of matting or old sacks, or a piece of soft timber should be placed in the top of the helmet to prevent the pile from being shattered. To lengthen piles *in situ* the heads are broken away, the reinforcement made up as for the original pile and the main bars overlapped 18 ins. or 2 ft. and tied in position with fine wire. The piles are then shuttered and the tops cast as in the case of columns.

(2) Cast in situ Piles.

Several proprietary methods are available. These have their individual merits and differences but share the general principle which consists of driving a circular steel tube fitted with a pointed cast-iron shoe to the required set. The required length of reinforcement is then assembled and lowered down the tube, concrete being afterwards poured in and the tube withdrawn, leaving a reinforced concrete pile in its place.

In cases where the required set cannot be attained at a reasonable depth by ordinary methods bulb toes are used. These are formed by depositing a small quantity of concrete, partially withdrawing the tube and tamping the concrete by means of a ram, thus causing it to spread. *In situ* piles can be used for lengths up to 70 or 80 ft. and they have the advantages that they can be constructed to the exact lengths required, thus obviating the necessity for splicing or removing surplus length, which often arises in the case of pre-cast piles; they can be formed with bulbs giving greater bearing value and they are not liable to be damaged in being handled or driven.

Loads Carried by Piles.

A timber or steel pile is usually of the same section throughout its length, and it is then only necessary to calculate the load borne by the greatest unbraced length of the pile, regarded as a strut or column. The part below ground may be regarded as held firmly or as quite free, according to the nature of the strata. Piles are often driven so closely together, or have to be of such strength for driving purposes, that they are more than strong enough for their loads. This also applies to piles which support their loads by skin friction only and are of ample section, or numerous, in order that the area subjected to skin friction may be great enough to support the loads. But it is often necessary for the sake of economy, especially when the piles are very long, to load them up to the safe limit.

The calculations of strength may then be made as follows :—

(1) If the pile is unbraced above the ground, the length measured from a few feet below the surface of the ground to the head of the pile is treated as a long column, and the section at the mid-point must be sufficient (see Column formulæ and 'Reinforced Concrete').

(2) If it is braced, the length taken is, in the case of a uniform pile, the longest unbraced length; that is, from a few feet below the ground to the point of bracing, or from the point of bracing to the head of the pile.

(3) If the pile is not uniform, the strength of the section at the mid-point of each of these lengths may have to be calculated separately, each length being treated as a long column.

(4) Bracing must not be considered effective unless it is effective in both directions, except in the following case:

(5) When the dimension of the pile is greater at right angles to the bracing than it is in the direction of the bracing, the two part lengths must be calculated as struts liable to cripple in the direction of the smaller diameter, and the whole length as a strut liable to cripple only in the direction of the larger diameter. If the formula for a square section of a reinforced concrete pile be used, the permissible load, expressed as a proportion of the short column load, may be thus modified. In each case modify the factor with l^2 in it, by multiplying by $\frac{r^2}{r_1^2}$ for the smaller dimension, and by $\frac{r^2}{r_2^2}$ for the larger dimension; where r = radius of gyration of a square section of the same area, r_1 = the smaller actual radius of gyration, and r_2 the larger actual radius of gyration. In the case of reinforced concrete piles, use the sums of the squares of the radii of gyration of the concrete and of the steel as in the formula (see 'Reinforced Concrete'). In this case the outside dimension might be the same, but the reinforcement stronger in one direction than in the other.

(6) For the portion of the pile below the ground :—

If the pile is in firm ground, or passes through a sufficient number of firm layers so that it is effectively steadied, any section of the pile should have sufficient strength, calculated as a short column; that is, it should take a load equal to the safe stress multiplied by the area. The load will be the whole load *less* an amount equal to the area of the pile above the section in question multiplied by the coefficient of friction. This will usually range from about 1 to 4 cwts. per square foot.

If the ground into which the pile is driven is soft and yielding, and is not regarded as effectively steadying the pile, the strengths at mid-points of each length l_1, l_2, . . . must be calculated as those of the mid-section of a long strut or column of that length. These lengths are measured from the head of the pile, or from the point of effective bracing, to successive points in the length of the pile, the last being measured to the tip. The load in each case is the whole load less the load carried by friction on the length of pile above the point in question. Unless the tip of the pile rests on a hard stratum, or is driven some way into a very firm stratum, the load vanishes before the point is reached, and very few calculations will actually be required.

(7) If the pile so rests upon a firm stratum that a large part of the load is carried from the lower end, the same procedure may be adopted, but the loads will, of course, be greater.

If the length of the pile below ground is small, and it is driven into or rests on a hard stratum, the whole length may have to be calculated as a strut carrying the whole load. Such length as is actually in firm ground should, however, be subtracted from the length for calculation.

In some cases the support given by the ground may not be regarded as equivalent to effective bracing, but may be considerable. The strength of any section may then be calculated in the first place as though the soil were quite firm and hard, giving a load W_1, and next as though the soil, though exerting the effect of skin friction, had no bracing effect, giving a load W_2. The actual load will be something between W_1 and W_2, depending upon the estimated value of the soil as a steadier of the pile. Even soft silt must be regarded as exerting a considerable steadying effect; but if there be progressive movement of silt, sand, or peaty ground, no steadying effect can be counted upon. Considerable movement, of course, demands the provision of special bracing, or, with short piles driven into a firm stratum, the calculation of a cantilever effect based upon the observed pressure as recorded by a torsion pressure meter or otherwise estimated.

Generally: to find the necessary strength at any section S, measure the distance upwards to the head or bracing, at A, and an equal distance downwards to a point B. Calculate the load P carried by skin friction above B, and (W — P) is the load carried by AB as a strut. The strength at S will be calculated as that of a short strut if most of the length BS be effectively steadied by the ground; if there be no such steadying the calculation will be that for a strut of length AB (but in no case any greater length). If the lower half of SB is in firm ground, or all of it in weak ground, the section may be calculated for a stress somewhere between that allowable as a short strut and that allowable on the length AB as a free strut.

The strength of the cross-sections of a pile does not change suddenly, and very few calculations need be made in actual practice; but any of the above may be necessary in a particular case.

The two most probable errors are, first, to calculate as free struts lengths which are really steadied by the ground, and, secondly, to neglect making for each section the reduction of load (regarding the section as the mid-point of a strut) due to skin friction from the surface of the ground to a point which is as far below the section as the bracing or head of the pile is above it. The load which the length in question itself carries by skin friction is imposed, upwards, upon some other mid-section nearer the head of the pile.

Loads on Reinforced Concrete Piles.

A well-designed reinforced concrete pile, 14 ins. by 14 ins. in section, will carry a load up to 100 tons (calculated). In actual practice such piles are carrying loads of 60 to 70 tons (at Bristol up to 65 tons; in the Thames, 15-in. × 15-in. piles, 70 tons).

Pile-sinking with the Water Jet.

In sinking piles the work may sometimes be much facilitated by the use of a water jet. In the case of hollow piles, the usual shoe may be replaced by a funnel-shaped end-piece, and the water pumped through the pile. In sinking cast-iron piles for bridge piers in New South Wales (*E. M. de Burgh*) it was found that a central nozzle with an aperture of 2 ins., and four holes of ½ in. diameter in the conical portion, gave good results in sand and soft clay, the piles being sunk through 14 ft. of sand in an average time of 6 minutes, and the actual rate of movement was about 8 ft. a minute. For sinking in sand not under water it was found expedient to flood the surface round the pile. In clay, the pile was loaded with 10 tons, and after 5 or 10 minutes pumping the pile would sink 6 to 9 ins. At a depth of 11 ft. in the clay, progress was very slow and practically ceased at 13 ft., with the 10-ton load. These piles had flanged joints.

Timber piles can be similarly sunk, the pipe being taken down the side of the pile. On the work referred to above, timber piles, 14 ins. in diameter at one end and 9 ins. at the other, were sunk 16 ft. on an average; the average time taken being 4 minutes, and the water pressure 5 to 7 lbs. per sq. in. in sand and 30 to 40 lbs. per sq. in. in clay. The pipe (2-in.) was taken to a 1½-in. augur hole in the point of the pile.

In driving steel sheet piling at Hodbarrow (*H.S. Bidwell*) a water jet was used in hard ground. Double ram pumps 5½ ins. and 6 ins. in diameter were used, with 10 to 14 h.p. boiler capacity, the water being delivered by a 5-in. pipe, diminishing to 3-in. and to 1½-in. From this a 1½-in. vertical pipe led to two ¾-in. jets. The jet pipes were kept working up and down, with a block and tackle, about level with or slightly below the point of the pile. After three or four tides, the ground had recovered its normal consistency.

Safe Load on Piles.

Rankine gives safe load on piles as follows :—

For piles driven till they reach firm ground, 1,000 lbs. per sq. inch of area of head.

For piles standing in soft ground, by friction, 200 lbs. per square inch of area of head.

Formulæ for Safe Load on Piles.

(1) *Wellington Formula.*

If P = safe load on a pile; h = height of fall in feet;

w = weight of ram; s = penetration per blow in inches.

(P and w must be in the same units—i.e. both in lbs. or both in tons, for example),

$$P = \frac{2wh}{s + 0\cdot1}$$

(2)

If P = safe load on pile in tons; a = sectional area of pile in sq. ft.;

w = weight of ram in tons; E = modulus of elasticity in tons per sq. in.

h = height of fall in ft.; x = weight of pile in tons;

s = depth driven by last blow in ft. l = length of pile in ft.

Rankine's formula,
$$P = \sqrt{\left(\frac{4Eawh}{l} + \frac{4E^2a^2s^2}{l^2}\right)} - \frac{2Eas}{l}.$$

Sanders' formula,
$$P = \frac{wh}{8s}$$

Dutch formula,
$$P = \frac{w^2h}{ks(w + x)}$$

k is a constant = 4 to 6 for concrete piles and 6 to 8 for timber piles.

In Rankine's formula the safe load is taken to be about $\frac{1}{5}$ of the greatest load; if there is a great vibration, the factor of safety should be increased to 8 or 9; where there is no vibration, a factor of safety of 3 or even less may be adopted.

'Engineering News' (U.S.A.) formula, $P = \dfrac{fwh}{p + 1}$

Where p is the average penetration in the last six blows; a factor ranging from 1 to 12 and recommended to be taken as 2.

Hiley's Formula (From the B.S.P. Pocket Book 1932).

A reliable pile driving formula for computing this resistance for all classes of piles and hammers has been obtained by A. Hiley, A.M.I.C.E. (*The Structural Engineer*, vol. 8, Nos. 7 and 8); it has shown satisfactory agreement with many actual loading tests, and is as follows:—

$$R = \frac{Wh\eta}{S + \frac{c}{2}} + (P + W) \quad \ldots \quad (1)$$

$$L = \frac{R}{F} \quad \ldots \quad (2)$$

Where R = resistance overcome in driving, expressed in tons.
W = weight of kinetic member or ram of hammer, in tons.
h = equivalent height of free fall of ram, in inches, taken as 80 per cent. of stroke for drop hammers, and 92 per cent. of stroke for B.S.P. single-acting hammers.
S = set, or penetration of pile per blow, in inches.
c = temporary compression of pile and cap, together with the temporary compression or 'quake' of ground beneath the pile point, in inches, caused by transmission of stress corresponding to R.
η = efficiency of hammer blow, representing the ratio the energy given out after impact bears to the striking energy of the ram.
P = weight of pile including helmet or anvil, in tons.
F = factor of safety against settlement, generally taken from 3 to 4.
and L = safe working load (including self weight of pile) which can be applied to the pile.

The value of η to suit given ratios of $\frac{P}{W}$ for typical conditions of pile driving, are set out in the following table:—

TABLE III.

Ratio of $\frac{P}{W}$	Case 1.	Case 2.	Case 3.	Case 4.
½	0·75	0·72	0·69	0·67
1	0·63	0·58	0·53	0·50
1½	0·55	0·50	0·44	0·40
2	0·50	0·44	0·37	0·33
2½	0·45	0·40	0·33	0·28
3	0·42	0·36	0·30	0·25
4	0·36	0·31	0·25	0·20
5	0·31	0·27	0·21	0·16
6	0·27	0·24	0·19	0·14

In using Table III, the ration of $\frac{P}{W}$ in the given case must first be ascertained.

Case 1 refers to double-acting hammers driving steel sheet piles or R.C. piles.
Case 2 refers to double-acting hammers driving timber piles.
Case 3 refers to single-acting and drop hammers driving timber piles, or R.C. piles fitted with helmet and wood dolly.
Case 4 refers to ditto, when the head of pile or cap is in poor condition.

Suitable allowances for temporary compression, *c*, in inches for typical cases of piles corresponding to medium driving conditions are given in the following table :—

TABLE IV.

Length of Pile.	Temporary Compression c in inches.							
Feet.	Sheet piles.		R.C. piles without helmet.		R.C. piles fitted with helmet and dolly.		Timber piles.	
	(1)	(2)	(1)	(2)	(1)	(2)	(1)	(2)
	in.	in.	in.	in.	in.	in.	in.	in.
20	0·04	0·08	0·27	0·39	0·47	0·79	0·36	0·57
30	0·06	0·12	0·33	0·51	0·53	0·91	0·44	0·73
40	0·08	0·16	0·39	0·63	0·59	1·03	0·52	0·89
50	0·10	0·20	0·45	0·75	0·65	1·15	0·60	1·05
60	0·12	0·24	0·51	0·87	0·71	1·27	0·68	1·21

TABLE V.

Item.	Size numbers of McKiernan-Terry hammers.								
	1	2	3	5	6	7	9B2	10B2	11B
Weight of ram in lbs.	21	48	68	200	400	800	1,500	2,500	3,625
Weight of anvil in lbs.	13	37	40	86	203	314	387	550	700
Weight of casing in lbs. . . .	111	258	567	1,214	2,297	3,886	4,873	6,950	8,860
Total weight of hammer in lbs. . .	145	343	675	1,500	2,900	5,000	6,760	10,000	13,185
Actual stroke in ins.	3·75	5·25	5·75	7·0	8·75	9·5	16·0	20·0	20·0
Average number of blows per minute .	500	500	400	275	230	195	130	105	110
Average kinetic energy of blow in ft.-lbs. . .	100	140	350	820	1,970	3,710	8,500	15,800	20,800

Columns 1 in Table IV refer to a medium driving resistance, assumed to correspond to a stress transmitted through the cross section of the pile of 4,000 lbs. per sq. in. for steel sheet piles; and 1,000 lbs. per sq. in. for R.C. or timber piles.

Columns 2 refer to a very hard driving resistance corresponding to stresses of 8,000 lbs. per sq. in. for steel sheet piles; and 2,000 lbs. per sq. in. for R.C. or timber piles on the cross sectional area.

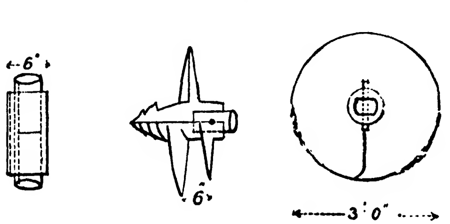
Intermediate values of c corresponding to other intensities of driving resistance between the values stated above, may be obtained by interpolation.

For McKiernan-Terry double-acting pile hammers, the energy Wh in the formula 1 should be expressed in inch-pounds: the resistance similarly will then be given in pounds.

The necessary particulars of these hammers and the kinetic energy (K.E.) given out when driving, are given in Table V.

Screw Piles.

Screw Piles, figs. 16 and 17, are either of wood or of cast or wrought iron; they are seldom less than 6, or more than 18 inches, in diameter. They should be cylindrical in section to enable them to be more readily turned. They are fitted with a screw disc at the foot of the pile, similar to an auger, which usually does not make more than one and a half turns. This



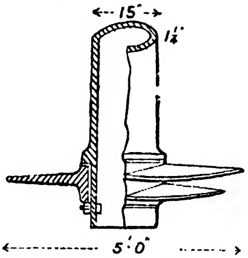


FIG. 16.—Wrought-iron pile with cast-iron disc. FIG. 17.—Cast-iron pile and disc.

disc is of cast-iron, which is made up to 6 feet in diameter. The piles are usually sunk by manual power acting on long levers fixed to a capstan head on the piles, so as to screw them down to a firm stratum. When screw piles have to be sunk through hard ground, a water jet may be used to facilitate sinking.

Cylinders.

Cylinders of cast-iron are sunk to a firm foundation by excavating inside and letting them sink by their own weight, or if that is not sufficient, adding additional weight. As the cylinder sinks, lengths are bolted on by means of flanges which are on the inside. When it is desirable to make the joints water-tight, a cloth strip saturated with thick paint, or a strip of bitumen cloth, is laid between the flanges.

Small cylinders from 4 to 10 feet in diameter are usually cast in lengths of from 5 to 10 feet, without any vertical joint, and from 1 to 1½ inches in thickness with 1½ or 2 inch internal flanges. The thickness in inches necessary to withstand the pressure of the water of small thin cylinders may be approximately found from the formula—

$$t = \cdot 001\, rh\,;$$

where r is the radius of cylinder, and h depth of water, both in feet, and assuming that the metal will stand with safety a compression of 5,000 lbs. per square inch. In no case should the thickness t be less than ½ inch. The inside of the cylinder is excavated by hand, or if the water cannot be kept out, by means of a grab, and is afterwards filled up with concrete or masonry.

At the St. Louis Bridge, U.S.A., the cylinders were 4, 6, and 8 feet in diameter, cast in 10-feet lengths, the metal being 1½ inch. thick with 2½-inch flanges.

The bottom cylinder of the Albert Bridge over the Thames was 21 feet in diameter and 4 feet 6 inches high, cast in one piece; on this was a reducing ring tapering to 15 feet in diameter; all the other rings were 15 feet in diameter and 6 feet high, also cast in one piece.

Brick or masonry cylinders built on an iron or wooden curb are frequently used instead of iron cylinders.

Pneumatic Process of Sinking Cylinders.

Cylinders and large caissons are often sunk by the pneumatic process. The cylinder is made as nearly as possible air-tight, and an air-lock or chamber, with two doors, one opening from the outer air and the other into the cylinder, is bolted on to it; both of these doors open downwards into the cylinder or against the pressure of the air. Air is pumped into the cylinder so as to drive the water out. The workmen then enter the air-lock, the door communicating with the outer air is closed, and the compressed air allowed to enter from the cylinder; when the pressure has become equal the door into the cylinder is opened and the workmen descend. The excavated material is carried up through the air-lock, or blown up through a tube by air pressure. The cylinder is buoyed up by the pressure of the air inside it, and when this is released it tends to sink. It has generally been considered that men cannot work in safety under a greater depth of water than 100 ft.; in the St. Louis Bridge 108 ft. was reached, and 147 ft. under mean low water is said to have been attained in the East River Gas Company's tunnel at New York.

Steel Sheet Piling.

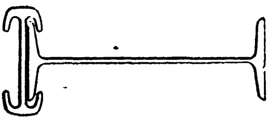


FIG. 18.—Universal Joist Section.

British standard joists are employed, having flanges 5 ins. wide and used in conjunction with a standard clutch or locking bar. The ranges of joists are 15 ins. by 5 ins., 12 ins. by 5 ins., 10 ins. by 5 ins., 8 ins. by 5 ins. and 6 ins. by 5 ins. The clutch is put on the flange of joist at works and the two driven as one. The section has great strength, and is very easily straightened for re-use and possesses high salvage value, as the joists can be used for constructional purposes. Supplied up to 65 ft. lengths.

In addition to the pattern shown above there are several proprietary steel sheet piles on the market. These have various sections, but in most cases consist of steel plates with interlocking grooves running their whole length. Most modern types are trough shaped in order to increase the stiffness of the piling by having most of the metal as far as practicable from the neutral axis.

Driving.

The driving of the steel sheeting is generally effected by means of double-acting pile hammers. The blows are given in rapid succession.

Light-weight sheeting, and also heavy piling, may at times be very advantageously driven with a water jet. Such material as mud, soft clay, and clear sand will ordinarily respond to the jet. If, however, the material is hard clay or a sand containing a heavy percentage of gravel, the water jet is an inadvisable adjunct. A supply pipe of 1 in. diameter and a $\frac{3}{4}$-in. nozzle will be suitable for moderate weights of piling having a width of, say, 12 ins. or less.

Pulling.

It is quite customary to pull piles that have been used in temporary service. The cost of pulling is considerable. While, in general, it is not as great as the cost of driving, still it is to be rated as equal to a large percentage of the expense of driving the very same piles which it is proposed to pull out again. This percentage may rise as high as 75, or even higher. Where the piling has been driven with a wooden maul, or could have been so driven, a simple lever may suffice. This lever may have attached to it metal straps or other devices to facilitate connecting the pile head with the lever. Simple perforations in the top of the pile serve for the introduction of bolts or pins. Ordinary hoisting arrangements may suffice for these piles and for still heavier sheeting. Hydraulic jacks have been used for rather difficult pulling. The jacks may be used mainly for starting the units. By setting up jacks on the heads of the adjacent piles they may be brought to bear beneath a head block provided with a slot through which the intermediate pile may pass on its way up. Another method that has been used simply sets up a framework provided with a strong cross-piece, to which blocks and tackle may be rigged. The line may be operated by hand or by power, as circumstances may require. For pulling piles which have been put down to a considerable depth, a percussion-piston pile-driving hammer may be hung upside down from a derrick boom and used thus to pull the piles. In pulling, the inverted hammer takes hold of the pile by means of the shackle, and steady tension is then exerted on the pile line, which pulls the shackle strap down upon the anvil block, and the hammer is placed in operation. Piles of 26 ft. penetration in clay, sand, and gravel have been withdrawn in from 2 minutes to 10 minutes each, without distortion. The Mc-Kiernan-Terry pile hammers can also be used for pulling purposes.

Timber Sheet Piling.

Timber sheet piles are usually from 9 ins. to 12 ins. wide and 3 ins. to 6 ins. thick. They have chisel-pointed shoes and the tops are bound with iron straps. The shoes are usually bevelled in order that each pile may be forced against the pile previously driven. The joints between the piles may be plain butt joints, rebated joints or of one of the forms shown in fig. 21, p. 448.

Reinforced concrete is sometimes used for sheet piles, the dimensions and method of use being much the same as in the case of timber sheet piles.

Uses of Sheet Piling.

Sheet piling is used for coffer dams, holding up the faces of excavations and similar temporary work and also for quay walls, retaining river banks, etc.

In the latter cases it may be used in the following ways :—

(1) Sheet piling alone driven deep and supporting the ground behind as a cantilever.

(2) As (1) but backed with concrete, in which case it serves as protection and shuttering for the concrete.

(3) As (1) but anchored back by means of tie rods fastened to walings across the front of the piles and to concrete anchor blocks buried in the ground some distance behind the piles. In this case the piles may be designed as simply supported slabs between walings and between the lowest waling and a point about two-thirds the distance between the river bed and the toes of the piles.

(4) Ordinary square piles are driven at regular intervals, walings are fixed behind these and sheet piles are driven at the back of the walings a relatively short distance into the river bed. The square piles, known as king piles, are anchored back to concrete blocks.

Caissons.

Very deep and large piers and abutments of bridges have recently been constructed with steel caissons braced internally so as to prevent them collapsing while sinking. These caissons are excavated and filled with concrete or masonry.

The foundations of the Hawksbury Bridge in Australia were sunk to a depth of 162 ft. below high-water level by dredging.

Frictional Resistance of Pneumatic Foundations (Schmott).

Tables of Coefficients of Friction.		Dry Material.		Wet Material.	
Material.	Soil.	First Movement.	During Motion.	First Movement.	During Motion.
Sheet-iron, without rivets .	Gravel and sand	·4015	·4583	·3348	·4409
Sheet-iron, with rivets .	" "	·3965	·4911	·4677	·5181
Cast-iron, unplaned . .	" "	·3677	·4668	·3616	·4963
Granite, roughly worked .	" "	·4266	·5368	·4104	·4800
Pine, sawn	" "	·4088	·5109	·4106	·4985
Sheet-iron, without rivets .	Sand	·5361	·6313	·3655	·3247
Sheet-iron, with rivets .	"	·7269	·8391	·5156	·4977
Cast-iron, unplaned . .	"	·5636	·6063	·4744	·3796
Granite, roughly worked .	"	·6473	·7000	·4728	·5291
Pine, sawn	"	·6633	·7340	·5787	·4793

The foundation must be vertical, being kept in equilibrium by the friction of its surface.

According to Mr. F. W. Sweeney in the *Journal of the Western Society of Engineers*, experience in sinking concrete caissons for bridge foundations has shown that building the walls with a batter is not only of no particular advantage in reducing the sinking friction, but it is positively detrimental to the proper guiding of the caisson. It is considered the best practice to have a perfectly plumb wall. When the depth of water will permit, it is generally cheaper to build up a river caisson on shore in the form of a wooden shell, launch it, tow it into position between guide piles and sink it by filling with concrete.

Freezing Process of Sinking Foundations.

The Freezing Process has been used to sink large foundations in treacherous material, especially quicksand. Pipes 10-inch in diameter, open at the bottom, are sunk vertically about 3 feet apart round the ground to be excavated ; inside of these 8-inch pipes, closed at the bottom, are lowered, and again within the latter small pipes. The freezing material is pumped into the smaller pipes, and returns to the cooling tank through the large ones. A solid cylinder of frozen ground is gradually formed round the pipes ; these frozen cylinders unite, forming a solid ring round the place to be excavated. The material to be removed will, of course, be frozen solid, and therefore much more difficult to excavate. An example, at Swansea, was described in detail in *The Engineer* of May 26, 1933. A chemical process of consolidation was also introduced a few years ago. (See p. 436.)

DAMS.

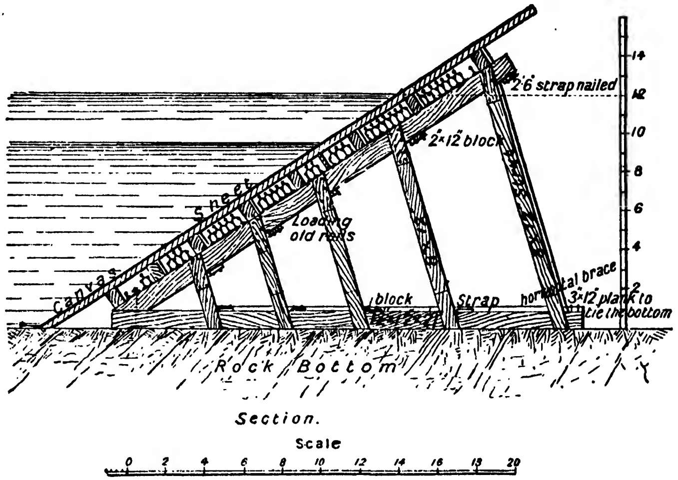


FIG. 19.

If the depth of the water does not exceed 4 feet, the bottom moderately firm, and there is no current, a dam of clay puddle about 3 feet wide on top may be formed round the space to be enclosed; before the dam is commenced care should be taken to remove all loose porous material

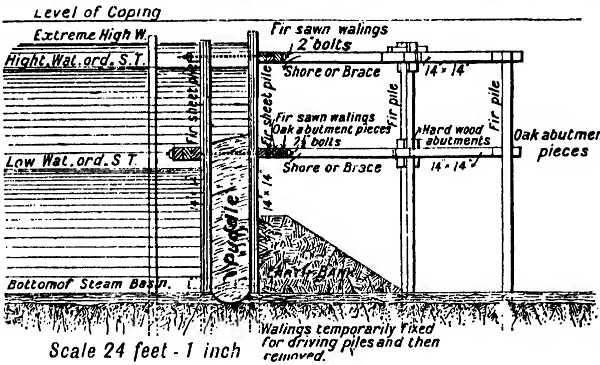


FIG. 20.

and roots from the site, and, if possible, excavate a trench for its foundation. Long lengths of waterproof canvas, sewn together and weighted with a chain stitched to the ends so as to keep it in place, have also been used. The canvas should be supported by a framework of timber (see fig. 19). These frameworks are spaced about 8 feet apart and sheeted with boarding between. The ground should be dredged level to receive them.

Puddle Dams.

Coffer dams of puddle usually consist of two parallel rows of sheet piling, the intervening space between them being filled in with puddle; sometimes three or even four rows, with a puddle wall between each, have been adopted in order to give greater stability and make the dam more watertight.

It is always desirable to make the puddle wall as thin as possible; it is seldom more than 4½ or 5 feet in thickness. Bolts through coffer dams should be avoided as far as possible, as the puddle in settling leaves vacant spaces along them, causing leaks; to remedy this, iron discs through which they pass have been used; these discs are embedded midway in the puddle wall. In coffer dams made for the construction of the East London Railway through the docks, through bolts were entirely dispensed with, their place being taken by buttress piles driven at intervals on the outside which withstood the pressure of the puddle.

Fig. 20 shows a coffer dam used at Portsmouth Dockyard extension. The shoring piles were driven 9 ft. centre to centre, and 12-in. by 12-in. struts, one horizontal and one inclined, were fixed to each.

Puddle.

(Alfred B. Searle.)

Puddle is a highly plastic paste made by mixing 'clay' with about one-fifth of its weight of water, the precise amount depending on the plasticity and dryness of the 'clay.' A perfectly pure 'clay' does not usually produce good puddle, as it is either deficient in plasticity (like china clay) or it shrinks excessively. The former defect can only be remedied by the use of a more highly plastic material; excessive shrinkage may be prevented by mixing a suitable proportion of sand with the clay. For the best results, a piece of puddle when dried should not shrink more than 1¼ in. per linear foot; on the other hand, it should not shrink less than ¾ in. per linear foot or it will probably not be sufficiently impermeable to water.

The materials used must be carefully selected and thoroughly well mixed; some contractors are very careless in this respect and will use anything which has been passed through a pugmill. The correct consistency for good puddle is that at which that paste can be squeezed easily in the hand without any appreciable quantity adhering to the latter when the pressure is released. Puddle made of the consistency of porridge is much too soft for best work, though it is better than an over-dry or badly mixed stiff paste.

Where possible, the materials used for making puddle should, before being mixed with water, be passed between a pair of rollers placed not more than ¼ in. apart, so as to crush any stones or coarse gravel present. It is also desirable that all roots and large stones should be picked out before sending the 'clay' to the mixer.

The usual method of preparing puddle is to mix the local 'clay' with sufficient water to form a soft paste, a pugmill being used for the mixing. The 'clay' may consist of a sandy loam with 50 per cent. of sand and, therefore, incapable of producing a mass impermeable to water except within extremely narrow limits of consistency, which are seldom investigated. On the other hand, it may be so rich in colloidal matter as to have an abnormally high shrinkage on drying; this is a very objectionable property, as puddle which shrinks greatly usually cracks and becomes unreliable. It may be prevented by using a plastic mixture of 'clay' and very fine sand which is well rammed into position, the proportion of sand being such that the puddle, when fully dried, has a shrinkage not exceeding 1 in. per linear foot. No definite figures can be equally suitable for all cases, but two parts of clay to one part of fine sand is often suitable, though much depends on the nature of the clay.

Too much water should not be used in making puddle; it is often better to use moist clay mixed with sand rather than to add sand to a clay-slurry in order to stiffen it. The stiffer puddle must be mixed mechanically in a powerful pugmill, and it must be rammed in position by mechanical rammers. Under these conditions it produces excellent results.

To produce a really satisfactory puddle, it is not sufficient to proceed along 'cookery-book' lines. The available materials should be mixed in different proportions, and each mixture should be tested in a hydraulic press to ascertain its permeability. A useful 'holder' for the puddle is a hollow cylinder made of Portland cement and sand, or even a 4-in. salt-glazed drain-pipe, into one end of which the puddle is placed. The cylinder or pipe is then placed vertically on a flat bed and the pressure applied. A little skill is required to make the puddle fit properly, but once this has been acquired, the test is easy to make.

The worst kind of puddle is that made by preparing a liquid slip, wash, or slurry of clay, and letting this settle. The sand and other non-plastic material normally present is removed in this method of preparing the slip, and the product has too great a shrinkage.

In trenches filled with properly made puddle, there appears to be no advantage in using a 'reinforcement' of strings of jute, as the plain puddle is quite impermeable when properly prepared. A mixture of wood-wool and clay makes an excellent puddle, but requires a machine for its production.

Permeability of Different Puddles.

Tests made under hydraulic pressure to determine the permeability of different puddles showed that it is possible to produce mixtures which, when only 6 ins. thick, will withstand 50 lbs. per sq. in. water-pressure for 5 hours without showing any signs of seepage. Puddles free from sand stood this test the best because of their greater density, but those containing sand are so much less liable to crack that, on the whole, they are the most suitable for general use.

Single Balk Dams.

Dams have also been most successfully constructed of single rows of sheet piling grooved and driven between waling pieces. Fig. 21 is the more usual arrangement; the piles are from 9 to 12 inches in thickness; a groove about 2 inches wide and 2 inches deep is cut out of each

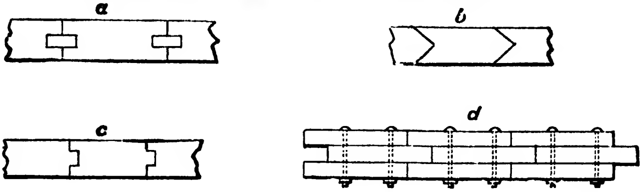


FIG. 21.

pile; into this is fitted a tongue, which is cut to the exact size to fit it. Sheet piles should be sawn die-square with the grooves perfectly straight and true. They should be of sufficient thickness to withstand the water pressure without bulging between the waling pieces. Small leaks at the joints may be stopped by caulking with oakum.

Cribs.

Timber cribs or boxes, which are usually made watertight and floated into position and then sunk, have been largely used in America for foundations of bridges on rock, or in places where there is no fear of undermining by the currents. They are filled with concrete.

Scaffolding.

(Note: Tubular steel scaffolding is now extensively used.)

Standards are upright poles, generally of spruce, 20′ to 50′ long, 5″ to 8″ diam. at butt; those under $2\frac{1}{2}''$ at tips are termed *rickers*, and run about 22′ long. They are firmly fixed in the ground (or in tubs filled with earth, if excavation is inconvenient), 4′. 6″ from wall, 10′ to 12′ apart. When greater lengths are required, two are lashed together, tip to butt, and tightened with wedges To these, on the side next the wall, are lashed horizontal *ledgers*, or *runners*, parallel to the wall at vertical intervals of 3′ 0″ to 5′. They support the *putlogs*, of squared timber (usually birch), 4″ × 3″, 6′ long, one end of which rests on the ledgers, the other in a hole in the wall left by omitting a brick header; they are about $3\frac{1}{2}'$ or 4′ apart, and support the *scaffold boards*, usually 9″ × $1\frac{1}{2}''$, sometimes 3″. Poles called 'braces,' lashed diagonally across every three or four standards, stiffen the scaffold. Masons often use a second row of standards and runners, close to the wall, to prevent the masonry being disfigured by putlog holes.

Fig. 22 (from Spon's 'Dict. of Eng.') shows the arrangement of a bricklayer's scaffold. When the scaffold has to be carried to a considerable height, other poles are lashed to the standards with ropes tightened by wedges. Poles are also lashed diagonally across every three or four standards in the shape of a St. Andrew's cross; these are called braces, and they serve to stiffen or brace the scaffold longitudinally.

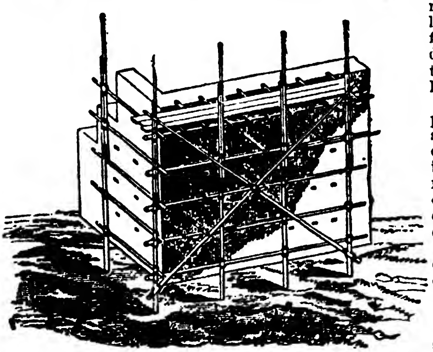


FIG. 22.

In buildings which do not admit of putlog holes in the walls, as where rubble stone or ashlar facing is used, and which do not require heavy machinery for hoisting, or strong timbers in the scaffold, two rows of standards with ledgers are used, one row being close to the wall, and the other at the usual distance, so that both ends of the putlogs may rest on the ledgers. Scaffolds such as we have just described are sometimes used for heights of 90′ or 100′ from the ground, as in building church steeples and similar work. In the erection of houses it is usual to construct a staging about 10′ square on the outside of the scaffolding, for the purpose of hoisting materials, and from which they are distributed for use.

This staging is usually formed with standards and ledgers in the same manner as the scaffold to which it is connected.

Scaffolding Knots.

The principles of a good knot are its facility in tying, its freedom from slipping, and its being easily untied. All knots will jam more or less when subject to a strain. The knots in fig. 23 are shown open before being drawn taut, in order to show the position of the parts. The names usually given, and their uses, are as follows :—

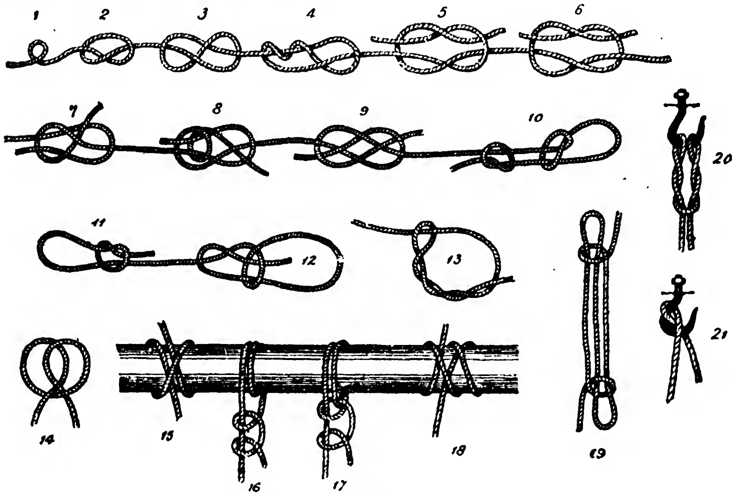


FIG. 23.

1. Bight of a rope.
2. Overhand, or thumb, knot, to prevent a rope running through the sheave of a block.
3. Figure of eight knot, used as No. 2.
4. Stevedore knot: is useful when the rope passes through an eye. It is easily untied after being strained.
5. Square, or reef, knot. This is a most useful knot for joining two ropes of same size. However tightly it jams, it is easily 'upset' and undone.
6. Granny, or thief, knot. This should not be used, as it will jam tightly but not slip (as erroneously supposed), will not 'upset' and consequently is difficult to undo.
7. Single sheet bend, or weavers' knot, used principally for joining two ropes of unequal sizes more securely than a reef knot.
8. Double sheet bend, more secure than No. 7.
9. Carrick bend, for fastening the four guys to a derrick.
10. Flemish loop.
11. Slip knot.
12. Bowline, for making a loop that will not slip. After being strained, this knot is easily untied. Commence by making a bight in the rope, then put the end through the bight and under the standing part, pass the end again through the bight, and pull taut. This knot should be tied with facility by everyone who handles ropes.
13. Timber hitch ; the greater the strain the more tightly it will hold.
14. Clove hitch, consisting of two half-hitches.
15. Clove hitch, as No. 14, showing its application around a pole.
16. Round turn and two half-hitches, for securing a rope to a ledger or for fastening the guys of derricks, shear legs, &c.
17. Fisherman's bend, used when a thick rope, such as a fall, is made fast to a ring.
18. Rolling hitch, used in a variety of ways, chiefly in making fast one rope to another that is held taut.
19. Sheepshank, for shortening a rope when the ends are inaccessible.
20. Catspaw. An endless loop, and used where great power is required.
21. Blackwall hitch. Easily applied, but requires watching; has a tendency to slip.
22. Square lashing. See p. 450.
23. Diagonal lashing. See p. 450.

Square Lashing.—This is used to fix a ledger to a standard and is made as follows: A clove hitch is first made round the standard just below the ledger. The short or running end of the rope is disposed of by twisting it round the long or standing end and the latter is then passed up in front of the ledger, horizontally behind the standard, down in front of the ledger and again horizontally behind the standard. Three more similar turns round the spars are then taken. Two turns known as frapping turns are next taken in a vertical plane between the two spars, round the four turns just mentioned, these being well beaten in and the frapping turns pulled as taut as possible. The lashing is completed by making two half hitches (equivalent to a clove hitch) on the ledger.

Diagonal Lashing.—Used for fixing diagonal braces to standards or to each other. It is made in the following way: A timber hitch is made round both spars diagonally across their intersection and three complete turns, exclusive of the timber hitch, are then taken round them in the same plane. Three turns are next taken across the other diagonal and the lashing is completed with two frapping turns and two half hitches as in the case of the square lashing.

Shoring Buildings.

A plank (fig. 24), C, 9 ins. wide, 3 ins. thick, the length varying with height of building, is placed against upper part of wall to be supported. In this rectangular holes are cut to admit of timber 'needles,' D, from 4 ins. × 6 ins. × 6 ins., and about 12 ins. long. These are let into

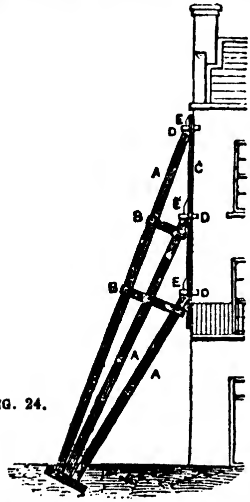


FIG. 24.

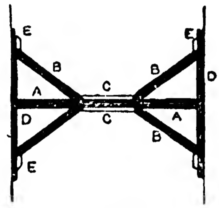


FIG. 25.

the wall about 4½ ins., and project beyond the plank about 4½ ins. to receive the ends of the 'shores,' A.

A cleat, E, is usually nailed to the plank on the upper side of each needle.

The struts, A, are from 6 ins. × 4 ins. in very small buildings to 12 ins. × 9 ins. or 12 ins. in very large; usually half timbers 12 ins. × 6 ins. They are fixed to a footing block, F.

The outer strut is called a 'top raker,' the inner the 'bottom shore,' the centre the 'middle raker.'

Short pieces, B, about 1 in. thick and from 6 ins. to 9 ins. wide are nailed on each side of the shores.

Struts between Houses.

A (fig. 25), horizontal strut, 6 ins. × 4 ins. to 9 ins. × 4 ins.

B, raking strut, 6 ins. × 4 ins.

C, straining-piece, 6 ins. × 3 ins.

D, plank, 9 ins. × 3 ins.

E, cleats.

The horizontal strut, A, is at height above ground equal to about two-thirds height from ground to eaves.

The Timbering of Trenches and Shafts.

Excavations must be timbered where the ground will not stand without support, in order to render possible the construction of the work in hand, to ensure the safety of the men engaged upon it, and to prevent settlement which may result in damage to adjacent buildings or other structures, mains of various kinds and road surfaces. Any 'loss of ground' may, in addition to causing damage, result in movement of the timbering because disturbance of the ground, if of a loose nature, may exert on the timbering pressure far in excess of that of the undisturbed ground. Movement of the timbering may result in insufficient room being available for the permanent work and necessitate 'poling back,' *i.e.* setting back the timbering to the required position—an operation which under some conditions is tedious and costly.

The amount of timbering needed in any excavation depends upon the nature of the ground and the length of time during which the excavation is to remain open, and may range from an occasional support to timbering completely covering the excavated faces. In extreme cases it is necessary even to timber the bottom of the excavation in order to prevent the ground from being squeezed upwards by the pressure at the sides.

It is very important that the area of ground which may be at any time temporarily unsupported during the fixing of timber should be restricted to safe limits. Although in many cases the ground will stand with little or no support for a short time, heavy timbering may be required if the excavation is to remain open for a long period.

In view of the difficulty of fixing timbering perfectly plumb and of the possible reduction in the size of the excavation due to pressure and in some cases the swelling of the ground, it is desirable, when permanent work is to be accommodated, to take out the excavation to dimensions somewhat in excess of those actually required.

The timber actually in contact with the ground consists either of poling boards or of long timbers called runners which are driven down as excavation proceeds. Unless the ground is of a strongly cohesive nature any cavities behind the timbering should be filled either with loose material rammed in or with bricks, wood or other suitable packing.

In the case of loose sand, especially if water be present, the passage of the particles through the small openings between the timbers is often prevented by 'stemming,' *i.e.* the insertion of hay, straw or similar material. This, although it does not prevent the flow of water, stops much of the sand which would otherwise accompany it.

Poling boards and runners are generally of spruce, although pitch pine is preferable and possibly cheaper in the long run on account of the longer period during which it will remain serviceable; heavier squared timber is usually of pitch pine; round bars of larch and wedges of beech or other hard wood, pitch pine being, however, frequently used.

TRENCHES.

(*a*) *Poling Boards.*—The boards are usually from 3 ft. 6 ins. to 4 ft. 6 ins. long and 1 in. to 1½ in. thick. In all cases they are held against the ground by longitudinal timbers called walings, between which and the boards pages (small wedges) are driven. Walings vary from 4 ins. by 2 ins. to 12 ins. by 12 ins., according to requirements, and when of rectangular section they are fixed with the longer dimension vertical for convenience, although this is not the most economical way of resisting lateral bending. The walings are held apart by struts, the best section for which is square, their function being to resist compression. Round and rectangular timbers are, however, frequently used. In most cases strips of wood, called lips, are nailed to the tops of the struts at the ends to hold them level with the walings while they are driven into position. See fig. 26. Large wedges, called driving wedges, are generally used to tighten up the struts, but in some cases slack blocks, as shown in fig. 27, are used, these being more easily removed.

When considerable pressure on the timbering occurs it frequently happens, especially when the timber has been in position for a long time, that the struts cannot subsequently be knocked out on account of their having been forced into the walings which have been partially crushed. When this is anticipated 'cutting-out pieces,' fig. 27, are inserted between the struts and the walings when these timbers are fixed. Cutting-out pieces are removed in the manner indicated by their name and the waste of valuable timber thus avoided.

Struts are fixed at both ends of the walings and as many intermediate struts as necessary are added, the distance apart of the struts depending on the nature of the ground, the size of the walings and the working room required between the struts. Two opposite walings with the struts between them form a complete frame. See fig. 28.

In order to prevent the dropping of the walings, vertical props, called puncheons, are wedged between them, as in fig. 26, the feet of the puncheons resting on the lips of the struts, if any. Lacing boards are also nailed vertically to the faces of the walings near their ends. It is essential that, as the excavation proceeds, the lowest walings should be securely propped from the bottom of the excavation. Should the bottom of the trench be very soft it is sometimes necessary to prevent the settlement of the timbering by slinging it from timbers laid transversely across the top of the trench.

In order to prevent the settlement of the timbering in deep trenches it is usual to insert at intervals, possibly under every third or fourth frame, midway between the struts, raking or back props, as shown in fig. 29, these being fixed as the excavation proceeds, resting on foot blocks securely driven into the ground.

There are three methods of fixing poling boards in general use, these being known as the middling board, tucking board and piling board systems.

Middling Boards.—In this method, shown in fig. 30, each setting of boards is held in position by a single waling. It is only used in fairly good ground and has the disadvantage that it does not readily lend itself to the adoption of the piling board method, to be described later, if bad ground is met with below. In fixing the first setting, the trench having been excavated to the requisite depth, the walings are placed in position resting on temporary supports with poling boards behind them opposite the points at which the struts are to be fixed. Packing pieces, called 'chogs,' are put in between the boards and the walings to allow room for the pages in front

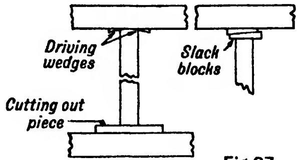


Fig.27

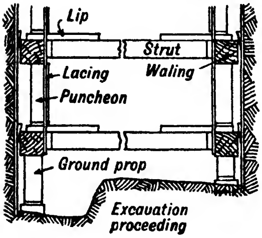


Fig.26

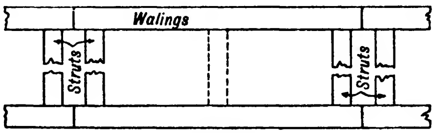


Fig.28

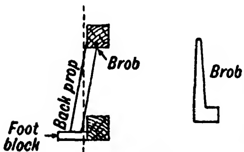


Fig.29

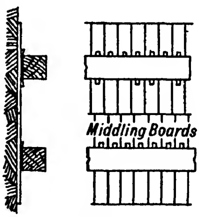


Fig.30

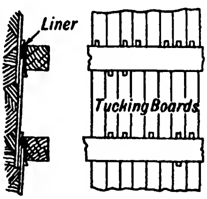


Fig.31

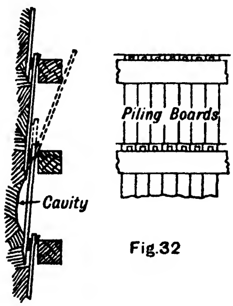


Fig.32

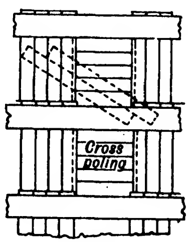


Fig.33

of the boards to be subsequently inserted. The struts are then fixed. The remaining boards are finally placed behind the walings and held against the ground by means of pages driven in behind the walings, usually at the top only but sometimes also at the bottom. In cases in which the ground is good enough to permit it open poling is frequently adopted. This consists of inserting the boards with spaces between. In the second and lower settings the whole of the boards may be placed in position before the walings are fixed, the lower ends being toed into the ground or kept in position by pegs or pages driven into the ground and the tops held by nails driven obliquely into the boards above.

Tucking Boards.—In this system, shown in fig. 31, which is used when the ground is not good enough for middling boards or where bad ground necessitating piling boards is expected to be met with below, the poling boards are held at both ends. The walings, with the exception of those at the top frame, have nailed to them liners of the same thickness as the boards. The poling boards above the walings pass behind the liners and their lower ends are about level with or slightly below the bottoms of the walings. The boards of the setting below are subsequently 'tucked' into the space below the liners. The method of fixing the frames is as described for middling boards except that the temporary support of the top walings while the struts are inserted is usually afforded by raking ground props which do not interfere with the fixing of the frame below.

Piling Boards.—These are illustrated in fig. 32 and are used in bad ground. The boards are usually bevelled off as shown at the lower ends and are in most cases driven as indicated by dotted lines. Distance pieces or chogs are, in the first place, inserted between the liners and the walings and these are taken out as the driving of the boards proceeds. The piling board method has the advantage that the sides of the trench can be supported during the whole of the time excavation is proceeding, the toes of the boards being kept slightly below the level to which the bottom has been excavated. Driving is commenced at a considerable angle from the vertical, and the boards assume the right batter as they are driven. In order to prevent them from approaching too near the veritcal chogs, are placed between them and the boards above. A small cavity behind the boards is inseparable from this method of driving and sometimes the method shown in fig. 33 is adopted to avoid this. In this case the boards are driven in on the slant laterally and are gradually righted. When only a small gap is left between the boards when driving has been done in opposite directions towards the same point, the gap is filled with cross poling as shown. If it is found necessary to resort to the use of piling boards after tucking boards, this is effected by removing the liners from the walings of the lowest tucking frame.

(*b*) *Runners*, shown in fig. 34, are frequently used in bad ground instead of poling boards. They are long timbers up to 20 ft. and occasionally more in length and from 2 to 3 ins. thick by 6 to 9 ins. wide. The lower ends are toed or bevelled off and the heads are usually bound with hoop iron. The runners are driven down behind the frames which are from 3 to 5 ft., apart the pages being loosened to enable this to be done. As in the case of piling boards the toes of the runners are kept slightly below the bottom of the trench and the sides are never unsupported. Sometimes runners are used for the full depth of the trench, in which case the top frame may be at the surface, a guide being fixed at a higher level to ensure the runners being driven vertically. A scaffold or other means whereby the men are enabled to drive the runners may be required, but it is sometimes possible to dispense with driving until the runners are some distance in the ground. Frequently a setting of poling boards is used at the surface as shown in fig. 34 to obviate the use of the guide above ground and reduce the height of the scaffolding (if any). This method is also adopted when the depth of the trench is slightly greater than can be reached by one or more sets of runners.

Whenever runners are used below either poling boards or another set of runners, an internal frame is needed to enable the runners to be driven in front of the frames above. This necessitates the taking out of the upper part of the trench to a width sufficient to allow for any internal frames which may be required.

It is obvious that gaps must occur between the runners below the struts and, if the nature of the ground permits, these small spaces are left until the fixing of the frames below enables boards to be inserted. Should it be found undesirable to leave these areas unsupported, cross poling may be inserted behind the runners as excavation proceeds. See fig. 35.

Occasionally horizontal runners are used, in most cases in small trenches as shown in fig. 36. Sufficient ground is excavated to enable a pair of runners on opposite sides to be placed in position on edge and temporarily strutted. When several runners have been put in vertical, walings, sometimes called 'soldiers,' are placed in position and strutted apart, one or more struts being used for each pair.

SHAFTS.

Shafts may be regarded as short lengths of trench timbered at the ends, and the methods of timbering hitherto described are applicable. The purpose of shafts is, in many cases, to enable tunnels to be driven from them, and they have therefore generally to be sunk to greater depths and to remain open for longer periods than trenches. It is necessary therefore that in these cases even greater care than usual should be taken in the timbering. In deep shafts raking or back props as previously described are of great importance.

The struts are fixed across the short dimension in a small rectangular shaft and the struts at the ends have, in addition to fulfilling their usual function of keeping the walings in position, to act themselves as walings. These end struts are sometimes referred to as 'stretchers.' Then lateral movement is prevented by means of either cleats or straining pieces spiked on to the walings, both being shown in fig. 37, the latter being preferable. The walings also act as struts, being subjected to compression as well as to bending. In the case of a shaft sufficiently large to require strutting in both directions as, for example, that shown in fig. 38, the transverse struts are usually continuous timbers, the longitudinal struts being inserted between them in short lengths.

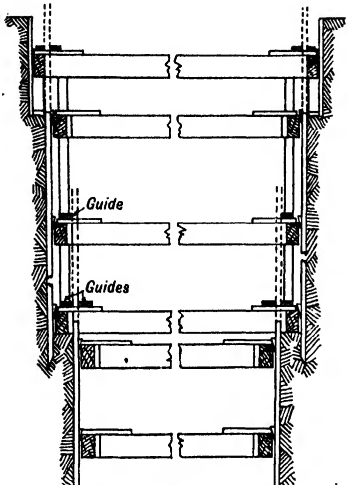


Fig. 34

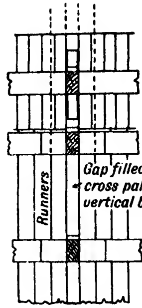


Fig. 35

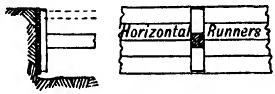


Fig. 36

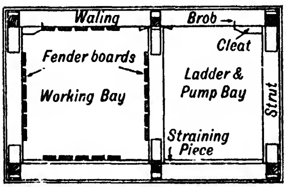


Fig. 37

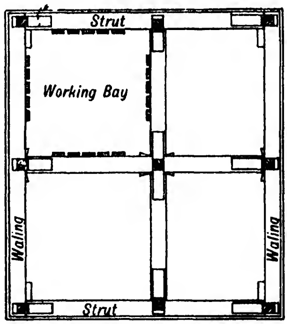


Fig. 38

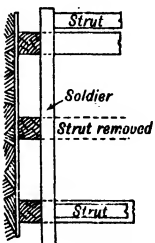


Fig. 39

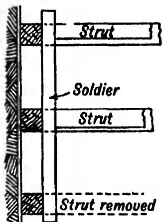


Fig. 40

In designing the timbering of a shaft it is important not to arrange for timbers of such lengths that, owing to the struts of the frames above that for which they are intended, they cannot be readily placed. In some cases walings instead of being the full length of a shaft, as is usual when practicable, have to be in sections. Sometimes it is possible to omit struts or to fix temporary ones in suitable positions until the timbers of the frame below have been placed in position. Props are fixed between the struts of adjacent frames at the points of intersection of transverse and longitudinal struts.

In a shaft divided into bays by struts it is necessary to have a working bay large enough to provide for the passage of skips, brick boxes and the like, and it is frequently necessary therefore to have bays of different sizes. Vertical fenders are nailed between the frames in the working bay to prevent the skips, etc., from being caught by the timbers. Provision must be made in the other bays for ladders, pipes of various kinds and pumps as may be necessary. It is desirable to keep the struts clear of the centre line of a tunnel to be driven from a shaft, in order to facilitate setting out.

In large shafts, in which great pressure is expected to be exerted on the timbering, it is necessary to prevent the upward buckling of the struts by means of raking struts fixed between them and convenient frames above.

Sheet Piling.

In bad ground timber sheet piling is sometimes employed in both trenches and shafts. The method is essentially similar to the use of runners, as sheet piles may be regarded as large runners. The toes of the piles are shod with iron and are cut diagonally so that each pile when driven is forced against the adjacent pile. If necessary the piling may be caulked, hoop iron and yarn being frequently employed for this purpose. Interlocking steel sheet piling is now used more frequently than timber piling.

Removal of Timbers or Parts of Timbers.

Should it be necessary for any purpose to remove one or more struts, this is done by means of 'soldiering,' which consists of placing near each end of the struts to be removed vertical timbers, called soldiers, extending over three or more walings and bearing against the walings from which the struts are to be removed. See figs. 39 and 40. The soldiers are held against the walings by means of soldier struts, and wedges and packings are employed to ensure that the walings are securely held. In the case of shafts the operation just described is necessary when the driving of tunnels is commenced from them.

Refilling Trenches and Shafts.

In refilling excavations only as much timbering is removed as can be done with safety and, unless the work is so situated that settlement is of no importance, it is almost as essential that the ground should be secured during refilling as during excavation. Thorough punning of the refilled material is very important. Sometimes water is used to assist in consolidating the material.

In some cases all timber is left in; in others the frames are removed, boards or runners being allowed to remain. Frequently, however, all timber can be taken out. Frames are removed just in advance of the refilling or construction of the permanent work.

If boards only are being left in, the securing of a setting of piling or tucking boards by filled-in material or permanent work holds the toe of the setting above on account of the overlapping of the settings. Middling boards, being used only in good ground, do not have to be left in. In removing middling boards or tucking boards the ground must remain unsupported between the removal of a setting of boards and refilling, but piling boards and runners can be gradually withdrawn as the filling proceeds. The top one or two frames and settings of poling boards are sometimes left in as a precautionary measure when deep excavations are refilled, even when the lower frames can be removed.

Pressure of Water.

The moment of pressure of water on a retaining wall per each foot of length of wall is $d^3 \times 10{\cdot}4$, where d is the depth of water in feet. This is the moment at 1 foot. The total pressure is $31{\cdot}2\,d^2$, assumed to be concentrated at one-third the depth from the bottom of the water. Hence the moment $10{\cdot}4\,d^3$. The weight of the wall may be assumed concentrated at its centre of gravity, and the effect of the pressure of the water is to turn the wall upon the point O (fig. 41). Calling H the height of the wall in feet, and t its thickness in inches, and assuming one cubic foot to weigh 120 pounds, the moment of stability is $\frac{Ht^2}{2{\cdot}4}$ for rectangular walls, per foot of length; or $d^3 \times 10{\cdot}4$ must not exceed $\frac{Ht^2}{2{\cdot}4}$; or $t^2 = \frac{25\,d^3}{H}$. This enables the thickness of a wall to be determined, that will exactly balance the pressure. Safety would be just secured by adding 50 per cent. to the thickness at the base, and subtracting the same amount from the top thickness, thus giving the wall a sloping back as per dotted line, increasing the acting leverage of the C.G. (centre of gravity) of the section, so that the resultant pressure will now fall within the middle third of the wall thickness. This method is approximate and useful for small construction. Large masonry dams require more special consideration, and may be designed by calculations at each few feet of depth, designing the wall in a series of steps and drawing a final shape to enclose the whole in one harmonious section, which must also fulfil the conditions of strength proper to the material and crushing stress. Special care is necessary that water does not get in between bed joints, as it would greatly assist to turn a wall over.

It is usually considered desirable that the resultant of pressure on a wall should fall within the middle third of its thickness. Thus in fig. 42, if $a\,b$ to any scale represents the total pressure $31{\cdot}2\,d^2$ concentrated at the centre of pressure upon the face of a wall, and $b\,c$ to the same scale is the weight of the wall, the line $b\,c$ being drawn through the C.G. of the cross section, then if the diagonal of the rectangle $a\,b\,c\,d$ cuts the base of the wall within its middle third of thickness, the wall may be assumed fully safe. If the diagonal fall outside the middle third, the thickness of the wall must be increased. It may be made safe by building the wall higher, and thus loading it; but the danger of this method is that a greater area is exposed to wind pressure, and if margins of safety are narrow the addition may be worse than useless in small work. Where a tank wall forms a continuation of an earth retaining wall, for example, special care is necessary.

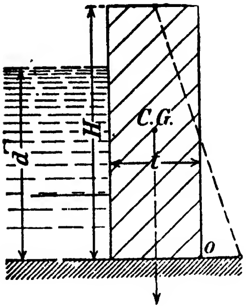


FIG. 41.

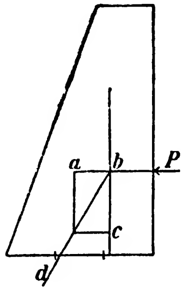


FIG. 42.

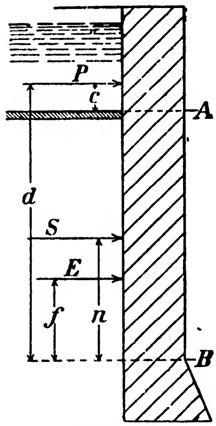


FIG. 43.

The upper part of the wall must be safe at the bed joint level with the bottom of the tank and the wall, taken as a vertical cantilever subject to a lateral thrust at a distance two-thirds of the water depth from the water surface; it must be safe against overturning at any joint lower down. It must also be safe against the added thrust of the earth behind the lower portion plus the further pressure due to the tank full of water above, which must be treated as per the rule on page 459, fig. 45. Thus in fig. 43 the wall must be safe on the line A against the moment of the water pressure P c. It must be safe at the line B against the moment $(P \times d)$ + the moment $(E \times f)$ + the moment $(S \times n)$. Here E is the horizontal component of the earth pressure, and f is one-third A B; S is the horizontal component of the surcharged load, and n is one-half of A B. The safety must be proved at sufficient other depths between B and A and below B. Economy is promoted by sloping the wall against the earth between the points A and B. The wall in the figure is palpably erroneous, being far too thin for its height; it might be made safe by a sufficiency of cross-ties buried in the concrete bottom of the tank, in which case the wall from A to B must be made safe by treating it as a beam exposed to a virtual load at S and E.

Equilibrium of Retaining-Walls.

By Rankine's theory the pressure of earth on a vertical plane, AB (fig. 44), may be considered as acting at a point, P, one-third AB from base A, and in a direction parallel to the surface. Let w = weight of earth in pounds per cubic ft., ϕ = angle of repose of earth, P = total lateral earth pressure per ft. of length, and h = vertical height of wall in feet.

For a bank with horizontal top $$P = \frac{wh^2}{2} \cdot \frac{1 - \sin\phi}{1 + \sin\phi}$$

For bank with any surface slope, θ, of indefinite length

$$P = \frac{wh^2}{2}\cos\theta \frac{\cos\theta - \sqrt{\cos^2\theta - \cos^2\phi}}{\cos\theta + \sqrt{\cos^2\theta - \cos^2\phi}}.$$

For bank with maximum surface slope, ϕ, of indefinite length

$$P = \frac{wh^2}{2}\cos\phi.$$

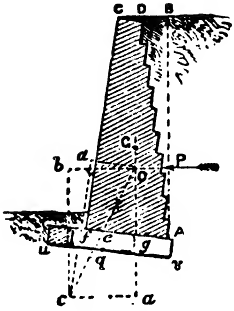


FIG. 44.

For bank with surface slope of definite length an intermediate value is taken.

To resist the pressure there is weight of wall and of earth, ABD, resting on its back, which call W; this acts vertically from centre of gravity, G. The revetment may fail by overturning round f, by crushing at f, by sliding on Af.

Overturning round f.—On any scale make oa = W, ob = P; then oc is resultant (R), acting in direction oe at e. If e falls within base, the revetment will not overturn without crushing the toe at f; but e should fall within the centre third of Af.

Crushing at f.—The resultant, R, must be resolved into two forces, cd (N), perpendicular to, and do parallel to, Af, acting at point e; the former is the crushing force.

Pressure per square inch at f on 1′ length of wall $= \frac{2N}{fg \times 12''}$, where $fg = 3fe$.

The pressure at f should not exceed power of material to resist crushing, divided by factor of safety, 4 to 8 (in practice seldom as high as 8).

Sliding on Af.—Force tending to produce sliding is od; force tending to resist it is N × coefficient of friction at the joint Af od should not exceed N × $\frac{4}{5}$ coefficient of friction.

Surcharged Wall.

If n = height of wall in feet, c = height of surcharge in feet, t = mean thickness of wall, in feet, to sustain horizontal bank; t' = mean thickness of wall, in feet, to sustain bank with indefinitely long natural slope, the factor of safety being about the same as for t; t'' = mean thickness in feet of surcharged wall; then

$$t'' = \frac{nt + 2ct'}{n + 2c}.$$

Effect of Weight of Buildings on Retaining-Walls.

Let w_1 = weight of building per unit of surface; the rest of the notation as at commencement. The total horizontal pressure, P, on revetment, due to the weight of building

$$= w_1(h - x)\frac{1 - \sin\phi}{1 + \sin\phi},$$

acting at $\frac{1}{2}$OB from O; and the total horizontal pressure, P_1, on revetment caused by the earth

$$= \frac{w(h - x)^2}{2} \cdot \frac{1 - \sin\phi}{1 + \sin\phi},$$

acting at $\frac{1}{3}$OB from O.

If the front of the building were farther back from AO the effect would be less, but cannot be determined.

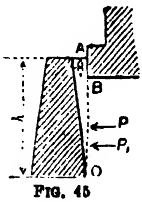


FIG. 45

Rankine's theory assumes the earth a homogeneous, incompressible granular mass with no cohesion. These assumptions are by no means true for some materials, and although giving results erring on the side of safety, more accurate results are obtained by use of Coulomb's theory. This theory assumes that the wall has to support a wedge which has the back of the

wall for one side and a plane termed the 'Plane of Rupture' for the other. By this assumption, where :—

Δ = angle of inclination of the back of the wall to the horizontal.

ε = angle of friction between the base of the wall and the foundation.

θ = the angle of surcharge.

$$P = \frac{wh^2}{2} \cdot \frac{\sin^2(\Delta - \phi)}{\sin^2\Delta \, . \sin(\Delta + \varepsilon)\left\{1 + \sqrt{\dfrac{\sin(+\phi\varepsilon) \, . \sin(\phi - \theta)}{\sin(\Delta + \varepsilon) \, . \sin(\Delta - \theta)}}\right\}^2}$$

Values of ε between concrete and—

Wet clay, 6° to 11°	Dry clay, 17° to 22°
Sand, 22°	Gravel, 22°

For supporting waterlogged ground the pressure due to water and the pressure due to backing are taken separately and then added, but the weight of the backing is reduced by the ratio

$$\frac{\text{Unit weight of backing under water}}{\text{Unit weight of backing in air}} = \frac{w - \left(\dfrac{100 - v}{100}\right) \times 62 \cdot 4}{w}$$

where v = the percentage voids in the backing.

The plane of rupture and total pressure by Coulomb's Wedge Theory can be found graphically by Rebhann's Method.

GRAPHIC DETERMINATION OF EARTH PRESSURE.

(Rebhann's Construction.)

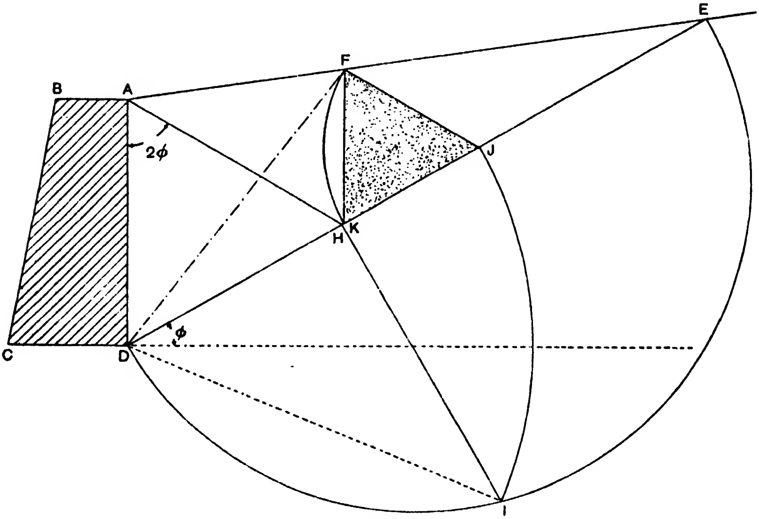


FIG. 46.

(a) *Wall with no surcharge.*—See fig. 46. AE represents the upper surface of the earth, AD the back of the wall, and DE the natural slope of the ground (*i.e.* the angle of repose = ϕ). Set off AH making an angle of 2ϕ with AD and intersecting DE in H. Upon DE as diameter describe a semicircle, and at H draw HI at right angles to DE. From D as centre and with radius DI describe an arc cutting DE in J. Draw JF parallel to AH, then DF is the required plane of rupture. From J as centre and with radius JF describe an arc to cut DE in K and join FK.

Then the area of the triangle FJK in square feet, multiplied by the weight of earth in pounds per cubic foot gives the magnitude of the earth pressure against the back of the wall per foot of its length. This pressure acts at a point two-thirds of the total depth from the surface down the back of the wall AD, and may, in the general case, be taken as acting parallel to DE.

(b) *Wall surcharged.*—See fig. 47. AGE represents the upper surface of the earth. First join DG and then draw A*g* parallel to DG to intersect EG produced in *g*. From *g* draw *gh* making

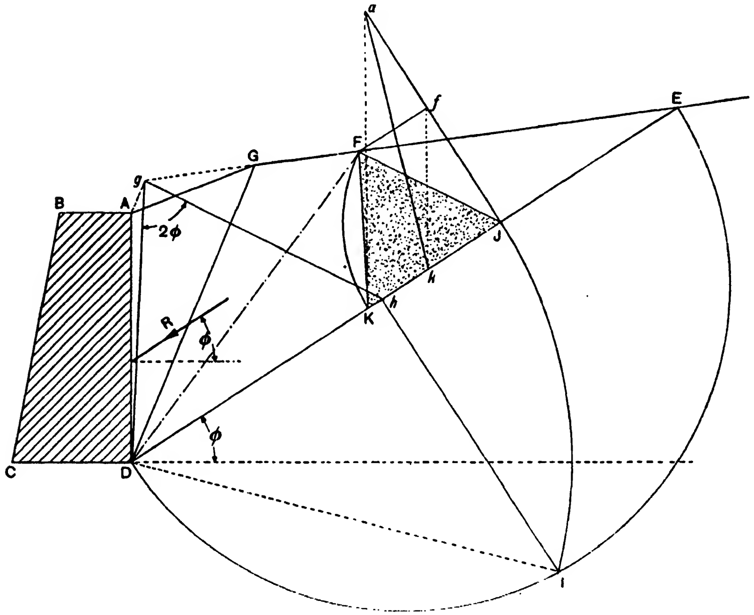


Fig. 47.

an angle of 2φ with D*g*. On DE as diameter describe a semicircle and at *h* draw *h*I at right-angles to DE. From D as centre and with radius DI describe an arc cutting DE at J. Draw JF parallel to *gh*; then DF is the required plane of rupture. The earth pressure triangle FJK is constructed as before. It is most convenient to reduce this triangle to a triangle *a*J*k* having the same area and of a height *a*J equal to the height of the wall. Project F to *f* parallel to DE, and join *a*K. From *f* draw *fk* parallel to *a*K, and join *ak*. Then *a*J*k* is the true earth pressure figure, and by projecting the centre of gravity of this figure on to the back of the wall the 'Centre of Pressure' through which the resultant R must act is obtained.

Foundations of Retaining-Walls.

Find the pressure perpendicular to *uv* (fig. 44). the weights and pressures being taken from point *v* instead of from A. If *q* is centre of pressure, *qu* should nearly equal *qv*, in order that the pressure may be uniformly distributed (this is unnecessary if the pressure at *u* is within what the earth can bear without yielding), and greatest pressure at *u* should not be greater than earth will bear, usually 1 to 1½ ton, or say 2,500 to 3,500 lbs. per square foot. Foundations to be carried deep enough to avoid effects of heat and frost.

Prof. Rankine advises that the distance, fe, of centre of pressure from outside edge in buttresses, chimneys, walls, &c., should be as follows:—In a

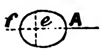
Rectangle (solid).

$fe = \text{or} > \frac{1}{3} fA.$

Square (solid).

$fe = \text{or} > \frac{1}{3} fA.$

Ellipse (solid).



$fe = \text{or} > \frac{3}{8} fA.$

Circle (solid).

$fe = \text{or} > \frac{3}{8} fA.$

Hollow square (factory chimneys).

$fe = \text{or} > \frac{1}{6} fA.$

Circular ring (factory chimney).

$fe = \text{or} > \frac{1}{4} fA.$

It is advisable to check the distribution of pressure over the foundation to see that the pressure nowhere exceeds the permitted maximum. This can be done graphically by means of the method given on page 471 (Graphic determination of distribution of pressure due to eccentric loading).

Example of Brick Retaining-Wall.

Brickwork in mortar 100 lbs. per cubic ft., with a resistance to crushing of 500 lbs. per sq. in. and factor of safety of 8. Coefficient of friction for damp mortar, ·7. Weight of earth (w), 110 lbs. per cubic ft.; angle of repose (ϕ), 37°. Pressure on foundation of sandy gravel not to exceed $1\frac{1}{4}$ ton per sq. ft. Height of wall (h) above footings, 18 ft. Wall sustains horizontal bank level with top. Top of wall two bricks thick, to allow of solid coping. Batter, $\frac{1}{12}$.

Assume for trial fA (fig. 44) $= \frac{1}{3}h = 6$ ft. The back of wall being built with offsets of $4\frac{1}{2}''$ at every 18″ in height, DB will = 3′. Foundations to be 3 ft. below ground.

Weight of wall above bed joint, fA	= 6,750 lbs.	per foot run of wall.	
„ earth in front of AB	= 3,025	„ „	„
Total weight on fA	= 9,775 lbs. = W	„	„

The wall is more likely to overturn at f than at any other point; it will be sufficient, therefore, to inquire into the stability of that joint.

$$P = \frac{wh^2}{2} \cdot \frac{1 - \sin\phi}{1 + \sin\phi} = 4{,}594 \text{ lbs., which acts at P; AP being one-third AB.}$$

Drawing the parallelogram of forces, oc cuts Af at e, making $fe = 9\frac{1}{2}''$.

Resolving oc perpendicular and parallel to Af gives cd = 10,100 lbs.

$$\text{Pressure per sq. in. at } f = \frac{2 \times 10100}{3 \times 9\frac{1}{2}'' \times 12''} = 59 \text{ lbs.}$$

This gives a factor of safety of 10, which is more than sufficient.

Angle ocd will be found to be 20°, tan 20° = ·36, which gives a factor of safety of 2, the coefficient of friction being ·7. The wall is therefore stable.

The pressure on foundations perpendicular to uv will be found to be 14,000 lbs., or $6\frac{1}{4}$ tons, acting at q. To equalise the pressure on the earth, let $qu = qv$, which will make $uv = 12\frac{1}{2}$ ft., giving a pressure of about $\frac{1}{2}$ ton per sq. foot; but as the earth will bear $1\frac{1}{4}$ ton without yielding, a much less width will suffice.

Practical Rules for Retaining-Walls.

A mean thickness of one-fourth height with counterslope of one-fifth, built in steps, will not be far wrong for light soils. This gives a bottom thickness of about one-third height, a centre thickness of one-fourth height, and a top thickness of one-seventh height. A mean thickness one-third height will generally do for stiff clay.

The following rule has been extensively used for brick retaining-walls:—

Top thickness, 1 ft. $10\frac{1}{2}$ in.; face batter of 1 in 6, and after every five courses from top increase thickness by half a brick; front batter should not be more than one-sixth, or it will let in moisture and encourage vegetable growth in the joints. Long walls should have counterforts at back. Walls retaining soils through which water freely passes, such as clean gravel and sand, should have a French drain of loosely packed rubble, &c., running along the back footings, from which good-sized weepholes, from 6 to 10 ft. apart, should lead through the base. With more retentive soils a French drain at least 9 or 12 inches wide should run nearly the whole way up the back of the wall. In some cases extra weepholes at higher levels may be advisable. The mouths of the weepholes should always be carefully protected by loose packing.

Trautwine's Rule for Retaining-Walls.

The horizontal thickness (*ab*, fig. 48) at the base of a vertical or nearly vertical retaining-wall *cdba*, which sustains a backing of either sand, gravel, or earth, level with its top, *cd*, as in the figure should not be less than the following in railroad practice, when the foundations are not more than about 3 feet deep.

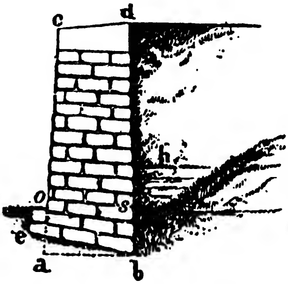


FIG. 48.

When the backing is deposited loosely, as usual, as when dumped from carts, cars, &c.—

Wall of cut stone or of first-class large-ranged rubble in mortar, *ab*.	·35 of its entire vertical height, *db*.
Wall of good common scabbled mortar-rubble or brick	·4 of its entire vertical height, *db*.
Wall of well-scabbled dry rubble	·5 of its entire vertical height, *db*.

With good masonry, however, we may take the height *ds* instead of *db*, and then the above proportions of *ds* will give a sufficient thickness at the ground-line, *os*.

When the backing is somewhat consolidated in horizontal layers, each of these thicknesses may be reduced, but no rule can be given for this.

The offset, *oe*, in front of the wall is not included in these thicknesses.

When, however, the backing is a pure clean sand or gravel, we should use only the full dimensions, inasmuch as the tremor caused by passing trains would neutralise any supposed advantage from ramming materials so devoid of cohesion. Such sand may be rammed with much advantage for the purpose of compacting it in foundations; but a different principle is involved in that case. When it is done, even with cohesive earths, with a view of saving masonry in retaining-walls, it is probable that the expense will generally be found quite equal to that of the masonry saved.

The base, *ab*, in fig. 48, is four-tenths of the height, *bd*. In the foregoing thicknesses at base, the back, *db*, of the wall is supposed to be vertical; and the face, *ca*, either vertical or battered (sloped or inclined backward) to an extent not exceeding about 1½ inch to a foot; which limit it is rarely advisable to exceed in practice, owing to the bad effect of rain, &c., upon the mortar when the batter is great. The base of a vertical wall need not, in fact, be as thick as one with a battered face; but when the batter does not exceed 1·5 inch to a foot the difference is very small.

A mixture of sand or earth with a large proportion of round boulders, paving pebbles, &c., will weigh considerably more than the materials ordinarily used for backing, and will exert a greater pressure against the wall, the thickness of which should be increased, say, about one-eighth to one-sixth part, when such backing has to be used.

The wall will be stronger if all the courses of masonry be laid with an inclination inwards, as at *oeb*; especially if of dry masonry, or if time cannot be allowed (as it always should be when practicable) for the mortar to set properly before the backing is deposited behind it. In retaining-walls, as in the abutments of important arches, the engineer should place as little dependence as possible upon mortar, but should rely more upon the position of the joints for stability.

An objection to inclining the joints in dry (without mortar) walls is that rain-water, falling on the battered face is thereby carried inward to the earth backing, which thus becomes soft, and settles. This may be in a great measure obviated by laying the outer or face courses horizontal; or by using mortar for a depth of only about a foot from the face. The top of the wall should be protected by a coping, *cd*, which had better project a few inches in front. After the masonry has been built up to the surface of the ground, the foundation pit should be filled up; and it is well to consolidate the filling by ramming, especially in front of the wall.

The back, *db*, of the wall should be left rough. In brickwork it would be well to let every third or fourth course project an inch or two. This increases the friction of the earth against the back, and thus causes the resultant of the forces acting behind the wall to become more nearly vertical, and to fall farther within the base, giving increased stability. It also conduces to strength not to make each course of uniform height throughout the thickness of the wall, but to have some of the stones (especially near the back) sufficiently high to reach up through two or three courses. By this means the whole masonry becomes more effectually interlocked or bonded together as one mass, and therefore less liable to bulge. Very thick walls may consist of a facing of masonry and a backing of concrete.

Where deep freezing occurs the back of the wall should be sloped forwards for 3 or 4 feet below its top, as at *co*, fig. 49. which should be quite smooth, so as to lessen the hold of the frost and prevent displacement.

FIG. 49.

A slight bulging in a new wall does not necessarily prove it to be actually unsafe. It is generally due to the newness of the mortar and to the greater pressure exerted by the fresh backing, and will often cease to increase after a few months. It need not excite apprehension if it does not exceed ¼ inch for each foot in thickness at *a*.

After a wall, *abco*, fig. 50, with a vertical back has been proportioned by Trautwine's rule, it may be converted into one with an offsetted back, as *aino*. This will present greater resistance to overturning, and yet contain no more material. Thus, through the centre, *t*, of the back, draw any line, *in*; from *n* draw *ns* vertical; divide *is* into any even number of equal parts (in the figure

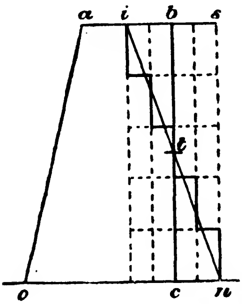


FIG. 50.

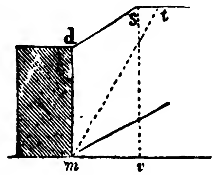


FIG. 51.

there are 4), and divide *sn* into one more equal part (in the figure there are 5). From the points of division draw horizontal and vertical lines for forming the offsets, as in the figure.

When, as in fig. 51, the backing is higher than the wall, and slopes away from its inner edge, *d*, at the natural slope, *ds*, of 1½ to 1, the following thicknesses at base will at least be found sufficient for vertical walls with sand. They are deduced from experiments.

In Table VI., the first column contains the vertical height, *sv*, of the earth, as compared with the vertical height of the wall, which latter is assumed to be 1; so that the table begins with backing of the same height as the wall, as in fig. 48. Without increasing the base these vertical walls may be changed to others with battered faces to any extent not exceeding 1½ inch to a foot, or 1 in 8, without sensibly affecting their stability.

TABLE VI.

Total Height of the Earth, compared with the Height of the Wall above Ground.	Wall of Cut Stone in Mortar.	Good Mortar Rubble or Brick.	Wall of Good Dry Rubble.	Total Height of the Earth, compared with the Height of the Wall above Ground.	Wall of Cut Stone in Mortar.	Good Mortar Rubble or Brick.	Wall of Good Dry Rubble.
	Thickness at Base, in Parts of the Height.				Thickness at Base, in Parts of the Height.		
1	·35	·40	·50	2	·58	·63	·73
1·1	·42	·47	·57	2·5	·60	·65	·75
1·2	·46	·51	·61	3	·62	·67	·77
1·3	·49	·54	·64	4	·63	·68	·78
1·4	·51	·56	·66	6	·64	·69	·79
1·5	·52	·57	·67	9	·65	·70	·80
1·6	·54	·59	·69	14	66	·71	·81
1·7	·55	·60	·70	25			
1·8	·56	·61	·71	or more	·68	·73	·83

When the slope *nr*, fig. 52, of 1½ to 1 starts from the outer edge, *n*, of the wall, greater thickness is required. Poncelet gives the following for this case, for dry sand (Table VII.).

TABLE VII.

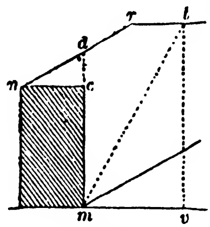


FIG. 52.

Total Depth of Earth, compared with Height of Wall.	Wall of Cut Stone in Mortar.	Wall of Brick-work.	Total Depth of Earth, compared with Height of Wall.	Wall of Cut Stone in Mortar.	Wall of Brick-work.
1	·35	·452	2·4	·762	1·02
1·1	·393	·498	3	·811	1·11
1·2	·439	·548	4	·852	1·18
1·3	·485	·604	6	·883	1·25
1·4	·532	·665	11	·909	1·28
1·5	·579	·726	21	·922	1·31
1·6	·617	·778	31	·926	1·32
1·7	·645	·824	Infinite	·934	1·34
1·8	·668	·847			
1·9	·690	·903			
2	·707	·930			

When the earth reaches above the top of the wall, as in figs. **51** and **52**, the wall is surcharged, and the earth that is above the top is called the surcharge. When the surcharge is carefully deposited above the wall, so as to slope back at a steeper angle than 1½ to 1, as, say, at 1 to 1, theory does not require the wall to be as thick.

WHARF WALLS

are an instance where the thickness should be increased, notwithstanding that the pressure of the water in front helps to sustain them. The earth behind such walls is not only liable to be very heavily loaded when vessels are discharging, but is apt to become saturated with water, especially below low-water level, and thus to exert a very great pressure against the walls. Moreover, the water gets under the wall, and by its upward pressure virtually reduces its weight, and consequently its stability. The same cause, of course, diminishes the friction of the wall upon its base. Such walls are, therefore, very liable to slide if the foundation is smooth and horizontal, and have done so even when the foundation had a considerable inclination backward.

TABLE VIII.

CONTENTS IN CUBIC YARDS FOR EACH FOOT IN LENGTH OF RETAINING-WALLS WITH A THICKNESS AT BASE EQUAL TO ·4 OF THE VERTICAL HEIGHT, IF THE BACK IS VERTICAL.

Ht. Ft.	Cub. Yds.	Ht. Ft.	Cub. Yds.	Ht. Ft.	Cub. Yds.	Ht. Ft.	Cub. Yds.	Ht. Ft.	Cub. Yds.	Ht. Ft.	Cub. Yds.
1	·013	10½	1·38	20	5·00	29½	10·9	48	28·8	74	68·5
1½	·028	11	1·51	20½	5·25	30	11·3	49	30·0	76	72·2
2	·050	11½	1·65	21	5·51	31	12·0	50	31·3	78	76·1
2½	·078	12	1·80	21½	5·78	32	12·8	51	32·5	80	80·0
3	·113	12½	1·95	22	6·05	33	13·6	52	33·8	82	84·1
3½	·153	13	2·11	22½	6·33	34	14·5	53	35·1	84	88·4
4	·200	13½	2·28	23	6·61	35	15·3	54	36·5	86	92·5
4½	·253	14	2·45	23½	6·90	36	16·2	55	37·8	88	96·8
5	·313	14½	2·63	24	7·20	37	17·1	56	39·2	90	101·3
5½	·378	15	2·81	24½	7·50	38	18·1	57	40·6	92	105·8
6	·450	15½	3·00	25	7·81	39	19·0	58	42·1	94	110·5
6½	·528	16	3·20	25½	8·13	40	20·0	59	43·5	96	115·2
7	·613	16½	3·40	26	8·45	41	21·0	60	45·0	98	120·1
7½	·703	17	3·61	26½	8·78	42	22·1	62	48·1	100	125·0
8	·800	17½	3·83	27	9·12	43	23·1	64	51·2	102	130·1
8½	·903	18	4·05	27½	9·46	44	24·2	66	54·5	104	135·2
9	1·01	18½	4·28	28	9·80	45	25·3	68	57·8	106	140·5
9½	1·13	19	4·51	28½	10·2	46	26·5	70	61·3		
10	1·25	19½	4·75	29	10·5	47	27·6	72	64·8		

If the back is stepped according to the rule on p **462** the proportionate thickness at base will, of course, be increased. Face batter, 1½ inches to a foot, or one-eighth of the height. Back either vertical, or stepped according to Fig. **50**. The strength is very nearly equal to that of a vertical wall with a base of ·4 its height. Experience has proved that such walls, when composed of well-scabbled mortar rubble, are safe under all ordinary circumstances for earth level with the top.

A retaining-wall is usually in greater danger for a few months after its completion than after time has been allowed for the mortar to harden perfectly and for the backing to settle. When there are suspicions of the safety of a new wall, it would be well to place strong temporary shores against it, at about one-third to one-half of its height above ground. In some cases permanent buttresses of masonry may be built for the purpose. They should be well bonded into the wall.

The pressure of the earth backing will be much reduced if the first few feet of its height be made up in thin horizontal layers, to be consolidated by being used by the masons instead of scaffolding, as shown at *h*, fig. 48, p. 461. Frequently this can be done without inconvenience, and at very trifling cost.

Reinforced Concrete Retaining Walls.

These are of two main types, viz. :—

(1) Simple cantilever walls, figs. 53A and B.
(2) Counterfort walls, fig. 54.

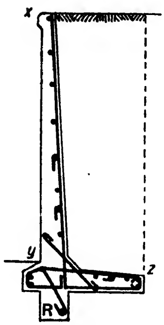


Fig. 53A.

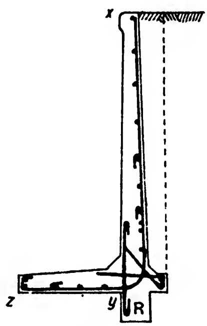


Fig. 53B.

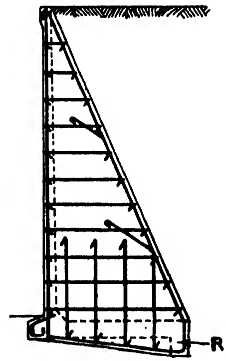


Fig. 54.

Type (1) are suitable for heights up to 15 ft. The portions *xy* are designed as cantilevers loaded with the earth pressure of the bank. In fig. 53A the portion *yz* is a cantilever loaded with the dead weight of the earth above it, and in fig. 53B the loading on *yz* is the upward reaction of the earth.

Type (2) are employed for greater heights and loading. The counterforts and toe are designed as cantilevers, but the wall and heel are designed as continuous slabs spanning the counterforts.

The rib R in all cases serves to resist forward sliding and is designed to take shear.

CHIMNEYS.

Stability of Brick and Masonry Chimney Shafts.

The limit of stability to be observed for chimney shafts is that the wind must never bring tension on any joint. For this to be the case the 'weight moment,' that is, the weight of the length of shaft above any joint, multiplied by the diameter (or 'width' in the case of a square shaft) of the shaft at that joint, must not be less than the 'pressure moment,' *i.e.* the total wind pressure against the length of shaft above the joint, multiplied by the height of the shaft above the joint.

The wind pressure is assumed to act horizontally on the projected area of the shaft. Since in the case of octagonal and circular chimneys this pressure is not normal over the whole surface, but varies from a maximum to zero, the pressure on the projected area is reduced accordingly, thus :—

Square chimneys 56 lbs per square foot (maximum generally assumed in British Isles).
Octagonal ,, 35 lbs. ,, ,, ,, (*i.e.*, $\frac{5}{8}$ of above).
Circular ,, 28 lbs. ,, ,, ,, (*i.e.*, $\frac{1}{2}$ of above).

Direct compression in the section due to weight of brickwork $= c = \frac{W}{A}$ tons per sq. ft.

Stress due to wind pressure $= s = \pm \frac{P \times h \times y}{I}$ tons per sq. ft.

Where W = the weight of brickwork (excluding lining) above any horizontal section (tons).
A = the area of the section (sq. ft.).
I = the moment of inertia of the section in foot units.
y = the width of the section along the line of the direction of the wind (feet).
P = the total wind pressure above the section (tons).
h = the height in feet of the centre of the wind pressure above the section.

For safe design $c - s$ should be not less than 200 lbs. per sq. ft. and $c + s$ should be not greater than the working pressure of the bricks or masonry employed. (For safe loads on materials, see p. 436.)

Dimensions of Chimneys. General Rules and Formulæ.

The Ministry of Health's Byelaws, Series IV, Buildings, Clauses 57–76 are compulsory in many localities (H.M. Stationery Office, London, price 1*s*. 6*d*. net).

Outside diameter at ground should not be less than $\frac{1}{10}$th the height, unless it is supported by some other structure.

Batter varies from 1 in 60 to 1 in 10; 1 in 24 is very common. Or the batter should be from $\frac{1}{16}$th to $\frac{1}{4}$ inch to foot on each side.

Thickness of brickwork: One brick (8 or 9 in.) for 25 ft. from the top, increasing $\frac{1}{2}$ brick (4 to $4\frac{1}{2}$ ins.) for each 25 ft. from the top downwards. If the inside diameter exceeds 5 ft., the top length should be $1\frac{1}{2}$ brick, and if under 3 ft. it may be $\frac{1}{2}$ brick for 10 ft.

Generally a much less height than 100 ft. cannot be recommended for boiler chimneys, as the lower grades of fuel cannot be burned as they should be with a shorter chimney.

Tall Chimneys should always stand alone: if connected with the rest of the buildings, the increased settlement due to their weight on so small a foundation area causes rupture in the masonry. The distance from furnace to shaft should not exceed $\frac{3}{4}$ height of shaft. They should be built and allowed to settle before the connecting flue is made.

Circular form is best; with the same quantity of material it covers a great area, is therefore more stable, and the effect of wind upon it is much less.

In any case the flue should be circular: it can hardly be too large, as it can always be reduced by dampers. It should be lined with a detached skin of firebrick for a certain distance up, increasing in proportion to the heat of the vapours carried off, and separated from the main shaft by an air-space.

Chimneys to receive Vapours of a very high temperature are built altogether of fire-bricks.

Caps tie head of chimney together, but projecting caps catch the wind, and cause oscillation. A dangerous chimney has been saved by removing the cap.

The *Scaffolding* used for building a chimney should be so arranged that it does not prevent the chimney from settling.

The *Intensity of Draught* required varies with the kind and condition of the fuel and the thickness of the fires. Wood requires the least, and fine coal or slack the most. To burn anthracite slack to advantage a draught of $1\frac{1}{2}$ inch of water is necessary, which can be obtained by a well-proportioned chimney 175 feet high.

Design of Tall Chimneys.

45 feet is an ordinary height to serve two steam boilers, but in some towns, as Manchester and Leeds, 90 ft. is the minimum allowed. They are sometimes proportioned for height according to the coal burnt per week of 56 hours, thus:—

4 tons per week	= 75 ft. high.	50 tons per week	= 150 ft. high.
13 „ „	= 100 „	100 „ „	= 188 „
26 „ „	= 120 „	150 „ „	= 200 „

Another rule is to make the height of the chimney three times length of boiler plus twice distance of furthest boiler to chimney; this allows 1 ft. of height for every foot the gases travel round the boiler, and 2 ft. of height for every foot of external flue.

A round chimney should not exceed twenty-five times its internal diameter in height.

A formula relating the internal area of cross section A (sq. ft.), the height H of the chimney (ft.) and the H.P. of the boilers served is:—

$$\text{H.P.} = 3\cdot33(A - 0\cdot6\sqrt{A})H.$$

This is based on 5 lbs. of coal for boiler H.P. and errs on the generous side.

London County Council By-Laws. (*Abstract.*)

London Building Act (Amendment) Act, 1935.

A chimney shaft forming part of a building shall be constructed in a manner approved by the district surveyor subject to (*a*) the standard of stability not being lower than stated below and (*b*) proper precautions are taken to prevent damage to the building through heat or through corrosion of structural steel.

Prior to erection, the site shall be properly cleared of unsuitable material and all cavities filled in an approved manner; if concrete is used it must conform to the by-laws.

The intensity of the calculated pressure on earth shall not exceed that allowed by the district surveyor. It shall comply with the by-law as required for a building (p. 322).

The shaft and footings shall be of suitable brickwork jointed with suitable mortar; the brickwork shall have a batter of at least 1½ ins. in every 10 ft. of height. The thickness of the brickwork at the top and for 20 ft. below shall be at least 8½ ins. and shall be increased at least 4 ins. for every additional 20 ft. or part thereof, measured downwards. Any cap, cornice, pedestal, plinth, string course or other variation shall not be included in the thickness. Footings shall be provided immediately below the base of the shaft and shall spread all round by regular offsets to a projection not less than the thickness of the enclosing brickwork at the base.

Any metal used in connection with a chimney shaft or the footings shall be properly protected.

The height of a chimney shaft shall be measured from the base to the top. It shall not exceed ten times the least width at the base if such base be square or twelve times the external diameter or the least width, respectively, if such base be circular or of any regular polygonal shape.

Any internal lining in a chimney shaft shall be provided as additional to and independent of the thickness of the brickwork and shall not be bonded to the latter.

Flues are ducts not exceeding 80 sq. ins. in area. They shall be surrounded by not less than 4 ins. in thickness of solid bricks or blocks properly bonded and jointed or of concrete cast in position or (except in party walls) of such materials in combination with metal. No flue shall be less than 7½ ins. across in every direction.

Every chimney or stack formed by the combination of two or more chimneys shall, in the case of a flue to a gas-fired appliance, be carried to a height of not less than 18 ins. and in all other cases to not less than 36 ins. above the roof-flat or gutter adjoining but shall not be carried up to a greater height than six times the least width of the chimney at the level of the highest point in such line of junction unless the chimney is adequately secured against overturning. Every chimney which projects from a wall shall be properly bonded to or otherwise properly attached to the wall.

The highest six courses of every chimney or stack constructed of bricks or blocks shall be jointed with cement mortar.

Chimneys having proper soot-doors of an area of not less than 40 sq. ins., fitted in proper frames, may be constructed at any angle; but all other chimneys shall not be inclined at less than 45° to the horizontal. All angles shall be properly rounded. (This requirement does not apply to flues from gas-heated appliances.)

The thickness of the upper side of every chimney where the course of the flue makes an angle of less than 45° with the horizon, shall be at least 8½ ins. except in flues provided solely for gas-fired appliances. Unless required solely for a gas-heated appliance the inside of every chimney shall be properly rendered or pargeted or lined with fireclay or stoneware at least ¾ in. thick or other like incombustible material of sufficient thickness. The spandril angles are to be filled in solid with incombustible material.

Where a chimney passes through a floor or roof within 9 ins. of combustible material or below or against any woodwork, the outside of such chimney must be properly rendered or pargeted.

Proper precautions must be taken to prevent the heat from any chimney or flue injuriously affecting concrete or structural steel.

A flue shall not be used for more than one fire or other heating apparatus except with the approval of the district surveyor.

ARCHES.

Setting-out Arches.

Semi-elliptical Arch (fig. 55).—To set out a *semi-elliptical arch*, draw a line, AB, equal to the span or transverse axis of the ellipse. At the centre of AB draw the semi-conjugate, CD, at right angles, equal to the rise. Then, from the vertex, C, with radius AD or DB equal to half the span, describe an arc intersecting AB in E and F.

The points E, F, will be the foci of the ellipse. If two nails or pegs be fixed in the foci, and a line attached to them equal in length to AB, then the curve traced by a nail, keeping this line stretched, will be the ellipse required the lines EGF, EHF, ECF, EIF, &c., being all equal to span AB, and to each other.

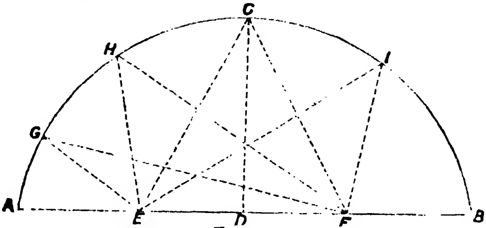


Fig. 55.

Many-centred Circular Arch.—Curves formed of arcs of circles of unequal radii, and similar in appearance to the ellipse, are sometimes adopted for the arches of bridges; with the same rise and span, they may be constructed to give a greater waterway, and in stone bridges they have been preferred by practical stonecutters, but in brick bridges they have no advantage in simplicity over elliptical arches. They may be described with three or a greater odd number of centres. The number of centres will depend on the relation between the span and rise; when the latter is one-third, or a greater fractional part of the former three centres may be used, but if the rise is less than one-third of the span, then five, or a greater odd number, must be taken. In practice it will be found troublesome to describe arcs from a large number of centres, nor, indeed, will occasion be found for using curves of this description. The following is a method of describing a curve composed of three arcs each of 60°.

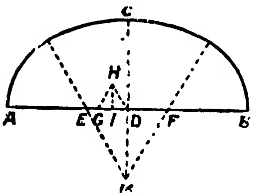


Fig. 56.

Let AB (fig. 56) represent the span, and CD the rise. Take DG = AD – DC, and on it describe an equilateral triangle, DGH; let fall the perpendicular HI, and take IK = HI. Lay off DF = DE. On EF describe the equilateral triangle EFK; then E, F, and K will be the centres required.

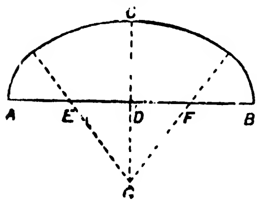


Fig. 57.

For an oval arch, rise one-third, bisect the half-spans AD and DB (fig. 57) in the points E and F, and produce versed sine CD to G, making DG = DC; then E, F, and G will be the three centres with which the curve may be described. In setting out arches, the practical difficulty arises from the elasticity of string. Instead of a string, soft wire about a tenth of an inch in diameter should be used. When the radius does not exceed 12 or 15 feet, a slip of wood may be used, with a nail at each end.

The Design of Brick and Masonry Arches.

A preliminary design is prepared by using the formulæ and tables given below, which is plotted. From this drawing the dead load on the arch can be calculated. The live load to which the arch will be subjected is then determined. For arches in buildings the loading given on p. 437 is suggested; and for bridges, either the Ministry of Transport or British Standard loading are recommended.

The lines of resultant pressure are then drawn for the following conditions:—

(*a*) Dead load over the whole span + live load over half the span.
(*b*) ,, ,, ,, ,, + live load over the whole span.

In the case of a three-hinged arch, the lines of resultant pressure must pass through the hinges; and to draw these lines adopt the same procedure whereby the polygon KRS is drawn in Fuller's Method given on p. 469. If the two pressure lines so produced do not lie wholly within the middle third of the arch thickness, the arch would be subject to tension, and should therefore be thickened until the lines do lie within the middle third.

In the case of a rigid arch the line considered is that which emerges from the arch ring at the haunches with the greatest inclination to the horizontal, and is known as the curve of least resistance, possessing as it does the least horizontal thrust at the abutments. The most expeditious method of plotting this curve is by Fuller's Method given on p. 469.

To find the Depth of Keystone for first-class Cut-stone Arches, whether Circular or Elliptic.

$$\text{Depth of key in feet} = \frac{\sqrt{\text{Rad.} + \text{half span}}}{4} + {\cdot}2 \text{ foot.}$$

For second-class work this depth may be increased about one-eighth part; or for brick or fair rubble, about one-third.

Depths of Keystones for Arches

Of first-class cut stone. For second-class add about one-eighth part. For brick or fair rubble add about one-third part.

Span.	Rise, in Parts of the Span.						
	½	⅓	¼	⅕	⅙	⅛	1/10
Feet.	Key. Feet.	Key Feet.	Key. Feet.	Key. Feet.	Key. Feet.	Key. Feet.	Key. Feet.
2	·55	·56	·58	·60	·61	·64	·68
4	·70	·72	·74	·76	·79	·83	·88
6	·81	·83	·86	·89	·92	·97	1·03
8	·91	·93	·96	1·00	1·03	1·09	1·16
10	·99	1·01	1·04	1·07	1·11	1·18	1·26
15	1·17	1·19	1·22	1·26	1·30	1·40	1·50
20	1·32	1·35	1·38	1·43	1·48	1·59	1·70
25	1·45	1·48	1·53	1·58	1·64	1·76	1·88
30	1·57	1·60	1·65	1·71	1·78	1·91	2·04
35	1·68	1·70	1·76	1·83	1·90	2·04	2·19
40	1·78	1·81	1·88	1·95	2·03	2·18	2·33
50	1·97	2·00	2·08	2·16	2·25	2·41	2·58
60	2·14	2·18	2·26	2·35	2·44	2·62	2·80
80	2·44	2·49	2·58	2·68	2·78	2·98	3·18
100	2·70	2·75	2·86	2·97	3·09	3·32	3·55
120	2·94	2·99	3·10	3·22	3·35	3·61	3·88
140	3·16	3·21	3·33	3·46	3·60	3·87	4·15
160	3·36	3·44	3·58	3·72	3·87	4·17	
180	3·56	3·63	3·75	3·90	4·06	4·38	
200	3·74	3·81	3·95	4·12	4·29		
220	3·91	4·00	4·13	4·30	4·48		
240	4·07	4·15	4·30	4·48			
260	4·23	4·31	4·47	4·66			
280	4·38	4·46	4·63				
300	4·53	4·62	4·80				

In large arches it is advisable to increase the depth of the archstones toward the springs; but when the span is as small as about 60 to 80 or 100 feet, this is not at all necessary if the stone is good, although the arch will be stronger if it is done. In practice this increase, even in the largest spans. does not exceed from ¼ to ½ the depth of the key, although theory would require much more in arches of great rise.

Thickness of Arches.

Centre of pressure at every joint of arch and abutment should fall within centre third.

Thickness of arch at crown $= \sqrt{r} \times B$, where r is the longest radius of curvature at intrados B, as deduced from practice, is as follows:—

		Feet.
Arch above ground standing solitary between abutments	B =	0·12
Arch forming one of series of arches with piers between	=	0·17
Underground archway in hard materials, such as rock or conglomerate	=	0·12
Underground archway in gravel or firm earth	=	0·27
Underground archway in wet clay or quicksand	=	0·48

Fuller's Method of drawing the Curve of Least Resistance for a Rigid Masonry or Brick Arch. (See fig. 58.)

Draw the outline of the suggested arch and mark off by dotted lines the middle third.

Draw a loading diagram showing the dead load ABCD over the whole arch, and the live load EFGB over half the arch.

Divide the diagram into an even number of parts having approximately equal horizontal length. Assume the load represented by the area of each part to act at its centre of gravity as a concentrated load, *i.e.* (1), (2), (3), etc.

Draw a polar diagram for these loads P*xy* with any pole P and from it plot the funicular polygon KLM.

It is now necessary to draw a polygon for this loading to pass through the centres of the haunches K and R and a point S selected at random on the maximum vertical, *i.e.* (4).

PH is drawn on the polar diagram parallel to the closing line of the polygon KM, and intersecting the load line at H. From H, the horizontal line P_1H is drawn so that the polar distance h_1 of $P_1 = \frac{LT}{SV} \times h$ where $h =$ the polar distance of P.

A polar diagram P_1xy is then drawn, and from it the polygon KSR is plotted.

The load lines (1), (2), (3), etc. will cut this polygon in points 1, 2, 3, etc. Select at random two points X and W in KR produced and join XS and SW. Project points K, R, 1, 2, 3, etc. horizontally on to these lines to intersect them in X, W, I, II, III, etc., and drop verticals at each point. On to these verticals project horizontally the intersection of the middle third of the arch with the load lines from which they are derived, obtaining the figure *a*, *b*, *c*, *d*, *e*, *f*. This figure is a distorted outline of the middle third having the same degree of distortion as was given to the line of resistance K, R, 1, 2, 3, etc., by projecting it on to the straight lines XS and SW. The Curve of Least Resistance similarly distorted will be represented by the two straight lines $\alpha\beta$ and $\beta\Delta$ which have the greatest inclination to the horizontal and lie wholly within the figure *a*, *b*, *c* *d*, *e*, *f*. If two straight lines cannot be drawn to lie wholly within the figure, the arch thickness must be increased. It will be noted that in the present case α, β and Δ coincide with *f*, *b* and W, but this is accidental and not essential.

The intersections of $\alpha\beta$ and $\beta\Delta$ on the verticals *f*, *a*, I, II, III, etc., are now projected horizontally back to the verticals K, R, 1, 2, 3, etc., thus giving the required curve $Q\beta R$.

For the reactions due to the curve, join QR intersecting ST in *t*. The rise of the original line

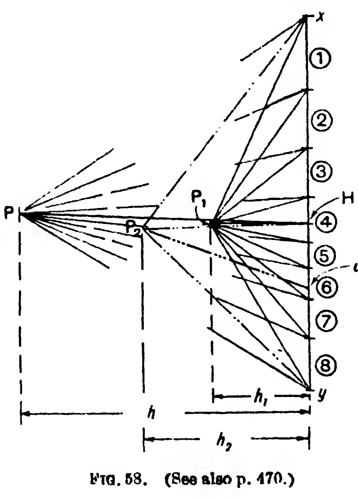


Fig. 58. (See also p. 470.)

FULLER'S METHOD OF DRAWING THE CURVE OF LEAST RESISTANCE FOR A RIGID MASONRY OR BRICK ARCH.

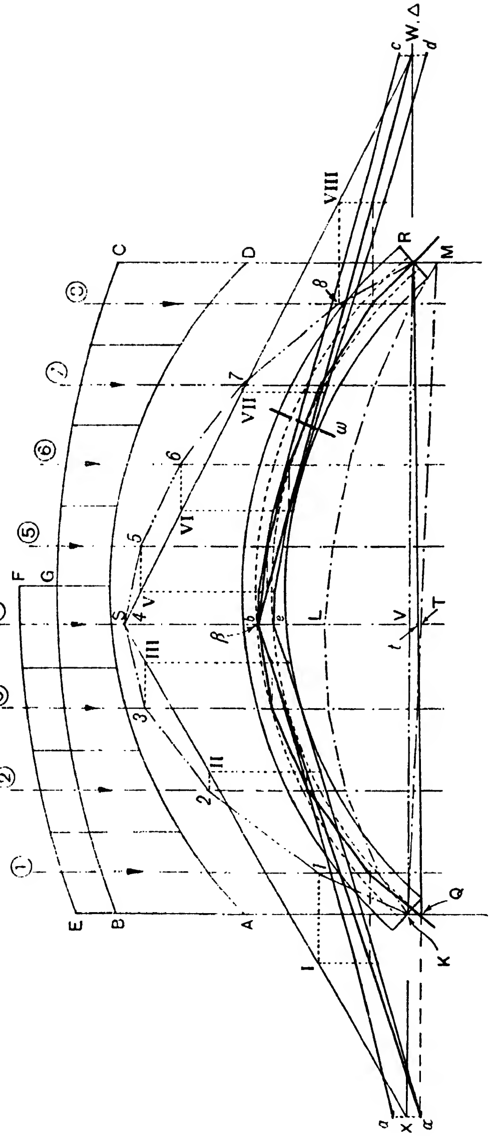


FIG. 58.

of resistance $= SV$ and the rise of $\alpha\beta\Delta = bt$. The horizontal thrust being inversely proportional to the rise, the horizontal thrust for $\alpha\beta\Delta = h_2 = h_1 \times \frac{SV}{bt}$ where h_1 is the horizontal thrust for the linear arch KSR, as scaled from the polar diagram.

From H draw P_2H parallel to QB, making the perpendicular distance of P_2 from $xy = h_2$, Then P_2 is the correct pole for the curve of least resistance and P_2x and P_2y are the reactions at the haunches, and are parallel to the direction of these reactions.

The intersection of the curve on any section ω gives the centre of pressure on that section and the corresponding ray $P_1\omega$ on the polar diagram gives the magnitude and direction of the pressure.

Where the centre of pressure does not coincide with the centre of gravity of the section, the pressure over the section will vary uniformly from a maximum at the side nearest to the centre of pressure to a minimum on the far side.

The variation of pressure over the section can be determined graphically as shown below, which method can be equally applied to loading on retaining wall foundations, bridge piers or any other form of eccentric loading.

Graphic Determination of Distribution of Pressure due to Eccentric Loading.
(See fig. 59.)

Assume a load P to be acting on unit length of a surface of width AB, and the centre of pressure to be at the point E. This load can be resolved into a normal load R, and a tangential load T. T should not exceed the resistance to sliding or cause greater than the permitted shear stress, as the case may be. Let C and D be the limits of the middle third of AB.

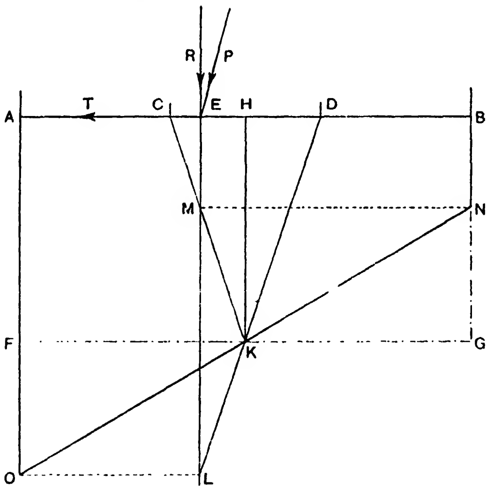


Fig. 59.

Draw the rectangle ABGF making $AF = \frac{R}{AB}$ to some convenient scale. This rectangle represents the pressure distribution on AB for a load R applied at the centre H.

From H and E draw HK and EL perpendicular to AB. Let HK cut FG in K. Join OK cutting EL in M, and draw DK produced to cut EL in L.

Project M horizontally to cut BG in N, and L to cut AF produced in O. Then AO equals the maximum and BN the minimum unit pressures on AB to the same scale as AF.

The figure ABNO is the pressure diagram for the surface. It is readily perceived that where E coincides with O, the figure would be a triangle as B and N would coincide indicating no pressure at B; and should E lie within the length AO, the perpendicular at E would cut KO produced above the line AB and therefore NB would be negative indicating tension.

Piers.

Thickness of piers, generally from ⅛ to ⅙ span. Short piers have vertical sides; high piers may have side batter of $\frac{1}{16}$.

In viaducts and long bridges, 1 pier in every 5 or 6 may be thickened as an abutment pier, so that an accident to a part may not cause the destruction of the whole.

In examining the stability of piers the live load is taken over the whole span on one side of the pier, and the dead load only, over the arch on the other side. The Resultant Pressure lines are then continued down to the foundations modified by the dead load of the pier and by each other giving a common resultant. For stability this resultant must remain in the middle third of the pier, and also it must not enter the ground at an angle to the horizontal less than the angle of friction. In addition the maximum pressure on the foundation must not exceed the maximum permitted bearing pressure for the particular ground.

For abutments the procedure and requirements for stability are the same except that in this case the earth pressure behind the abutments must be considered in place of the thrust from the adjacent arch.

Centering for Arches.

Centering is made of such stuff as is available. Fig. 60 shows what would be suitable for a 20 ft. arch of brick 2 ft. 3 ins. thick. The king post and struts might be of 2-in. stuff. The lagging is of 1½-in. plank, with centres 3 ft. apart; if 4 ft. to 5 ft. apart it should be of 2-in. plank. The struts are often placed at right angles to the arch, as shown in dotted lines.

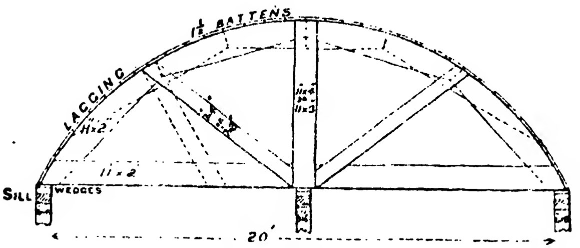


FIG. 60.

In centering for *concrete arches* the lagging should be of narrow battens, 3 ins. to 4½ ins., laid close, the joints being run in with whiting and plaster of Paris. The following dimensions have been followed:—25 ft. 6 ins. span: centres were 2 ft. apart, of 9-in × 1½-in stuff; lagging 4½ ins. × 1½ in., a 9-in. × 3-in. plank cutting into four for lagging, into two for ribs. There were two lines of intermediate supports. For flat or prismoidal roof of concrete 2-in. or 3-in. sheeting was used, on centres of 3-in. stuff, 4 ft. to 6 ft. apart.

The centres rest by the ends of their chords upon wooden *striking* or *lowering wedges*, for striking or lowering the centre after the completion of the arch. These consist of pairs of wedge-shaped blocks, as W, W, fig. 61, of hard wood from 1 ft. to 2 ft. long, about half as wide, and a quarter or more as thick (sufficient to lower the centre from, say, 2 ins. to 6 ins. or more, according to span and other circumstances).

It is of the utmost importance, especially in large arches, that the *centres should be lowered very slowly*, otherwise the momentum acquired by so heavy a body as an arch descending suddenly, even but 2 ins. or 3 ins., might possibly affect its shape, or even its safety. For this reason the wedges should not have a taper steeper than about 1 in 6 or 8 for arches of less than about 50 ft. span, or than 1 in 8 or 10 for larger spans. Vertical lines at equal distances apart should be drawn on the long sides of the wedges, as a guide for lowering them all to the same extent at a time; and this should not exceed in all about half an inch a day in intervals of about an eighth of an inch for 50 ft. spans, or about ·1 to ·25 of an inch per day in all for spans over 100 ft. Slowness is especially to be recommended in *brick arches*, not only because their greater number of joints exposes them to greater derangement of shape, but because even good brick has much less than the average crushing-strength of good granite, limestone, or sandstone, and therefore is far more liable than they to crack, or even to crush, when the strains are thrown almost entirely upon their edges.

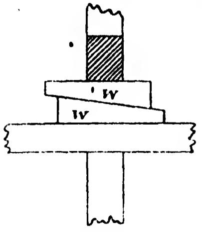


FIG. 61.

Easing and Striking Centres.—Much depends on temperature, mass of masonry or brickwork, composition of mortar, etc. With ordinary lime, centres should be eased while mortar is quite green, as soon as the arch is loaded to stability, and struck on an average after five to eight days. With small arches, one or two and a half bricks thick, 6 ft. to 8 ft. span, built in dry weather, the centres may be struck in three or four days. With slow-setting cement, if the arch be small and slight, so that it can be loaded to stability before the mortar of portion first commenced has set, the centres may be eased, as with lime, but they may be struck sooner, and the limit may be taken as the time at which the joints would become too hard set to admit of being raked out for pointing without seriously injuring the brickwork. If, in consequence of the cement being strong and quick-setting, or of the span or thickness of the arch being considerable, the arch cannot be completed before the portion first commenced has set, it is doubtful whether the centering should be eased at all, for fear of disturbing the setting of the cement. Centering for concrete should not be disturbed for at least fourteen days from the time of laying the concrete.

The main points to attend to are—to complete the arch to stability rapidly, and to prevent disturbance of the work while setting, by traffic over it, or vibrations from adjoining work, otherwise the setting power, of cement specially, is destroyed.

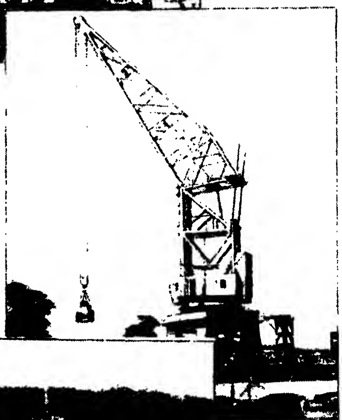
See also Descriptive Section XIII.

SECTION XIV

BRIDGES AND BRIDGEWORK (METAL)

(Revised by J. D. W. Ball, A.M.I.C.E.

ARROL



Fixed and Opening Bridges of all types, Steel Frame Buildings, Cranes and Transporters, Mechanical Engineering Work, Dock Gates, Sliding and Floating Caissons, Air Locks, Hydraulic Machinery, Pipe Lines, Spiral Castings, Surge Tanks, Sluices and Equipment for Hydro-Electric Stations.

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SECTION XIV

PART I

STEEL BRIDGEWORK

TYPES — ECONOMICAL SPAN — PROPORTIONS — STANDARD CLEARANCES — LIVE LOADS — BRIDGE FOUNDATIONS — PERMISSIBLE STRESSES — WIND PRESSURE — CENTRIFUGAL FORCE—MOMENTUM — IMPACT—TEMPERATURE—BEARINGS—FLOORING—CAMBER, RIVETING, ETC.

(Revised by J. D. W. Ball, A.M.I.C.E.)

TYPES.

Bridges consist of two parts:

(*a*) The substructure, comprising abutments, piers, caissons, and other forms of foundations.

(*b*) The superstructure, which comprises the bridge structure itself.

There are three known methods of carrying weight over space—*i.e.* by means of:

(1) The beam; (2) the arch; (3) the rope.

All bridges can be reduced to one, or a combination, of these types.

(1) THE BEAM.—The strength of the beam depends on transverse bending. The theoretic stress in the extreme fibres is inconsistent with experiments but the so-called discrepancy diminishes with the thickness of the web and the ratio of depth to span of the beam.

The principal groups of girder bridges are:

(A) Girders of uniform depth; (B) girders of uniform horizontal stress.

Group A: Parallel Girders.—In this case the shearing force is resisted by the web, and the diagram of moments forms the stress diagram.

Group B: Parabolic Girders—In this case, for uniform loading the shearing force is resisted by the flanges and the diagram of moments forms a diagram of depth.

All these types can be used in their inverted form, and it is from these forms that the Arch and Rope are evolved.

(2) The Arch.—For any given system of loading a theoretical arch can be designed so as to be in compression only, the neutral axis of the arch coinciding with the line of pressure: but unless it can resist bending, the arch will be in a state of unstable equilibrium. The horizontal stress varies with the rise.

(3) The Rope.—The shape which the loaded cable takes up is the same as the link polygon for the given load system drawn from one support to the other with a polar distance equal to the horizontal component of the pull in the cable. Stable equilibrium is obtained under changes of load. The horizontal stress varies with the dip.

Economical Span.

In bridging a large opening by a number of small spans, the economical span may be approximated in the following manner:—

Let

P = cost of one pier; G = cost of main girders for one span erected; n = number of small spans; l = length of small spans; L = total span

then,

$$n = \frac{L}{l}.$$

$$\text{Cost of piers} = (n-1)\,P.$$

Cost of main girders $= nG$, and G may be taken as proportional to the square of the span, so that,

$$G = al^2,$$

therefore

$$\text{Total cost} = C = (n-1)\,P + nal^2,$$

where a is a constant.

To get the minimum value of C, differentiate with respect to l and put it equal to zero. Therefore

$$\frac{dC}{dl} = 0.$$

But

$$C = \left(\frac{L-l}{l}\right) P + \frac{Lal^2}{l}.$$

$$= \frac{LP}{l} - P + Lal,$$

therefore

$$\frac{dC}{dl} = \frac{-LP}{l^2} + La = 0$$

therefore

$$\frac{P}{l^2} = a,$$

or

$$P = al^2 = G.$$

From this it follows that the most economical condition is when the cost of one pier equals the cost of one main girder for one span.

If G equals cost of 100 ft. span, the result may be expressed:

$$\text{Economical span} = \frac{100\sqrt{P}}{\sqrt{G}}.$$

(Ency. Brit.)

Economical Proportions of Girders.

Plate Girders.

Let

x = depth of girder in ins.; W = weight of girder in lbs.; f = allowable working stress (pounds per sq. in.) on gross area; t = thickness of web in ins.; L = extreme length of girder in feet; M = total bending moment at centre (inch-pounds).

Then the weight of the web is $\frac{10}{3}Ltx$, and that of the flanges, assuming the flange plates are the required theoretical lengths, $= \frac{10}{3}\left(1 \cdot 6 \frac{ML}{fx}\right)$,

therefore

$$W = \frac{10}{3} Ltx + \frac{10}{3}\left(1 \cdot 6 \frac{ML}{fx}\right),$$

$$\text{and W is a minimum when } x = 1 \cdot 27 \sqrt{\frac{M}{ft}}.$$

This formula is based upon the assumption that no portion of the web is included in flange area.

If the resistance of web is considered, then

$$x = 1 \cdot 46 \sqrt{\frac{M}{ft}}.$$

If the flanges are of constant section throughout, the foregoing formulæ become

$$x = 1 \cdot 41 \sqrt{\frac{M}{ft}} \text{ (web neglected); } x = 1 \cdot 63 \sqrt{\frac{M}{ft}} \text{ (web considered).}$$

Lattice Girders.

In trusses of the N type the maximum stiffness or minimum deflection is obtained when

$$x = \frac{d}{2}\sqrt{\frac{y+2}{y-2}} \times y,$$

where

d = length in feet between verticals; y = number of bays.

(*Sir Wm. Arrol & Co., Ltd.*)

For plate girders, the economic depth may be taken roughly as from $\frac{1}{10}$ to $\frac{1}{15}$ of the span, $\frac{1}{12}$ often being adopted; and for lattice girders should be not less than $\frac{1}{10}$ of the span, preferably $\frac{1}{8}$.

If shallower girders are necessary, the sections must be increased, so that the maximum deflection will not be greater than if the limiting ratio had not been exceeded.

Breadth of Flanges.

For plate girders this may be taken as about $\frac{1}{3}$ of the depth, or $\frac{1}{30}$ to $\frac{1}{40}$ of the span.

The width of booms in lattice girders should not be less than $\frac{1}{12}$ of the unsupported distance.

Cross Girders.—Depth should not be less than $\frac{1}{10}$ of the span, preferably $\frac{1}{8}$.

Rail Bearers should have a depth of not less than $\frac{1}{15}$ of the span.

Spans, Standard Clearances, etc.

Over-bridges carrying roads over railways are usually of the following spans (see fig. 1):

Over single line, 16 to 18 ft. Over double line, 28 to 30 ft.

Widths of over-bridges, between inside of parapets, are regulated by Act of Parliament, as follows, the roads being of the widths given for 50 yards each way:

Turnpike roads, 35 ft. Other public carriage roads, 25 ft. Private roads, 12 ft.

Under-bridges carrying roads under railway, span and headroom fixed by Act of Parliament.

Turnpike road, clear headroom, 12 ft. at springing, 16 ft. for centre 12 ft. of arch. Where tramways cross the bridge a minimum headroom of 18 ft. should be provided.

Other public and carriage roads, clear headroom 12 ft. at springing, 15 ft. for centre 10 ft. of arch.

Private roads, clear headroom 14 ft. for centre breadth of 9 ft. of arch.

In all cases, under-bridges must have a parapet at least 4 ft. 6 ins. above rail level.

STANDARD CLEARANCES FOR BRIDGES.

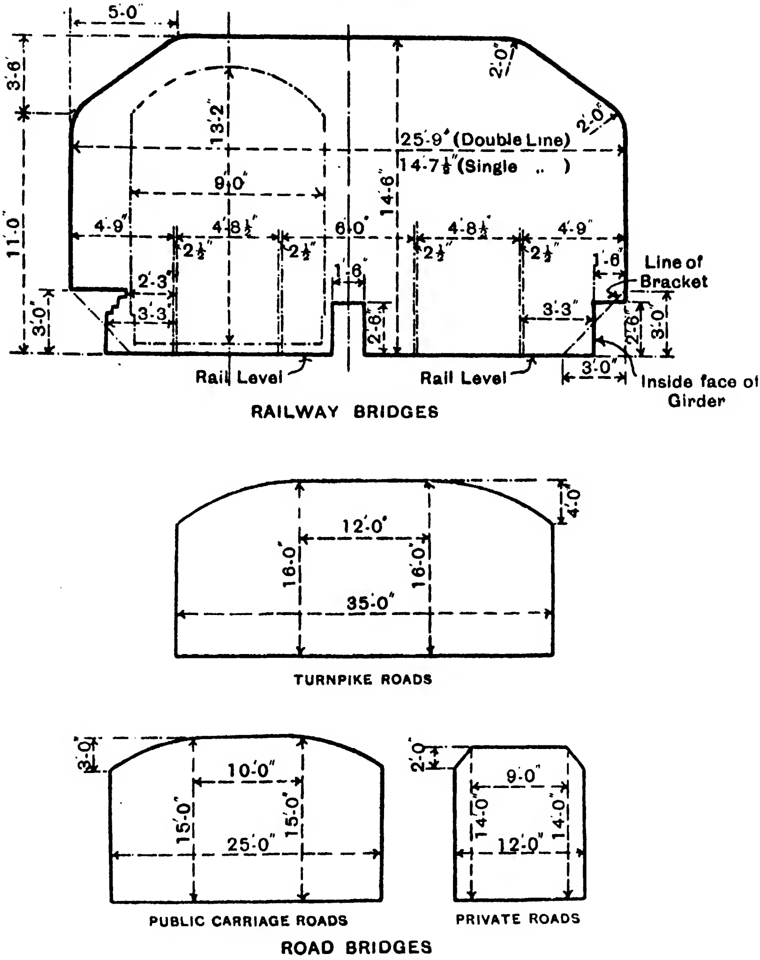


FIG. 1.

Live Loads on Bridges.

Rolling Load on Railway Bridges.

(1)

The effective weight of the rolling load, per foot lineal of each line of way, will depend upon the width of span. The heaviest train would consist of locomotives coupled closely together, but while a bridge of short span might frequently be covered by such a train, or by the two engines that often head the train, a bridge of very long span would never be subjected to such a load per foot lineal over its entire length. But for spans shorter than the length of one engine, the load may be very much greater than the weight of the engine per foot of its extreme length. Taking the worst position of the wheel-loads, and the heaviest class of engines now in use upon English lines of railway, the equivalent distributed load per foot lineal may be estimated as follows:—

For spans of	10	feet, a load of	4·40	tons per foot.	For spans of	100	feet, a load of	2·07	tons per foot.
„	20	„ „	3·25	„ „	„	150	„ „	2·00	„ „
„	30	„ „	2·88	„ „	„	200	„ „	1·90	„ „
„	60	„ „	2·30	„ „	„	300	„ „	1·75	„ „

In designing cross girders the maximum axle load is taken at 22 tons.

These loads are exclusive of any allowance for impact which may be allowed for by varying the unit stress per sq. in. to suit the application of the load:—From 4·5 tons per sq. in. for cross girders to 6·5 tons per sq. in. for main girders 100 feet span and over. The better plan, however, is to calculate the impact allowance from one of the impact formulæ given on p. 500.

The loads given above are considered to be the **greatest** that can be obtained **with** the present loading gauge of British railways.

(2)

Bridge to be designed for a moving load of two engines coupled and a uniformly loaded train or each track. Greatest stresses in any possible circumstances to be taken.

Where not specified take the following:—

***For Main Girders.*—Distributed rolling load per track = 1·5 tons per foot run, and an excess rolling load to occupy any position on the bridge at the same time equal to $15 \text{ tons} + \left(\frac{\text{span in feet} - 10}{10}\right)$ but not greater than 25 tons.**

***For Cross Girders.*—The load on each cross girder from each track:**

For all girders up to 7 ft. centres = 19 tons,
„ „ beyond 7 ft. „ = 1·5 tons per ft. run of track plus an excess load of

$$10 \text{ tons} - \left(\frac{10 - \text{centres of cross girders in ft.}}{5}\right),$$

but not greater than 10 tons. ***(Sir Wm. Arrol & Co., Ltd.)***

Locomotive Weight Distribution on Bridges.

In the case of the three-cylinder tank engines—London and North Eastern Railway—these engines are of the 0-8-4 type, 31 ft. 2 ins. total wheel base, 45 ft. 2¼ ins. long over buffers. The total weight in working order is 194 tons 9 cwt., the weight on the coupled wheels being 75 tons 14 cwt.

The general weight distribution is

	T.	C.
Weight per ft. on coupled wheels	4	9
„ „ „ engine wheel base	3	7
„ „ „ over buffers	2	6

For further details of locomotives refer to 'Weights of Locomotives and Rolling Stock,' Section XXXII, Part II.

British Standard Unit Loadings for Railway Girder Bridges and Highway Girder Bridges.

TABLE I.

British Standard Unit Loading for Railway Bridges for a Single Line of Way.

For Gauges of 4 ft. 8½ ins. and over.

FIG. 2.

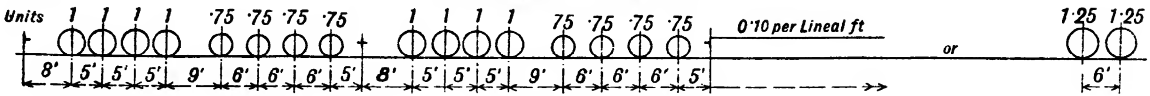


Diagram of Unit Loading.

In this table the unit of axle load is taken as 1 ton (2,240 lbs.).

For convenience of calculation the train load has been taken as equivalent to a train of vehicles with axles spaced 10 ft. apart, each axle carrying a load of one unit, the first axle being spaced 5 ft. from the end.

For British railways the loading recommended in the Requirements for Passenger Lines and Recommendations for Goods Lines of the Minister of Transport in regard to Railway Construction and Operation (1925) is 20 units for heavy main lines, with lower multiples for lines constructed for lighter loading.

1	2	3	4	5	6	7	8	1	2	3	4	5	6	7	8
	Main Girders and Rail Bearers.						Cross Girders.		Main Girders and Rail Bearers.						Cross Girders.
Span.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.	Span.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.
	At centre of span.		At ¼ point of span.		At centre of span.	At abutments.			At centre of span.		At ¼ point of span.		At centre of span.	At abutments.	
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.	Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
4	1·250	2·500	0·938	2·500	0·625	1·250	1·250	10	3·125	2·500	2·813	3·001	0·625	1·750	2·000
5	1·563	2·500	1·172	2·500	0·625	1·250	1·250	11	3·636	2·645	3·281	3·182	0·625	1·818	2·182
6	1·875	2·500	1·406	2·500	0·625	1·250	1·333	12	4·219	2·813	3·750	3·333	0·625	1·875	2·333
7	2·188	2·500	1·641	2·500	0·625	1·429	1·571	13	4·808	2·959	4·219	3·462	0·673	1·923	2·462
8	2·500	2·500	1·875	2·500	0·625	1·563	1·750	14	5·500	3·143	4·688	3·572	0·714	1·964	2·571
9	2·813	2·500	2·344	2·778	0·625	1·667	1·889	15	6·250	3·333	5·156	3·667	0·750	2·000	2·717

16	7·000	3·500	5·625	3·750	0·781	2·125	2·844	49	45·190	7·378	35·859	7·806	1·395	4·367	6·367
17	7·750	3·647	6·094	3·824	0·809	2·235	2·956	50	46·752	7·480	37·125	7·920	1·415	4·420	6·460
18	8·500	3·778	6·563	3·889	0·833	2·333	3·056	52	50·091	7·706	39·750	8·154	1·452	4·519	6·712
19	9·329	3·928	7·031	3·947	0·855	2·421	3·145	54	53·583	7·938	42·375	8·370	1·486	4·648	6·944
20	10·313	4·125	7·500	4·000	0·875	2·500	3·225	56	57·076	8·154	45·000	8·571	1·518	4·768	7·196
21	11·298	4·304	8·250	4·190	0·893	2·571	3·333	58	60·569	8·354	47·625	8·759	1·547	4·879	7·466
22	12·284	4·467	9·000	4·364	0·909	2·636	3·432	60	64·259	8·568	50·250	8·933	1·575	4·983	7·717
23	13·272	4·616	9·750	4·522	0·924	2·696	3·522	62	68·133	8·791	52·875	9·097	1·613	5·081	7·952
24	14·260	4·753	10·500	4·667	0·938	2·750	3·635	64	72·184	9·023	55·750	9·292	1·648	5·203	8·203
25	15·250	4·880	11·250	4·800	0·950	2·830	3·740	66	76·428	9·264	58·750	9·495	1·682	5·318	8·439
26	16·240	4·997	12·094	4·962	0·962	2·904	3·837	68	80·673	9·491	61·750	9·686	1·713	5·456	8·676
27	17·231	5·105	12·984	5·129	0·972	2·972	3·954	70	84·918	9·705	64·875	9·886	1·743	5·586	8·914
28	18·223	5·207	13·875	5·286	0·982	3·036	4·063	72	89·390	9·932	68·250	10·111	1·771	5·736	9·139
29	19·216	5·301	14·766	5·431	0·991	3·095	4·164	74	94·014	10·164	71·625	10·324	1·807	5·878	9·351
30	20·375	5·433	15·656	5·567	1·000	3·150	4·283	76	98·639	10·383	75·250	10·561	1·842	6·013	9·553
31	21·563	5·565	16·547	5·694	1·032	3·226	4·395	78	103·264	10·591	79·000	10·803	1·875	6·141	9·756
32	22·750	5·688	17·438	5·813	1·063	3·297	4·500	80	107·928	10·793	82·750	11·033	1·906	6·272	9·963
33	23·938	5·803	18·328	5·924	1·091	3·364	4·621	82	112·917	11·016	86·875	11·301	1·936	6·405	10·159
34	25·125	5·912	19·313	6·059	1·118	3·426	4·735	84	117·907	11·229	91·000	11·566	1·964	6·533	10·369
35	26·313	6·014	20·344	6·200	1·143	3·486	4·843	86	123·007	11·442	95·125	11·798	2·000	6·663	10·570
36	27·500	6·111	21·375	6·333	1·167	3·542	4·965	88	128·506	11·682	99·250	12·030	2·034	6·795	10·761
37	28·688	6·203	22·406	6·459	1·189	3·615	5·081	90	134·006	11·912	103·375	12·252	2·067	6·922	10·944
38	29·875	6·289	23·438	6·579	1·211	3·684	5·191	92	139·506	12·131	107·500	12·464	2·098	7·052	11·120
39	31·155	6·391	24·469	6·692	1·231	3·750	5·295	94	145·006	12·341	112·188	12·731	2·128	7·184	11·309
40	32·523	6·505	25·500	6·800	1·250	3·813	5·394	96	150·506	12·542	116·875	12·986	2·156	7·310	11·490
41	33·891	6·613	26·531	6·902	1·268	3·872	5·488	98	156·006	12·735	121·563	13·231	2·184	7·439	11·673
42	35·260	6·716	27·656	7·024	1·286	3·929	5·595	100	161·922	12·954	126·250	13·467	2·210	7·570	11·860
43	36·629	6·815	28·828	7·151	1·302	4·000	5·721	105	177·974	13·560	138·109	14·030	2·276	7·876	12·295
44	37·998	6·909	30·000	7·273	1·318	4·068	5·841	110	195·315	14·205	150·531	14·597	2·355	8·182	12·736
45	39·368	6·999	31·172	7·389	1·333	4·133	5·956	115	213·565	14·857	163·625	15·177	2·426	8·478	13·183
46	40·737	7·085	32·344	7·500	1·348	4·196	6·065	120	232·281	15·485	176·750	15·711	2·492	8·775	13·633
47	42·107	7·167	33·516	7·606	1·362	4·255	6·170	125	251·581	16·101	190·313	16·240	2·556	9·064	14·072
48	43·628	7·271	34·688	7·708	1·375	4·313	6·271	130	271·540	16·710	204·375	16·769	2·631	9·354	14·523

Impact must be added, see p. 501 (British Standard Specification—extract).

TABLE I.—*continued.*

1	2	3	4	5	6	7	8
	Main Girders and Rail Bearers.						Cross Girders.
Span.	Maximum bending moment. At centre of span.	Total equivalent uniformly distributed load. At centre of span.	Maximum bending moment. At ¼ point of span.	Total equivalent uniformly distributed load. At ¼ point of span.	Maximum shear. At centre of span.	Maximum shear. At abutments.	Maximum cross girder or pier reaction.
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
135	291·501	17·274	218·438	17·259	2·711	9·637	14·963
140	312·071	17·833	233·750	17·810	2·793	9·921	15·393
145	333·319	18·390	249·688	18·368	2·886	10·200	15·862
150	354·566	18·910	265·625	18·889	2·973	10·480	16·300
155	376·233	19·418	282·375	19·432	3·055	10·755	16·774
160	398·726	19·936	299·250	19·950	3·136	11·031	17·219
165	421·397	20·431	316·313	20·449	3·219	11·303	17·697
170	445·143	20·948	334·250	20·973	3·297	11·576	18·147
175	468·889	21·435	353·000	21·516	3·381	11·846	18·629
180	492·766	21·901	371·750	22·030	3·461	12·117	19·083
185	517·731	22·388	390·938	22·541	3·543	12·384	19·568
190	542·699	22·850	410·625	23·053	3·624	12·653	20·026
195	567·814	23·295	430·313	23·538	3·703	12·918	20·513
200	594·037	23·761	450·750	24·040	3·785	13·185	20·975
205	620·262	24·205	472·063	24·563	3·863	13·449	21·463
210	647·591	24·670	493·625	25·073	3·938	13·714	21·929
215	675·019	25·117	515·250	25·563	4·012	13·977	22·419
220	702·450	25·544	537·750	26·073	4·091	14·241	22·886
225	730·873	25·987	560·250	26·560	4·167	14·502	23·378
230	759·729	26·425	583·125	27·043	4·239	14·765	23·848
235	789·660	26·882	606·875	27·546	4·311	15·026	24·340
240	819·594	27·320	631·250	28·056	4·388	15·288	24·813
245	850·700	27·778	655·625	28·544	4·461	15·547	25·306
250	881·901	28·221	680·625	29·040	4·532	15·808	25·780
255	913·859	28·670	705·938	29·529	4·602	16·067	26·275
260	946·256	29·116	732·000	30·031	4·677	16·327	26·750
265	979·203	29·561	759·188	30·559	4·749	16·585	27·245
270	1012·871	30·011	786·500	31·072	4·819	16·844	27·722
275	1046·541	30·445	814·625	31·598	4·887	17·102	28·218
280	1081·079	30·888	842·750	32·105	4·961	17·361	28·696
285	1116·016	31·327	871·313	32·611	5·032	17·618	29·193
290	1151·992	31·779	900·375	33·117	5·100	17·876	29·672
295	1188·362	32·227	929·438	33·607	5·168	18·132	30·169
300	1225·806	32·688	959·250	34·107	5·240	18·390	30·650

Impact must be added, see p. 501 (British Standard Specification—extract).

BRITISH STANDARD UNIT LOADING FOR HIGHWAY BRIDGES.

FIG. 3. (Each 10 ft. width of carriageway of the bridge is to be assumed as completely occupied by a succession of Standard trains.)

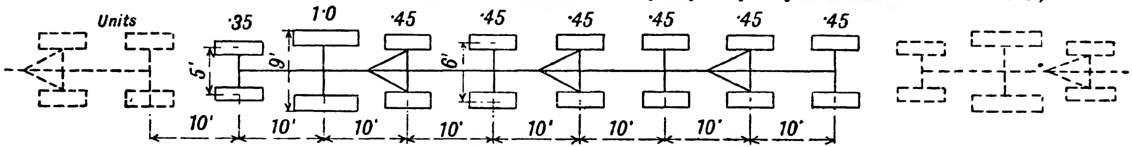


Diagram of Unit Loading.

In this table the unit of axle load is taken as 1 ton (2,240 lbs.).

When the passage of exceptional loads over a bridge is to be anticipated, a special multiple of the unit loading should be specified as applicable to road bearers, cross girders, and short panel lengths.

The multiple recommended by the Minister of Transport as a minimum for all road bridges in Great Britain is 15 units.

1	2	3	4	5	6	7	8
	Main Girders and Longitudinal Rail Bearers.						Cross Girders.
Span.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.
	At centre of span.		At ¼ point of span.		At centre of span.	At abutments.	
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
4	1·000	2·000	0·750	2·000	0·500	1·000	1·000
5	1·250	2·000	0·938	2·000	0·500	1·000	1·000
6	1·500	2·000	1·125	2·000	0·500	1·000	1·000
7	1·750	2·000	1·313	2·000	0·500	1·000	1·000
8	2·000	2·000	1·500	2·000	0·500	1·000	1·000
9	2·250	2·000	1·688	2·000	0·500	1·000	1·000
10	2·500	2·000	1·875	2·000	0·500	1·000	1·000
11	2·750	2·000	2·063	2·000	0·500	1·041	1·073
12	3·000	2·000	2·250	2·000	0·500	1·075	1·133
13	3·250	2·000	2·438	2·000	0·500	1·104	1·185
14	3·500	2·000	2·681	2·043	0·500	1·129	1·229
15	3·750	2·000	2·953	2·100	0·500	1·150	1·267

Impact must be added, see p. 501 (British Standard Specification—extract).

TABLE IV.—*continued.*

1	2	3	4	5	6	7	8
	Main Girders and Longitudinal Road Bearers.						Cross Girders.
Span.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.
	At centre of span.		At ¼ point of span.		At centre of span.	At abutments.	
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
16	4·000	2·000	3·225	2·150	0·500	1·169	1·300
17	4·250	2·000	3·497	2·194	0·500	1·185	1·329
18	4·500	2·000	3·769	2·233	0·500	1·200	1·356
19	4·821	2·030	4·041	2·269	0·500	1·213	1·379
20	5·175	2·070	4·313	2·300	0·500	1·225	1·400
21	5·529	2·106	4·584	2·328	0·502	1·257	1·463
22	5·906	2·148	4·856	2·354	0·505	1·286	1·518
23	6·356	2·211	5·128	2·378	0·507	1·313	1·570
24	6·806	2·269	5·400	2·400	0·508	1·338	1·617
25	7·256	2·322	5·672	2·420	0·510	1·360	1·660
26	7·705	2·371	5·944	2·439	0·512	1·381	1·700
27	8·155	2·416	6·244	2·467	0·513	1·400	1·737
28	8·605	2·459	6·600	2·514	0·514	1·418	1·771
29	9·055	2·498	6·956	2·559	0·516	1·434	1·803
30	9·505	2·535	7·313	2·600	0·517	1·450	1·833
31	9·954	2·569	7·669	2·639	0·518	1·479	1·890
32	10·404	2·601	8·025	2·675	0·519	1·506	1·944
33	10.854	2·631	8·381	2·709	0·520	1·532	1·994

1	2	3	4	5	6	7	8
	Main Girders and Longitudinal Road Bearers.						Cross Girders.
Span.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum bending moment.	Total equivalent uniformly distributed load.	Maximum shear.		Maximum cross girder or pier reaction.
	At centre of span.		At ¼ point of span.		At centre of span.	At abutments.	
Ft.	Ft. Tons.	Tons.	Ft. Tons.	Tons.	Tons.	Tons.	Tons.
34	11·304	2·660	8·738	2·741	0·521	1·556	2·041
35	11·754	2·687	9·094	2·772	0·521	1·579	2·086
36	12·204	2·712	9·450	2·800	0·522	1·600	2·128
37	12·654	2·736	9·806	2·827	0·523	1·620	2·168
38	13·167	2·772	10·163	2·853	0·524	1·639	2·205
39	13·722	2·815	10·519	2·877	0·524	1·658	2·241
40	14·278	2·856	10·875	2·900	0·525	1·675	2·275
41	14·834	2·894	11·381	2·961	0·531	1·702	2·329
42	15·390	2·931	11·888	3·019	0·537	1·729	2·381
43	16·027	2·982	12·394	3·074	0·542	1·753	2·430
44	16·702	3·037	12·900	3·127	0·548	1·777	2·477
45	17·377	3·089	13·406	3·178	0·553	1·800	2·522
46	18·052	3·139	13·913	3·226	0·558	1·822	2·565
47	18·727	3·188	14·419	3·272	0·562	1·843	2·606
48	19·402	3·234	14·925	3·317	0·567	1·863	2·646
49	20·077	3·278	15·431	3·359	0·571	1·882	2·684
50	20·752	3·320	15·938	3·400	0·575	1·900	2·720
52	22·102	3·400	16·950	3·477	0·583	1·952	2·823

54	23·452	3·474	18·019	3·559	0·590	2·000	2·919
56	24·802	3·543	19·200	3·657	0·596	2·045	3·007
58	26·213	3·616	20·381	3·748	0·603	2·086	3·090
60	27·778	3·704	21·563	3·834	0·608	2·125	3·167
62	29·344	3·786	22·744	3·913	0·621	2·176	3·268
64	31·101	3·888	23·925	3·988	0·633	2·223	3·363
66	32·901	3·988	25·106	4·058	0·644	2·268	3·452
68	34·701	4·082	26·400	4·141	0·654	2·310	3·535
70	36·501	4·172	27·750	4·229	0·664	2·350	3·614
72	38·301	4·256	29·100	4·311	0·674	2·397	3·711
74	40·101	4·335	30·450	4·389	0·686	2·442	3·803
76	41·901	4·411	31·800	4·463	0·697	2·484	3·889
78	43·761	4·488	33·150	4·533	0·708	2·524	3·972
80	45·779	4·578	34·500	4·600	0·719	2·563	4·050
82	47·797	4·663	36·188	4·707	0·734	2·623	4·173
84	50·001	4·762	37·875	4·810	0·749	2·681	4·290
86	52·251	4·861	39·563	4·907	0·763	2·736	4·402
88	54·501	4·955	41·250	5·000	0·776	2·789	4·509
90	56·751	5·045	42·938	5·089	0·789	2·839	4·611
92	59·001	5·131	44·625	5·174	0·801	2·897	4·726
94	61·251	5·213	46·356	5·260	0·813	2·952	4·836
96	63·501	5·292	48·175	5·353	0·824	3·005	4·942
98	65·810	5·372	49·994	5·442	0·835	3·056	5·043
100	68·279	5·462	51·813	5·527	0·845	3·105	5·140
105	74·750	5·695	56·359	5·725	0·880	3·240	5·410
110	81·500	5·927	61·531	5·967	0·911	3·364	5·655
115	88·250	6·139	67·016	6·216	0·940	3·496	5·917
120	95·279	6·352	72·500	6·444	0·367	3·617	6·158
125	102·875	6·584	78·828	6·727	1·000	3·746	6·416
130	110·750	6·815	85·156	6·987	1·031	3·865	6·654
135	118·625	7·030	91·625	7·240	1·059	3·993	6·907
140	126·746	7·243	98·375	7·495	1·086	4·111	7·143
145	135·384	7·469	105·125	7·733	1·118	4·236	7·393

150	144·258	7·694	112·156	7·976	1·148	4·353	7·627
155	153·348	7·915	119·328	8·212	1·177	4·474	7·871
160	163·427	8·171	126·500	8·433	1·203	4·588	8·100
165	173·508	8·413	134·516	8·696	1·233	4·724	8·376
170	184·756	8·694	142·531	8·943	1·262	4·853	8·635
175	196·131	8·966	150·688	9·185	1·289	4·987	8·903
180	207·673	9·230	159·125	9·430	1·314	5·114	9·156
185	219·876	9·508	167·563	9·661	1·351	5·246	9·419
190	232·251	9·779	176·281	9·896	1·387	5·371	9·668
195	244·626	10·036	185·141	10·127	1·421	5·501	9·928
200	257·278	10·291	194·000	10·347	1·453	5·625	10·175
205	270·501	10·556	203·609	10·594	1·488	5·754	10·432
210	284·001	10·819	213·219	10·830	1·523	5·876	10·676
215	297·501	11·070	223·141	11·071	1·555	6·003	10·930
220	311·278	11·319	233·688	11·330	1·586	6·125	11·173
225	325·626	11·578	244·234	11·579	1·621	6·251	11·424
230	340·251	11·835	255·063	11·829	1·654	6·372	11·665
235	354·876	12·081	266·031	12·075	1·686	6·495	11·913
240	369·779	12·326	277·000	12·311	1·717	6·613	12·150
245	385·251	12·580	288·813	12·574	1·751	6·746	12·418
250	401·001	12·832	300·625	12·827	1·783	6·874	12·676
255	416·751	13·075	312·578	13·075	1·814	7·006	12·939
260	432·779	13·316	324·813	13·326	1·844	7·133	13·192
265	449·376	13·566	337·047	13·567	1·877	7·263	13·453
270	466·251	13·815	349·563	13·810	1·909	7·389	13·704
275	483·126	14·055	362·219	14·050	1·940	7·518	13·962
280	500·279	14·294	374·875	14·281	1·970	7·643	14·211
285	518·001	14·540	388·375	14·536	2·002	7·771	14·467
290	536·001	14·786	401·875	14·782	2·034	7·895	14·714
295	554·001	15·024	415·516	15·024	2·064	8·022	14·968
300	572·233	15·260	429·438	15·269	2·093	8·145	15·213

Impact must be added, see p. 501 (British Standard Specification—extract).

Ministry of Transport. Roads Department.

Standard Load for Highway Bridges.

The British Standards Institution's Loading (15 units) is accepted by the Ministry of Transport for the conditions to which it applies, but the Ministry of Transport's Standard Load, or the equivalent uniformly distributed load, tabulated below, is more commonly adopted.

The live load to be used consists of two items: (1) The uniformly distributed load which varies with the loaded length, and which represents the ordinary axle loads of the Ministry of Transport's standard train, perfectly distributed. (2) An invariable knife-edge load of 2,700 lb. per ft. of width applied at the section where it will, when combined with the uniformly distributed load, be most effective, *i.e.* in a freely supported span:—(*a*) for bending moment at midspan: at midspan point; (*b*) for shear at the support: at the support; (*c*) for shear at any section: at the section.

This knife-edge load represents the excess in the M.T. Standard train of the heavy axle over the other axles, this excess being undistributed (except laterally as already assumed).

In spans of less than 10 ft. (*i.e.* less than the axle spacing) the concentration serves to counteract the over-dispersion of the distributed load.

The uniformly distributed load, applicable to the 'loaded length' on the bridge or member in question, is selected from the table.

The 'loaded length' is the length of member loaded in order to produce the most severe stresses. In a freely supported span the 'loaded length' would thus be (*a*) for bending moment: the full span: (*b*) for shear at the support: the full span; (*c*) for shear at intermediate points: from this point to the farther support.

In arches and continuous spans, the 'loaded length' can be taken from the influence line curves.

In slabs the knife-edge load of 2,700 lb. per ft. of width is taken as acting parallel to the supporting members, irrespective of the direction in which the slab spans.

In longitudinal girders, stringers, etc., this concentrated loading is taken as acting transversely to them (*i.e.* parallel with their supports).

In transverse beams the concentrated loading is taken as acting in line with them (*i.e.* 2,700 lb. per ft. run of beam).

If longitudinal or transverse members are spaced more closely than at 5 ft. centres, the live load allocated to them shall be that calculated on a 5 ft. wide strip. With wider spacing this strip will be equal to the girder spacing.

In all cases, irrespective of span length, one knife-edge load of 2,700 lb. per ft. of width is taken as acting in conjunction with the uniform distributed load appropriate to the span or 'loaded length.'

Ministry of Transport. Standard Load for Highway Bridges.

Tabulated Values of Equivalent Distributed Loads.

Loaded Length.	Lbs. per Sq. Ft.	Loaded Length.	Lbs. per Sq. Ft.	Loaded Length.	Lbs. per Sq. Ft.
Ft. Ins.		Ft.		Ft.	
3 0	2,420	100	208	1,200	100
3 6	2,020	150	192	1,300	97
4 0	1,700	200	180	1,400	94
4 6	1,445	250	170	1,500	90
5 0	1,225	300	163	1,600	88
5 6	1,033	350	156	1,700	85
6 0	872	400	150	1,800	82
6 6	735	450	145	1,900	79
7 0	625	500	140	2,000	77
7 6	525	600	132	2,100	76
8 0	444	700	125	2,200	74
8 6	374	800	119	2,300	73
9 0	314	900	114	2,400	72
9 6	265	1,000	108	2,500	70
10 to 75 ft.	220	1,100	104	Over 2,500	70

These loads include the necessary allowance for impact.

Bridge Foundations.

LOADS ON BRIDGE FOUNDATIONS.

CYLINDER BRIDGE PIERS. APPROXIMATE SAFE LOADS PER SQUARE FOOT (*J. Newman*).

	Tons per Sq. Ft.
Firm sand in estuaries and bays	5·0 to 5·6
Considered safe on firm sand (in Holland)	6·0
Very firm, compact sand, foundations at depths not less than 20 ft.	6·7 to 7·8
Firm shale and clean gravel	6·7 to 8·9
Compact gravel	7·8 to 10·0

Calculated loadings (neglecting friction), tons per square foot: Benares bridge, 11·19; Jubilee bridge, 9; Gorai bridge, 8·5 (all deep foundations); Blackfriars bridge, 4·75: Rupnarayan bridge, 6·7 (or, assuming skin friction as 5 cwts. per square foot, the loading on the base of the pier is about 4·5 tons per square foot).

CLYDE BRIDGE, CALEDONIAN RAILWAY.

Caissons 48 to 89 ft. long and 20 to 23 ft. wide:

Land Piers.—Distances from river-bed about 50 and 60 ft.: depths below level of river-bed about 20 and 30 ft. Depths below quay level and loadings, 50 ft., 4·12 tons per sq. ft.; 61 ft., 4·73 tons; 57 ft., 4·75 tons.

River Piers.—Depths below river bed and loadings: 44 ft., 5·66 tons per sq. ft.; 44 ft., 5·96 tons; 48 ft., 6·04 tons; 44 ft., 6·12 tons.

SKIN FRICTION ON BRIDGE FOUNDATIONS.

(Caissons and 'Wells' or 'Cylinders.')

Foundations of the Empress Bridge over the Sutlej.

Brick Cylinders, 19 Ft. in diameter, sunk on an average 110 Ft.	Skin Friction. Cwts. per Sq. Ft. Mean.	Max.	Min.
Average of 36 cylinders sunk to depths varying from 10 to 27 ft., average 19 ft., under their own weight only, 25½ to 33 tons, average 31 tons.	0·85	1·33	0·63
Average of 100 observations in sinking 36 cylinders, at depths from 17 to 64 ft., loads 31 to 249 tons, average 132 tons besides the weight of the cylinders.	2·13	4·08	1·29
Average of 9 observations in sinking 3 cylinders by pneumatic process at depths varying from 44 ft. to 63 ft.	2·71	3·52	2·32

Papaghni Bridge.—Cast-iron cylinders 12 ft. in diameter and brick cylinders of the same diameter. The following frictional resistances were noted when sinking the piers; cwts. per sq. ft. In the upper sand 2·08 to 2·20; in black clay, 3·50 to 5·60; in silt below the clay, 2·72 to 4·28; in the lower sand, 2·58 to 3·16.

Chitravati Bridge.—Friction through 33 ft. of sand, 10 ft. of clay, and 7 ft. of clay, sand, and boulders—2·32 to 3·77 cwts. per sq. ft. Friction through 33 ft. of sand, 10 ft. of clay, and 3 ft. of sand and clay—2·93 to 3·62 cwts. per sq. ft.

Tavy and Laira Bridges, Devonshire.—2·0 to 2·8 cwts. in mud.

Concrete Cylinders at Haulbowline Dockyard (H. E. Oakley).—Twenty-three cylinders, 8 ft. in diameter, were sunk through silt. The average skin friction for a cylinder varied from 4·2 to 8·6, the mean being 5·9. In only three cases was the average for a cylinder less than 5·2 cwt per sq. ft. In seven cases the maximum resistance recorded exceeded 9 cwts. per sq. ft.

Caissons of Barnes Railway Bridge.—The caissons are 28 ft. by 18 ft. The skin friction in London clay was 2·03 cwts. for one, and 2·95 cwts. for the other, per sq. ft.

A Shaft sunk at Oxford Circus, London.—Skin friction only 1·0 cwt. per sq. ft. The low value being attributed to (1) the cylinder keeping its shape very well; (2) being kept very upright; and (3) a pocket being formed below the cutting edge and pugged clay inserted.

A Shaft sunk at Greenwich.—In sand and ballast, skin friction 4 to 4½ cwts.; in shelley clay, 6 cwts. per sq. ft.

Waterway of Bridges.

The simplest cases of waterway calculation are those in which the conditions are such that the amount of heading up is not restricted. In these cases the limiting velocities which may be allowed decide the area of waterway which is necessary. Above the bridge the mean velocity of the stream will be reduced by the heading up.

It may be assumed that the water surface will fall to the original water level before it passes the piers, unless the channel of the river below the bridge has been altered. Taking the water level as the same, the new depth will be the same as the original depth, *plus* any lowering of the bed of the stream at the bridge site. The new width may be taken as the length of waterway. These two dimensions give the new area from which the new velocity can be calculated directly from the discharge, or by

$$V_2 = V_1 \times \frac{d_1 l_1}{d_2 l_2}$$

where

d_1 = original depth; d_2 = new depth; l_1 = mean width of stream cross-section; l_2 = length of waterway.

The new mean velocity is that to which the piers of the bridge are subjected, as well as the bed of the stream for a short distance below the bridge. This distance depends on the velocity and depth of the current. The limiting velocities can best be decided by a study of records of the behaviour of local strata and local masonry under similar conditions.

The following figures provide a rough guide to the velocities usually allowed.

Limiting Velocities in Feet per Second.

On clay, 0·70 to 1·0; on sand which must not be disturbed (shallow foundations) 1·2 to 1·5; on firm, pebbly beds, 3·0 to 3·5; on stony or shingly beds, 4·0; on stratified rock, 6·0; on hard rock, 10·0.

For inferior brickwork, with better masonry in the cutwaters, and on the floor of the waterway at the feet of the cutwaters, about 5·0; better masonry, up to about 8 ft. per second. For ordinary pitching (floor) about 5·0. For very well-shaped bridge openings with thick piers allowing of easy curves, higher values may be allowed, and very high values may be permitted when the masonry is exceptionally strong, such as ashlar of hard stone with fine joints.

Waterway Calculations.

In the case of a bridge or culvert intended for the passage of flood waters which pool up on the upstream side, the discharge may usually be calculated without velocity of approach, especially when the water level on the downstream side is not much lower. This is an ordinary case where a railway embankment crosses land liable to flooding. The head is the difference between the water level below the opening and that above it, when the latter has reached its maximum. This may be a natural maximum, or one fixed by the discharge capacities of the bridges and culverts through the embankment. The amount of headway above this level, under the bridge floor, will depend upon what flotsam is carried by the water in floods.

The formula for discharge is :—

$$D = c\,A\,\sqrt{2gh},$$

where

D = discharge in cubic feet per second (cusecs); A = area of opening in square feet up to the lower water level; h = head, or difference of water levels (in ft.); g = 32·2 approx.; c is a coefficient depending partly upon the size of the opening, and partly upon its form, especially the curves of the cutwaters and wing walls.

Ordinary values for c are in many cases 0·90; for large and well-designed openings, 0·92 to 0·95; for smaller openings, 0·85 to 0·90.

In the case of culverts through an embankment, or for bridges, for the maximum discharge before the embankment is overtopped, it may be necessary to know the discharge when the opening is drowned. If it is drowned on both sides the discharge is :—

$$D = c\,A\,\sqrt{2gh},$$

where

h = difference of water levels, and c = about 0·75 to 0·85, according to the form and size of the opening.

Where the floor of the bridge presents a sharp edge a fairly low value of c should be taken and a fairly high value for well-shaped culverts.

In British measure, $\sqrt{2g} = 8 \cdot 02$; in metric measure, $\sqrt{2g} = 4 \cdot 419$.

When the upstream end is drowned and the downstream end is not drowned, the discharge is found by :—

$$D = cl \sqrt{2g} \left[\frac{2}{3}\left(h^{\frac{3}{2}} - h_2^{\frac{3}{2}}\right) + \left(h_1 - h\right)h^{\frac{1}{2}}\right],$$

where

l = the width of the opening ; h = the difference in water levels ; h_1 = the head above the bottom of the opening ; h_2 = the head above the top of the opening ; c may be taken as 0·75 to 0·85.

When the span is a long one and the waterway deep, a fairly high value may be given to c, but for a culvert the value may be taken as lower than that for the same culvert when wholly drowned.

In a considerable number of cases the waterway of the bridge is nearly as great as that of the stream, or quite as great when a single span clears the flood channel. Even with a considerable number of spans the bridge may obstruct the waterway very little if the piers are thin and are provided with good cutwaters and joinwaters. The velocity of approach may then be taken as v, the mean velocity of the stream, and the following relation will serve for preliminary calculations :

$$D = c\,A \sqrt{2g\,(h + h_a)}\,;$$

where

$ha = \frac{v^2}{2g}$; $A = dl$, l being the waterway of the bridge, and d the depth to the floor under the bridge.

For sluggish streams h may be neglected, and for a stream with a sharp fall it may be taken as the fall in a length measured about ¾ that of the piers. The equation, so used, may be useful in preliminary calculations ; where, for instance, the mean velocity of the stream is measured, but not its cross-sectional area, or where the discharge has been separately found. In the case of a sluggish stream it may be possible to fix the value of d, approximately.

Calculations involving Afflux.

Where the actual fall is small, as in most large rivers, and where the bridge considerably impedes the river, the total head driving the water through the opening is usually calculated as : $(x + h_a)$, where x = the afflux and h_a = head equivalent to the velocity of approach. The discharge is then :

$$D = c\,A \sqrt{2g\,(x + h_a)},$$

where

A = the smallest cross-sectional area of the waterway under the bridge, usually taken as that given by a water surface at the original water level.

$$x = \frac{v^2}{2g}\left(\frac{l^2}{c^2 l_1^2} - \frac{d^2}{(d + x)^2}\right),$$

where

l = the mean width of the stream ; d = mean depth of the stream below the bridge ; l_1 = the total width of waterway in the bridge openings ; c = coefficient of contraction, say 0·9 to 0·95 for large bridge openings and about 0·85 for small openings with only moderately good cutwaters and wing walls.

First insert the following provisional value of x in the right-hand side of the above equation.

$$x \text{ (provisional)} = \frac{v^2}{2g}\left(\frac{1{\cdot}1\, l^2}{l_1^2} - 1\right).$$

This gives a new value of x.

Repeat the solution of the longer equation with the new value of x on the right-hand side.

Next, to find h_a.

The head is calculated as that at one place, due to afflux and velocity of approach, and the velocity of approach is, therefore, the new reduced velocity of the deepened stream above the bridges, thus :

$$va = \frac{d}{d + x}\,v, \text{ and } h_a = \frac{v_a^2}{2g}\,;$$

we can then use the equation:

$$D = c A \sqrt{2 g (x + h_a)}.$$

If the length of the waterway is only estimated at first, within limits, take a probable value, and having found x, find l_1 by the equation:

$$l_1 = c \frac{vl (d + x)}{\sqrt{[2gx (d + x)^2 + v^2 d^2]}}.$$

In some cases x will be decided by the maximum water level which may be allowed, and the value so found will be used in the equations.

Discharge through Long Culverts.

The foregoing formulæ for discharge are suitable for short culverts, and for fairly long culverts when the velocities are low. For long culverts and for high velocities the head necessary to carry the water through the barrel regarded as a pipe must be added. The heads may be found by a pipe discharge formula, or, for small diameter culverts, may be taken from the tables (see 'Water Supply,' Section XVIII., Part VI.).

(Reginald Ryves, M.Cons.E.)

Weights of Bridgework.

Dead Loads.—The following are weights of various types of bridge floors, etc.:—

Jack arch flooring	140 lbs. per cu. ft.
Steel trough flooring	13–56 per sq. ft.
Roadway setts in cement	120 ,,
Timber sleepers	125 lbs. each.
Ballast (average depth 12 ins.)	100 lbs. per sq. ft. of floor.
Sleepers and ballast	100 lbs. per cu. ft.
5-lb. rails, chairs and all fastenings	$1\frac{1}{4}$ cwt. per lin. ft. of single track.
Jarrah wood-block paving $2\frac{1}{2}$ ins. thick, including mastic, pegs, and screws	$13\frac{1}{2}$ lbs. per sq. ft.
Tar macadam	120 lbs. per cu. ft.
Concrete filling over buckled plates	140 lbs. per cu. ft.
Fish plates	40 lbs. per pair.
,, bolts	$5\frac{1}{2}$ lbs. per set (4).
Oak keys	$2\frac{1}{4}$ lbs. per pair.
Chairs	45 lbs. each.

Approximate Weights of Girders, etc.

Unwin's Formula.

W_1 = weight of girder per ft. run, exclusive of cross-girders or flooring

$$= \frac{Wr}{cs - lr},$$

where

W = load to be carried in tons; l = span in ft.; r = ratio of span to depth; s = working stress in tons, sq. ins.; c = a constant, which may be taken from 1,200 to 1,400 for small plate girders and 1,700 to 1,900 for truss bridges.

Johnson, Bryan & Turneaure Formula.

Deck plate girders	$w = 9l + 120$
,, lattice girders	$w = 7l + 200$
Through pin bridge	$w = 5l + 350$

(These figures are for single tracks)

where

l = span in ft.; w = weight of each girder in lbs. per ft. run.

American Formulæ.

(1)

The following empirical formulæ, although roughly approximate, are useful as guides in assuming dead loads of ordinary truss road bridges (exclusive of flooring, cross-girders, and joints:—

For a capacity of 100 lbs. per sq. ft.,

$$W = s + 5b + \frac{(b-6)}{8} + \frac{l^2}{300} \quad . \quad . \quad . \quad . \quad . \quad (1).$$

For a capacity of 80 lbs. per sq. ft.

$$N = s + 5b + \frac{bl}{16} + \frac{l^2}{300} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2).$$

For any other capacity of live load s_1, find W by formula (1), remembering $s = 100$; call required dead load W_1, then $W_1 = W - \left[1 + \frac{(b-12)l}{320}\right](100 - s_1)$,

where

W = dead load per lin. ft. exclusive of floor and joists (lbs.); s = live load per sq. ft. (lbs.); b = clear width of roadway (ft.); l = length of span (ft.).

(2)

For weight of small plate and angle girders,

$$P = \frac{Wl}{20d}$$

where

P = weight of beam (lbs.); W = total live load supported (lbs.); l = distance in feet between supports; d = total depth of beam in inches.

Skeleton Weights.

These formulæ are useful for obtaining the first approximation to the weight of main girders or trusses, but a more reliable approximation is obtained by calculating the skeleton weight and adding a percentage for details.

For trusses or systems of lateral bracing $P = \frac{3 \cdot 4k}{2240} \Sigma lA$;

where P is the total weight of the truss in tons, l is the length in feet of each member between intersection points, A the corresponding cross sectional area in square inches, and 3·4 lbs. the weight of a 1 inch square bar, one foot in length. Average values for the constant k are as follows:—

If A = area provided, including lacing bars, $k = 1 \cdot 2$.
If A = area provided, excluding lacing bars, $k = 1 \cdot 3$.
If A = area required, excluding lacing bars, $k = 1 \cdot 4$.

The method of skeleton weights is also extremely useful for determining economic outlines for trusses with regard to type, depth, number of bays, etc.

Working Stresses.

The following figures are those of a leading structural engineering company, in which a reduced working stress is used.

Permissible Maximum Stresses.

(A) *Railway Bridges.*

The working stresses shown below have been proportioned to allow for dynamic action of the live load on lightly loaded girders or members of girders.

(B) *Road Bridges.*

To provide for the effects of the dynamic action of the live load on lightly loaded girders or members of girders, the live load stresses to be increased by 25 per cent. in the case of main girders and 33⅓ per cent. in the case of floor girders.

(C) *Railway and Road Bridges.*

(1) The combined stresses resulting from the rolling load, dead load, wind, momentum, and centrifugal force, shall not produce a greater tensile stress than half of the elastic limit, or equal to $\frac{3}{10}$ of the minimum ultimate tensile strength of the material, nor more than the corresponding compressive shearing, bearing, and bending stresses as hereinafter set forth, but

(2) The combined stresses resulting from the rolling load and dead load alone, exclusive

of wind, momentum, and centrifugal force, shall not produce greater tensile stresses than those given below.

TENSILE STRESSES.

Railway Bridges.

For main girders, cross-girders, and rail bearers of plate construction:

Under 20ft. span	4½ tons per sq. in.
20 ft. to 25 ft.	4¾ ,, ,,
25 ft. to 30 ft.	5 ,, ,,
30 ft. to 50 ft.	5¼ ,, ,,
50 ft. to 80 ft.	5½ ,, ,,

For trusses and lattice girders:

80 ft. and under 160 ft. span—	
Bottom chords	5½ tons per sq. in.
Diagonals	4½ to 5½ tons per sq. in.
160 ft. and under 200 ft. span—	
Bottom chords	5¾ tons per sq. in.
Diagonals	4½ to 5¾ tons per sq. in.
200 ft. to 400 ft. span—	
Bottom chords	6 to 7 tons per sq. in.
Diagonals	4½ to 7 tons per sq. in.
All spans—for wind bracing	8½ tons per sq. in.
Floor suspenders	2½ ,, ,,

Road Bridges.

For main girders, cross-girders, and stringers:

Bottom chords	7 tons per sq. in.
Diagonals	5–7 tons per. sq. in.
Wind bracing	8½ ,, ,,
Floor suspenders	3½ ,, ,,

Note.—The 4½ tons stress on diagonals of railway bridges, and the 5 tons stress on road bridges, will apply to those at the centre portion of the span and to the counterbracing at the same point. The higher stresses will apply at the end portions where variations of stress are not so great. Intermediate diagonals will be subject to stresses lying between these limits.

COMPRESSION STRESSES.

Railway and Road Bridges.

For plate girders, the gross area of the compression flange shall not be less than that of the tensile flange, nor shall the compression stresses per sq. in. be more than 85 per cent. of the corresponding tensile stresses.

For truss and lattice girders the compression stress per sq. in. shall, in the case of riveted members, not exceed the fraction '$0{\cdot}95-0{\cdot}003R$' of the corresponding specified tensile stress nor in the case of pin joints the fraction '$0{\cdot}95-0{\cdot}0045R$,' where $R =$ the ratio of the length of the unbraced portion of the member to its least radius of gyration; nor in any case shall it exceed 85 per cent. of the said tensile stress.

No compression member in railway bridges shall have a greater length than 100 times its least radius of gyration, or 45 times its least width, except for wind bracing, which may have a length not exceeding 120 times its least radius of gyration.

In the case of road bridges no compression member shall have a length exceeding 120 times its least radius of gyration, except wind bracing, which may have a length not exceeding 140 times its least radius of gyration.

ALTERNATING STRESSES.

Railway and Road Bridges.

Members subject to alternate tension and compression stresses shall be proportioned as struts to resist the greater stress plus half the lesser, except for wind bracing, which shall be proportioned for the greater stress only. The sum of the stresses to be used for connections.

PERMISSIBLE UNIT STRESSES IN SWING BRIDGES.

The unit working stresses in main girders shall be 10 per cent. less than those for simply supported spans. *(Sir W. Arrol & Co., Ltd.)*

PERMISSIBLE UNIT STRESSES, IN POUNDS PER SQUARE INCH.

Kind of Strain, and Loading.		Structural Steel.	Axle Steel.	Steel Castings.	Cast Iron.	Rolled Copper.	Brass.
Tension	(A)	16,000	18,000	12,000	3,000	6,000	3,000
	(B)	10,600	12,000	8,000	2,000	4,000	2,000
	(C)	5,300	6,000	4,000	1,000	2,000	1,000
Compression	(A)	16,000	18,000	16,000	12,000	6,000	3,000
	(B)	10,000	12,000	10,000	8,000	4,000	2,000
Bending	(A)	16,000	18,000	15,000	6,000	—	—
	(B)	10,600	12,000	10,000	4,000	—	—
	(C)	5,300	6,000	5,000	3,000	—	—
Shear	(A)	12,000	14,000	9,000	3,000	3,600	—
	(B)	8,000	9,600	6,000	2,000	2,400	—
	(C)	4,000	4,800	3,000	1,000	1,200	—
Torsion	(A)	10,000	12,000	7,500	3,000	—	—
	(B)	6,600	8,000	5,000	2,000	—	—
	(C)	3,300	4,000	2,500	1,000	—	—

A. For a static load; *B*. For a varying load producing stresses of tension or compression only; *C*. For a varying load producing equal maximum strains in opposite directions, accompanied by shocks and vibrations.

The permissible unit stresses for bending (given opposite *B*)—10,000 for cast steel and 4,000 for cast iron—should be used for determining the strength of toothed gearing. The teeth in cut gearing should conform to the following proportions :—

$$P = s\,p\,f\,y.$$

where,

P = pressure on tooth in pounds ; s = permissible unit stress ; p = pitch, in inches ; f = face of tooth, in inches ; y = a factor.

For wheels moving at slow speed (up to about 100 ft. per minute), where strength only is to be considered, y = 0·05.

The strength of uncut teeth should be computed for a face of one and one-half times the pitch, or

$$P = \frac{3\,s\,p^2\,y}{2};$$

which conforms approximately with the assumption that the pressure is carried on one corner of the tooth.

For higher velocities, which tend to increase the shock and wear, this value should not exceed :—

$$y = 0{\cdot}05\ \frac{10}{\sqrt{v}} = \frac{0{\cdot}5}{\sqrt{v}}$$

where

v = velocity in pitch circle, in feet per minute.

For bronze wheels, the same values should be used as for cast iron.

FIXED BEARINGS.

For the permissible unit pressure for fixed bearings of different metals, those for compression, in the table, should be used.

Working Stresses in Structural Steel.

(BRITISH STANDARD SPECIFICATION FOR GIRDER BRIDGES.)

Except as hereinbefore modified, structures shall be so designed that the calculated working stresses in structural steel of the 'A' quality specified in Clause 1, Part 1, of this Specification shall not exceed the following :—

For Parts in Tension.

On the net section for axial stress,

9 tons (20,160 lbs.) per sq. in. (14·17 kg. per mm.²).

For Parts in Compression.

On the gross section of the compression flanges of plate girders and I beams with outside edges stiffened with angles or channels,

$$9\left(1-0\cdot0075\,\frac{l}{b}\right) \text{ tons per sq. in.}$$

Ditto, with unstiffened edges,

$$9\left(1-0\cdot01\,\frac{l}{b}\right) \text{ tons per sq. in.}$$

where,

l = the greatest unsupported length as defined in Clause 11, Part 4, of this Specification, and

b = the breadth of the flange,

provided that the gross area of the compression flange shall be not less than the gross area of the tension flange.

On the gross section of compression members of truss and lattice girders with riveted connections for axial stress,

$$9\left(1-0\cdot0038\,\frac{l}{r}\right) \text{ tons per sq. in.}$$

On the gross section of compression members of truss and lattice girders with pin connections for axial stress,

$$9\left(1-0\cdot0054\,\frac{l}{r}\right) \text{ tons per sq. in.}$$

Where

l = the greatest length of the unbraced portion of the member

and

r = the least radius of gyration,

provided that, in truss and lattice girders, the working compressive stress on the gross section shall in no case exceed 7·65 tons (17,136 lbs.) per sq. in. (12·05 kg. per mm.^{2}).

For Parts in Shear.

On the gross section of web plates,

5·5 tons (12,320 lbs.) per sq. in. (8·66 kg. per mm.^{2}).

On shop rivets, turned tight-fitting bolts and pins,

6·5 tons (14,560 lbs.) per sq. in. (10·24 kg. per mm.^{2}).

For Bearing Areas.

On shop rivets, turned tight-fitting bolts and pins,

15 tons (33,600 lbs.) per sq. in. (23·62 kg. per mm.^{2}).

NOTE.—For connections which are to be made in the field or where black bolts are used instead of rivets, an excess of 15 per cent. in the case of field rivets and 20 per cent. in the case of black bolts over the number required according to the above working stresses, both for shear and bearing, shall be provided.

For Pins subjected to Bending.

On the extreme outer fibres,

13·5 tons (30,240 lbs.) per sq. in. (21·26 kg. per mm.^{2}).

Where structural steel of the milder quality specified in Clause 1, Part 1, of this Specification for material to be pressed cold is used, the foregoing working stresses where applicable shall be reduced by 10 per cent.

FOR PLATE GIRDERS AND BEAMS EMBEDDED IN CONCRETE.

On the tension and compression flanges of plate girders and I beams, the compression flanges and webs of which are solidly embedded in a continuous concrete matrix or jack arches, and with an effective cover of not less than 2 ins. concrete throughout over upper surfaces of the top flange, and where no account has been taken of the strength of the embedded matrix :—

10 tons (22,400 lb.) per sq. in. (15·75 kg. per mm.^{2}).

HIGH TENSILE STRUCTURAL STEEL.

In May 1934 the British Standards Institution published the British Standard Specification No. 548, for High Tensile Structural Steel for Bridges and General Building Construction, relating to steel having a range of tensile breaking strength from 37 to 43 tons per sq. in., and a yield point varying from 19 to 23 tons per sq. in., according to the thickness of the plate or section.

The working stresses usually allowed in steel of this quality are about 40 per cent. higher than the values specified for mild steel (structural steel, A quality) on p. 495.

High-tensile steel has been increasingly used of late years, especially in the construction of long-span bridges.

FIRST-CLASS BRITISH PRACTICE IN BRIDGEWORK.

Brief Specification.

1. Materials to conform to the British Standard Specifications.

2. All steel to be by open-hearth acid or basic process.

3. No material to be used until tested and passed.

4. Steel castings to be Grade 'C.'

5. Bolts and nuts, coach screws, clips, etc., to be Grade 'C' wrought iron.

6. All oak to be English or Scotch.

7. Edges of all flange, end, and web plates and ends of all bars to be planed, or where planing not possible, to be dressed fair and true by hand.

8. Wherever shearing is adopted a $\frac{1}{8}$ in. strip to be left for planing, plates and bars showing less than $\frac{1}{8}$ in. rejected.

9. All plates, bars, angles, tees, etc., carefully levelled, straightened, joggled, or upset by pressure and not by hammering, both before and after they are drilled.

10. Templates and gauges to be made to correspond exactly with each other in order that all holes may be straight through and through. Steel gauge $\frac{1}{16}$ in. less in diameter than hole shall pass through easily, irrespective of thickness of plates or bars.

11. Holes for turned bolts to be a driving fit.

12. All holes to be drilled and burrs carefully removed. Holes for rivets to be $\frac{1}{16}$ in. larger than nominal size of rivet.

13. Wherever practicable, all riveting to be done by machine. All loose rivets and such as may be cracked or have burnt or badly formed heads or with heads eccentric with the shaft or where the shaft does not fully fill the hole, cut out and replaced by others.

14. No edge of rivet or bolt hole to be nearer edge of plate or bar, etc., than a distance equal to one and a half times the diameter of rivet.

15. All work riveted or bolted together to be absolutely in contact over whole surface. Immediately before being assembled together whole of surfaces to be thoroughly cleaned and painted with a thick coat of genuine red lead paint.

16. All bolts to be forged and not welded. Screwed to Whitworth's thread. Those taking a shear stress to be so screwed that no part of the thread is in contact with bearing surface.

17. The barrel of all turned bolts to be $\frac{1}{16}$ in. larger in diameter than the screwed portion. Machined washers to be provided.

18. All bolts to be dipped in hot boiled linseed oil.

19. Cover plates connecting different thicknesses of plates to be planed off, the step to be sloped off.

20. All work to be temporarily erected and put together with service bolts.

21. All steelwork to be scraped and cleaned of all rust, mill-scale, and dirt and brushed with wire brushes.

22. All steelwork not to be galvanised to be given two coats of Dixon's Silica Graphite Paint No. 2, or natural colour, and a further two coats after erection, the last a finishing coat of approved colour. No oils or thinners to be used.

Girder Bridge Specification.

A specification issued by the British Standards Institution (No. 153—1933) deals with railway and highway bridges. The first part is concerned with materials of construction (structural and rivet steel, cast steel, steel for pins and rollers, wrought iron and cast iron); these are required to be in accordance with existing B.S. specifications, and the latter are printed as appendices, making the present document complete in itself. The second part gives requirements as regards workmanship, and includes clauses on straightening, planing and shearing, drilling, punching and reaming, rivets and riveting, smithed work, bolts and nuts, temporary erection at contractors' works, painting, measurement, and packing and marking for export. Parts 3, 4 and 5 (revised in 1937) deal with loads and stresses, details of construction, and erection.

WIND PRESSURE.

During a period of fourteen years, from 1890 to 1904, observations were made at fourteen different stations in the United Kingdom, and it was found that the maximum mean wind velocity only reached the following values:—

77–80	miles	per hour	on one occasion.
73–77	"	"	"
70–73	"	"	five occasions.
65–69	"	"	"

The records of many stations in the British Isles from 1906 to 1918 show that the maximum gusts recorded were :—

99 miles per hour, once.
90 ,, ,, twice.
85–90 ,, ,, nine times.

Velocities very much in excess of these values have been officially recorded by the Robinson anemometer, the velocity of the cups being multiplied by 3, to give the actual velocity of the wind. This figure is now recognised to be too high, and a value of 2·2 is substituted as a multiplier.

The velocity of wind, and therefore the pressure, increases with the height above ground.

The following observations are due to Mr. Adam Hunter, M.I.C.E., and were taken on the Forth Bridge by means of small gauges 1·5 sq. ft. in area placed at different heights.

Year.	Date.	Pressure in Pounds per sq. ft. at Various Heights.				
		50 ft.	163 ft.	214 ft.	214 ft.	378 ft.
1901	Jan. 26	—	15	25	—	65
,,	Nov. 23	—	50	55	55	60
1902	Dec. 13	—	27·5	31	34	18
1903	Jan. 10	15	20	25	27·5	60
,,	,, 31	—	19·5	29	26	65
,,	Mar. 18	20	20	25	29	31
,,	,, 21	10	20	20	22·5	54
1904	,, 26	—	20	32	27	52
,,	Dec. 29	—	22·5	22·5	32·5	—
1905	Jan. 21	—	21	30	23	—
,,	Mar. 18	—	32·5	32·5	42	60
,,	Feb. 28	10	22	20	20	38
1906	Jan. 26	15	—	—	—	59
,,	,, 11	10	20	23·5	25	30
,,	Feb. 8	10	15	25	25	55
Average . . .		13·0	23·0	28 0	30·0	50·0

The two readings at 214 ft. are at opposite ends of the bridge.

Relation between Velocity and Pressure.

(See Sec. XXXII, Part II.)

Notes on Wind Pressure on Bridges.

There is considerable shielding effect of the leeward members of a framed structure by those on the windward side, dependent mainly upon their distance apart.

The determination of effective wind pressure on bridges presents many difficulties, arising chiefly out of the irregular shape and construction of those other than plate girder spans.

Plate girders are not, strictly speaking, flat plates, and up to 10 per cent. increase in area is often allowed for the effect of their cup-shape form.

The following remarks are taken from the British Standard Specification :—

'Where wind pressure has to be taken into consideration, it shall be treated as a moving load not subject to any impact effect and shall be assumed to act horizontally at a slight angle to the transverse axis of the structure so as to take effect on the exposed area of the flooring and the leeward parts in the case of openwork structures, except where any portion may be temporarily screened by a moving load. Where the leeward girder is at a distance from the windward girder not exceeding twice its depth, the effective area of the former shall be taken at half the exposed surface, but where the distance apart exceeds twice the depth, the whole exposed surface shall be taken into account. When the structure is unoccupied by a moving load, the maximum pressure shall be assumed to be 50 lbs. per sq. ft. (244·12 kg. per m.^{2}), but when there is a moving load thereon a pressure of 30 lbs. per sq. ft. (146·47 kg. per m.^{2}) in the case of railway bridges, and 20 lbs. per sq. ft. (97·65 kg. per m.^{2}) in the case of road bridges, shall be assumed as acting on the exposed surfaces both of the structure and the moving load, the pressure on the latter being taken as acting through the centre of gravity of the exposed area. The maximum results from the wind blowing in either direction, and with the structure loaded or unloaded, shall be taken. Where, owing to the position of the bridge or local conditions, these pressures cannot be reached or may be exceeded, the engineer may, at his discretion, alter any of them, as required.'

In providing the necessary anchorage for the bridge a train of empty passenger carriages to be assumed to be on the bridge, weighing 10 cwt. per lin. ft., and in the case of a double track the leeward track only to be loaded.

It is customary to specify a reduced wind pressure for a loaded span for the following reasons :—

(*a*) Because the total area exposed to the wind is increased, and hence the allowance necessary for the rise in pressure due to gusts above that corresponding with the maximum mean wind velocity is not so great as is necessary with a smaller area.

(*b*) Because the resistance offered by a body like a train is not so high as that offered by a surface like a bridge.

During a storm of great intensity the speed of trains crossing bridges will probably be reduced, owing to a feeling of insecurity on the part of drivers, but it cannot be assumed that the impact of trains, except for spans less than 150 ft., will be reduced ; for this depends more upon critical speeds than upon their maximum velocities.

It is therefore necessary, in such cases, where a reduced wind pressure is assumed for calculation, to carefully consider the effects of wind in all its aspects, treating it particularly as a moving load on the train.

The shielding effect of double bars will be to a large extent eliminated if the wind is treated as acting at a slight angle as mentioned.

In bridges of large dimensions it is necessary to make a somewhat different and more exact computation of wind pressures, and where the spaces are large, separate calculations for each individual member based upon their size, shape, and form, may be essential.

Centrifugal Force.

South American Railway Practice.

Centrifugal force on bridges on curves taken at 0·02 of the live load for each degree of curvature up to 5°, and reduced by 0·001 for every degree of curvature above 5°—the force considered as acting at a height of 5 ft. above rail level. In short spans where impact is high, a percentage for impact is added.

British Practice.

Centrifugal force for each degree of curvature, assumed to be 1 per cent. of the maximum rolling load on all tracks for a speed of 30 miles and under, and 1 per cent. added for each increase in speed of 10 miles per hour. Considered as acting at 5 ft. above rail level.

The radius in feet (R) is reduced to degrees of curvature (D) by means of the following formula:—

$$D \times \frac{5730}{R}.$$

Theoretically, centrifugal force varies *directly* as the weight of the train, and as the square of the velocity at which it travels, and *inversely* as the radius of the curve, or

$$P = \frac{W}{g} \times \frac{V^2}{r},$$

where

W = weight of train in tons ; V = its velocity in ft. per sec. ; r = radius of curvature of rails in ft. ; g = 32, the force of gravity ; P = centrifugal force in tons.

The whole of the centrifugal force is not usually neutralised by the superelevation of the rails, hence the allowances are made as above.

British Standard Specification for Centrifugal Force.

Where a structure carrying a railway is situated on a curve, provision shall be made in designing the members for the stresses due to the centrifugal action of the moving loads, each track on the structure being considered as occupied. The allowance for the centrifugal effect shall be calculated from the following formula :—

$$C = \frac{WV^2}{15\ R}$$

where,

C = the centrifugal effect per lineal foot considered as a moving load, acting at a height of 6 ft. (1·83 m.) above the level of the rails, unless otherwise specified by the engineer ;

W = the equivalent distributed live load per lineal foot;

V = the allowable maximum speed of the train in miles per hour, as specified by the engineer; and

R = the radius of the curve in feet.

No increase for impact effect shall be made on the stresses due to the centrifugal action.

Momentum of Trains.

Tractive Force and Braking Stresses.

In bridges of large dimensions special attention should be given to the details of the structure, to provide for the longitudinal stresses resulting from sudden application of continuous brakes to the train while on the bridge. The horizontal force resulting from such action can be taken as one-fifth of the weight of the train.

The adhesion weight of locomotives is usually about 450 lbs. per ton of load on the coupled axles.

(Refer also to Power, Tractive Force, and Adhesion of Locomotives, Train Acceleration, etc., Section XXXII., Part II., 'Railways.')

Longitudinal Forces.

Where a structure carries a railway, provision shall be made for the stresses due to the tractive effort of the live load and the braking effect resulting from the application of the brakes to such load while passing thereover, these forces being considered as acting at rail level.

(*a*) For railways worked by steam or electric locomotives, the amount of the tractive effort on one track shall be ascertained by multiplying one and three-quarter times the maximum end shear due to the live load on that track by a factor equal to :—

$$\frac{20}{L + 75}$$

where,

L = the span in feet. The load on the track shall be calculated from the shear table for end shear as set out in Appendix I., or such other loading as may be specified by the engineer. The factor shall not exceed 0·15 as a maximum.

The braking effect shall be similarly determined by using a factor equal to :—

$$\frac{12}{L + 90} + 0{\cdot}075$$

also limited to a maximum of 0·15.

(*b*) In the case of lines worked solely on the electrical multiple unit system, the amount of the tractive effort on one track shall be ascertained by multiplying the sum of the actual wheel loads on the span by a factor equal to :—

$$\frac{3}{L - 10} + 0{\cdot}10$$

where,

L = the span in feet. The factor shall not exceed 0·20 as a maximum.

The braking effect shall be similarly determined by using a factor of 0·20 for all spans.

(*c*) Where the structure carries more than one track, all the tracks shall be considered as being occupied simultaneously by the live loads, tractive and braking forces being applied to alternate tracks. The maximum effect on any girder with two tracks so occupied shall be allowed for, but where there are more than two tracks, a suitable reduction may be made on these forces for the additional tracks at the discretion of the engineer.

No increase for impact effect shall be made on the stresses due to the longitudinal forces.

(*B.S.I.*)

Impact.

If a load is applied suddenly to a structure, vibration will ensue and the strain—and thus the stress—will reach twice the value which would occur if the load were gradually applied. There is a great similarity between the results of experiments on variation of stress and sudden loading. The French Government obtain working stresses by means of the Launhardt-Weyrauch formula, but in America and this country various impact formulæ have been devised, but which are by no means final.

Theory prescribes that the following circumstances at least should be taken into account in any impact formula:—

(1) The length of the bridge;
(2) The ratio of live to dead load;
(3) The natural frequency of the bridge;
(4) The fluctuation in rail pressure due to balance weights, hammering, etc.;
(5) The size of the driving wheels.

To supply the necessary information required under (3), a systematic research is required dealing with bridges of various sizes and types.

Waddell's Formula.

$$I = \frac{400}{L + 500},$$

where

L = length of that portion of the span which is covered by the live load when the maximum stress under consideration is produced, and I is the percentage by which the maximum static stress is to be increased.

Allowance for Impact in Bridges.

S = span in ft.; V = velocity in ft. per sec.; T = time in secs. required by a train travelling at velocity V to travel from the point where a given member commences to receive the stress to the point where the stress is a maximum (i.e. in a plain girder, T is the time taken to pass half the span); D = maximum allowable deflection; t = time taken for a free falling body to fall through a height D, then

$$t^2 = \frac{D}{16}, \text{ or } t = \tfrac{1}{4}\sqrt{D}$$

If

P = percentage allowance for impact, then

$$P = \frac{50V\sqrt{D}}{S}.$$

As the formula is correct for a single concentrated load only, it gives excessive allowances for long spans, and P may be reduced $\frac{1}{30}$th of its found value for each 25 ft. span in spans of over 25 ft.

In all cases where V is such that the load traverses the half span in less time than t, the percentage for impact must be 100, the maximum V being fixed at 100 ft. per sec.

For a deflection of $\frac{1}{2500}$ of the span, $P = \dfrac{V}{\sqrt{S}}$, less the allowance for spans above 25 ft.

These formulæ are based on the idea that impact must vary with the time of application of the load. On a long bridge even a high speed train only reaches its position of maximum bending moment after several seconds, so that the bridge deflects slowly to its maximum deflection and goes very little beyond what the train at rest would produce. This applies to main members. Floor beams and their attachment, however, receive maximum stress very rapidly, for the train advances to full effect in the brief space of one panel length. On floor stringer end-connections the first end is loaded absolutely instantaneously; the stringer itself takes its maximum load in about half a panel length, and so on; each member is to be allowed for impact according to the time required from its first feeling the load to the time of maximum moment. No simple load can produce an impactive stress more than double its static stress, but a swaying locomotive may have more than half its load on the wheels of one side, so that a wheel which may cause a maximum load on, say, the first end of a floor stringer, may happen to be loaded at the moment perhaps 30 per cent. more than its half share of the load on its axle, and at the same time it may happen that the balance weight is just delivering its downward hammer blow. Thus care is needed to give ample strength to details even of larger bridges, but the main members of a large bridge are but little affected by the trivial details of locomotive swaying, hammer blow, &c.

British Standard Specification for Impact Effect on Road Bridges.

In the case of road bridges the addition to the live load, which shall cover all impact effects due to the vehicle and its load or to irregularity of surface, shall be ascertained by multiplying the live load by a factor derived from the following formulæ:—

(1) For bridges carrying one line of traffic only:—

$$I = 0 \cdot 75 - 0 \cdot 002\,l \text{ with a maximum value of } 0 \cdot 60.$$

(2) For bridges carrying two or three lines of traffic:—

$$I = 0 \cdot 65 - 0 \cdot 002\,l \text{ with a maximum value of } 0 \cdot 50,$$

where I = the factor and l = the effective span in feet.

For a main girder the effective span shall be the distance between the centres of bearing plates or rocker pins.

For a cross girder, other than an end cross girder or intermediate cross girder at which the continuity of the flooring is interrupted, l shall be the distance between centres of the cross girders on either side thereof. In the case of an end cross girder or intermediate cross girder at which the flooring is interrupted l shall be the distance between the centres of the cross girder and the adjacent cross girder, or where the flooring is continued beyond the cross girder the distance between the centre of the adjacent cross girder on the one side and the termination of the floor members on the other, except that where the floor members terminate on an abutment or pier the centre of the bearing of the floor members thereon shall be taken as the limit of l on that side.

For a longitudinal bearer the effective span shall be the distance between the centres of the adjacent cross girders supporting the bearer or the centre of the cross girder supporting the bearer at one end and the centre of its bearing on the abutment or pier at the other.

When a member supports or assists in supporting more than three lines of traffic the engineer shall specify any required modification of formula (2).

In the case of all end cross girders on which the flooring is terminated and for bridges in which the deck does not form a rigid combination with the bridge structure, as in the case of timber decking bolted thereto, no maximum limit shall be imposed on the value of I.

The allowance for impact on road bridges is based upon the assumption that the carriage-way on and immediately adjacent to the bridge is properly constructed and maintained in good surface.

No addition for impact shall be made to the live load due to pedestrian or equivalent light traffic.

Bridge Stress Committee's Recommendations for Impact Effect.

The Bridge Stress Committee found all existing locomotives in this country to be covered by the British Standard Unit Loading multiplied by 18 units, and that three of the four main railways designed bridges for main lines and principal branches for 18 units.

The Committee consider that the effects of existing locomotives are covered by the following combinations :—

18 units of load with a total hammer blow of	5 tons at	5 revs. per sec.
16 ,, ,, ,, ,,	12·5 ,,	5 ,,
15 ,, ,, ,, ,,	15 ,,	5 ,,

For the design of new bridges the Committee recommends the following loadings which may be taken to cover any locomotives which are likely to be designed in the future :—

Loading A, 20 units of load with a total hammer blow of	$0 \cdot 2\ n^2$ tons
Loading B, 16 ,, ,, ,, ,,	$0 \cdot 5\ n^2$ tons
Loading C, 15 ,, ,, ,, ,,	$0 \cdot 6\ n^2$ tons

where n is the number of revolutions per second, generally taken to be 5, or at most 6.

For the A loading the total hammer blow is divided equally between the two central coupled axles, or alternatively for short spans, between the two axles loaded with 1·25 units (page 482, fig. 2).

For the B loading the total hammer blow of $0 \cdot 5n^2$ tons is distributed as follows : $0 \cdot 18\ n^2$ tons on each of the central coupled axles and $0 \cdot 07\ n^2$ tons on each of the outer coupled axles, or alternatively for short spans a hammer blow of $0 \cdot 278\ n^2$ tons is supposed to be concentrated on one of the two axles loaded with 1·25 units.

For the C loading the total hammer blow of $0 \cdot 6\ n^2$ tons is distributed as follows : $0 \cdot 217\ n^2$ tons on each of the central coupled axles, and $0 \cdot 083\ n^2$ tons on each of the outer coupled axles, or alternatively for short spans hammer blows of $0 \cdot 055\ n^2$ tons on one of the two axles loaded with 1·25 units, and $0 \cdot 278\ n^2$ tons on the other axle.

For rail bearers or small culvert superstructures, the hammer blow is by no means equally divided between the two wheels of an axle. For the A loading six-fifths of the hammer blow, and for the B and C loadings five-sixths of the hammer blow should be calculated as applied to one wheel.

To provide against lurching in the case of deck spans each girder should be calculated to carry five-eighths of the total load instead of one-half, equivalent to an increase of 25 per cent. In the case of through bridges the corresponding maximum proportion of the load carried by each girder should be calculated on the supposition that either rail load of any track may be subject to an increase of 25 per cent. and the remaining rail load to a decrease of 25 per cent.

To cover the effect of rail joints and other irregularities provision should be made for an additional concentrated load of $\frac{n^2}{6}$ tons on each track, equivalent to a distributed load of $\frac{n^2}{3}$ tons, where n as before is the number of revolutions per second.

These additions to the British Standard Unit Loading make up the total impact allowance in accordance with the recommendations of the Bridge Stress Committee, except that for long spans the effect of the synchronization of two locomotives is considered.

Temperature Variation and Stresses.

Variation of Temperature and Expansion.

Records which have been made at Greenwich show the average range for twenty years to be 11° below and 13° above the mean temperature. The maximum annual range of temperature in the shade may be 85°, and in the sun as much as 137°. The theoretical expansion which these temperatures would cause in steel would be $\frac{5}{8}$ in. and 1 in. respectively in a length of 100 ft.

The observed movement in the Forth Bridge is $\frac{7}{16}$ in. per 100 ft., while the rise in the crown of one of the cast-iron arched ribs of the Old Southwark Bridge was $1\frac{1}{4}$ ins., the length of the chord of the intrados of which was 246 ft. and the versed sine, 23 ft. 1 in.

In tropical and sub-tropical countries the variation between the highest day temperature and the lowest night temperature may be as much as 150° F. or more.

Provision for Expansion.

For spans less than 40 ft. it is not customary to provide bearings with means of expansion, but for spans in excess of this, provision for expansion to the extent of 1 in. in every 100 ft. should be allowed. No allowance is usually made for expansion and contraction in the direction of the width of the structure, but in large and wide bridges it will be necessary to consider this.

Temperature Stresses.

The forces due to unequal temperatures on different parts of a lattice girder bridge act very slowly and, due to the slow yielding property of ductile metals such as steel, cause the parts of the frame to adjust themselves to the gradual stresses to which they are subjected. Nevertheless temperature stresses in large bridges must be carefully considered.

If the ends of a bar or portion of a bridge are rigidly fixed, a difference of temperature of 100° F. will cause a stress of 8·51 tons per sq. in.

Let

l = length of a bar at 0° F. (in feet); l_1 = its length at t° F. (in ft.); ϕ = coefficient of linear expansion for 1° F.; E = Young's modulus,

then

$$\text{Elongation for } t^\circ = \phi . t . l; \quad l_1 = l(1 + \phi t);$$

$$\text{Stress per }^\circ\text{ F.} = \phi \times E \text{ tons per sq. in.}$$

Bedplates and Bearings.

Types of Bearings.

Bridges of spans below 100 ft. are not usually fitted with any special means for allowing free expansion, but are simply fitted with a sole plate resting on a bearing block, the holes in the sole plate at one end of the bridge being slotted to allow of movement. The bearing block may rest upon hair-felt or sheet lead.

Bridges over 100 ft. span should rest at one end on roller bearings of not less than 4 ins. diameter, or some other form of expansion bearing.

The average type of roller bearing is usually very liable to rust up and become filled with dirt, etc., that falls from the bridge, and so becomes practically useless, unless very carefully designed with a view to eliminating this source of trouble.

For bearings of large span bridges to be really satisfactory, they should have some means of providing for the deflection of the bridge under load. This is now arranged for by having some sort of knuckle at both ends of the bridge; a bearing of this type is shown in fig. 4, which, however, is not to be regarded as satisfactory as regards the roller portion, which should be protected from rust and dirt.

The pressure in pounds per lineal in. on rollers of mild steel **should not exceed 560*d*, where *d* equals the diameter of the roller in ins.**

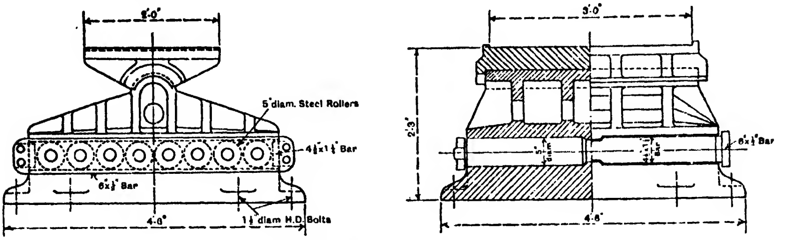


Fig. 4.—Combined Knuckle and Roller Bearing.

The bedplates should be of sufficient area to distribute the load over the masonry abutments, the limiting or working pressures being

Granite	**25 tons per sq. ft.**
Hard stone	**20 ,, ,,**
P.C. concrete (1-2-4)	**18 ,, ,,**

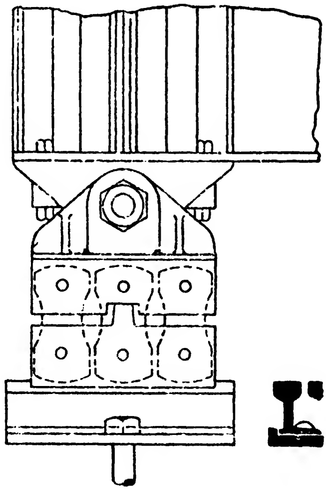


Fig. 5.—Combined Rocker and Roller Bearing.

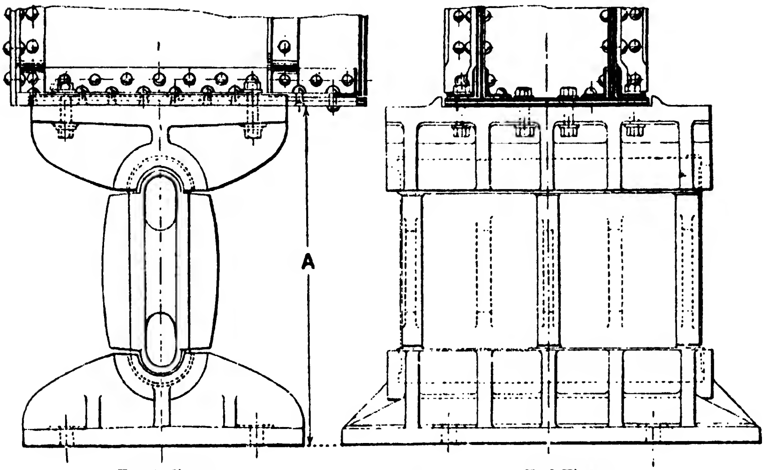
A form of end bearing is shown in fig. 5, which is a combined rocker and expansion bearing for the free end of the bridge. The pin in this case takes the place of the knuckle in providing for deflection.

Rigby's Patent Expansion Rocker Bearing.

A very excellent bridge bearing for spans of 100 ft. and over, is the one designed and patented by the late Mr. H. Rigby, M.I.C.E.

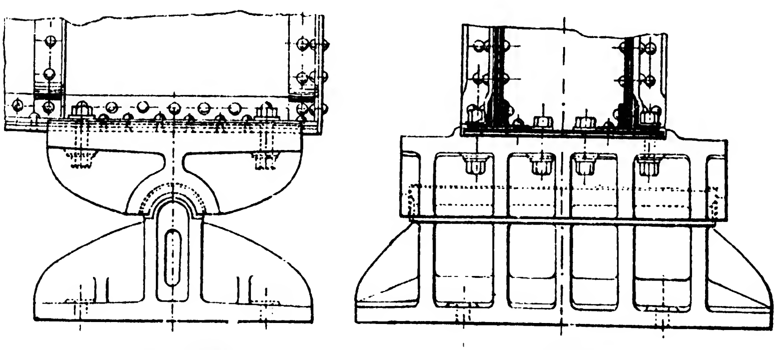
The bearing for both fixed and free ends is shown in figs. 6 and 7.

In this type the knuckles provide for deflection and the rocker for expansion. These bearings are considerably cheaper than most other forms, much simpler in construction, and are quite immune from the effects of rust and dirt in operation.



Front View. End View.

Fig. 6.—Rigby's Patent Expansion Rocker Bearing.



Front View. End View.

Fig. 7.—Rigby's Patent Bearing for Fixed End.

Flooring.

Buckled Plate Flooring.

The resistance of square buckled plates, bolted or riveted down all round, is double the resistance of the same plate merely supported all round, and if the two opposite sides be wholly unsupported, its resistance is reduced in the proportion of 8 to 5.

The stiffness of buckled plates is as the square of the thickness, and inversely as the curvature.

Two inches curvature suffices for 4 ft. square and ¼ in. thick.

Ordinary plates are made 3 ft. and 4 ft. square.

Weight and Safe Load of Buckled Plates, 3 ft. square.

No.	Thickness of Metal.	Weight in lbs.	Safe Load distributed in Tons.			
			Passive.		Impulsive.	
			Per Plate.	Per Sq. Ft.	Per Plate.	Per Sq. Ft.
1	18 B.W.G. .	17·3	0·27	0·03	0·20	0·022
2	16 ,, .	23·6	0·43	0·048	0·32	0·036
3	12 ,, .	38·7	0·64	0·071	0·48	0·053
4	⅛ in. . .	45·0	1·00	0·112	0·75	0·083
5	3/16 in. . .	67·5	2·5	0·278	1·7	0·180
6	¼ in. .	90·0	4·5	0·5	3·0	0·333
7	5/16 in. . .	112·5	6·2	0·689	4·7	0·522
8	⅜ in. . .	135·0	9·0	1·0	1·8	0·755

The safe loads may be doubled for buckled plates of puddled steel.

Buckled plates are usually made 3 ft. to 6 ft. square and ¼ in. to ⅜ in. thick. They can also be obtained in long lengths having several buckles to the plate.

To calculate the load uniformly distributed over a buckled plate, which will crush it—the plate being square and fastened all round the edges—multiply the depth to which the plate is buckled by the square of the thickness, both in inches, and by 165; the product will be the crushing load in tons (nearly). Central load which will crush a buckled plate, about one-third of uniformly distributed load. *(Rankine.)*

Corrugated Plates and Troughs.

These plates and troughs are made of ordinary steel plate and can be obtained in sizes from 2½ ins. deep to 16 ins. deep, and in lengths up to 30 ft., without joint in the smaller sizes and up to 40 ft. in the larger sizes and thicknesses. The advantages claimed are:—

(1) Ordinary plate is used and pressed in moulds in such manner that no deformation of the plate occurs.

(2) The sections can be made of any thickness to suit requirements and are therefore very economical from the point of view of weight.

The smaller sizes are manufactured with several corrugations in one plate and the larger sizes with only one trough or corrugation.

Distribution of Load.

It may be assumed that the wheel loads are distributed over 5 ft. run of the floor if the sleepers are parallel with the troughs and 10 ft. if at right angles to the troughs.

The strength and stiffness of trough or corrugated flooring, when riveted up, is considerable. Each trough may be treated as a girder, but then each girder is connected to its fellow; thus when the load is applied to, say, one trough it cannot deflect without dragging down its adjoining troughs for some distance on each side.

Floor Troughing.

Troughs, fig. 8, are composed of special rolled sections of the form of splayed channels and can be obtained in many sizes. The advantages claimed are that the webs are thin and the bottoms and tops thick, and so approach what is required from the girder point of view, also that the riveting is situated at the neutral axis, where the bending stresses are a minimum.

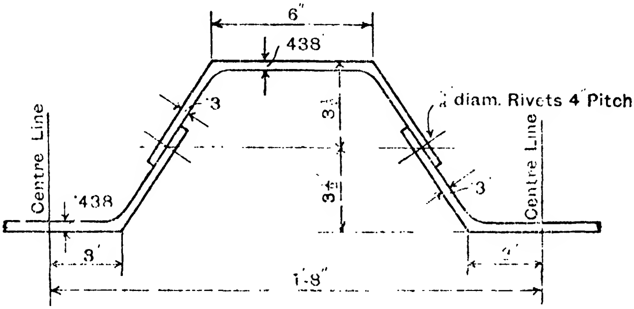
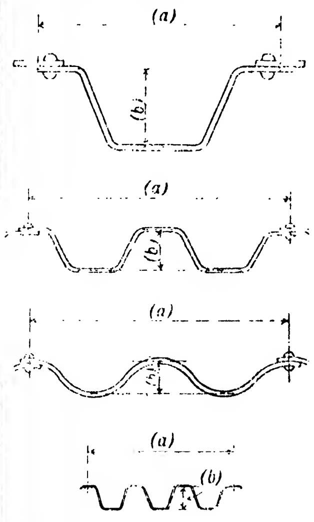


Fig. 8.—Steel Trough.

Weight over sq. ft. of area covered = 24·52 lbs Section Modulus = 21·62.

Steel Troughing.



Section.	Depth Centres. (b)	Normal Width Centres. (a)		Thicknesses.	Max. Length Pressed.	
	Ins.	Ft.	Ins.	Ins.	Ft.	Ins.
A	15	2	8	$\frac{3}{8}$–$\frac{1}{2}$	40	0
B	12	2	8	$\frac{3}{8}$–$\frac{3}{4}$	40	0
C	10	2	6	$\frac{3}{8}$–$\frac{3}{4}$	40	0
D	9	2	0	$\frac{3}{8}$–$\frac{3}{4}$	40	0
F	7½	2	0	[illegible]	40	0
G	6½	1	8	$\frac{1}{4}$–$\frac{5}{8}$	35	0
H	6	1	8	$\frac{1}{4}$–$\frac{5}{8}$	35	0
I	5	2	8	$\frac{3}{16}$–$\frac{7}{16}$	30	0
K	4	2	8	$\frac{3}{16}$–$\frac{7}{16}$	30	0
L	3	2	0	$\frac{3}{16}$–$\frac{3}{8}$	30	0
O	4	2	8	$\frac{3}{16}$–$\frac{7}{16}$	20	0
P	3	2	0	$\frac{3}{16}$–$\frac{7}{16}$	20	0
M	3	1	6	$\frac{1}{8}$–$\frac{3}{16}$	10	0
N	2½	1	0	$\frac{1}{8}$–$\frac{3}{16}$	10	0

CAMBER.

The provision of an upward camber in bridge girders gives a better appearance to the bridge, and, moreover, obviates the additional stress, due to the centrifugal force of the train running round the curve, which would otherwise occur due to deflection.

Bridges of 100 ft. span and upwards are usually given a camber of 1 in. for every 40 ft. in length. Plate girders of small dimensions are not usually cambered.

For parallel or bow-topped lattice girders the increase in the length of the top boom and bays therein can be found as follows:—

If

c = camber in ins.; N = number of bays; D = depth between centres of flanges: l = length of one bay in ft.; L = horizontal length of bottom boom in feet; x = increase in length of one bay in top boom in inches; y = total increase in length of top boom,

then $$x = \frac{8cD}{N^2 l},\quad y = \frac{8cD}{Nl}\quad \text{when } c = 1 \text{ in. in } 40 \text{ ft.};\quad x = \frac{LD}{5N^2 l},\quad y = \frac{LD}{Nl}.$$

RIVETING.

Rivet Pitches.

The pitch of rivets in the direction of the stress should not exceed 6 ins., or 16 times the thickness of the thinnest outside plate or bar, nor less than 3 times the diameter of the rivet.

At the ends of compression members the pitch should not exceed 4 times the diameter of the rivets for a distance equal to twice the width of the member.

In the flanges of girders the pitch should not exceed 4 ins. In webs composed of two or more plates, the rivets, used solely for causing the several thicknesses to act as one plate, should not be spaced more than 12 ins. apart in any direction.

The distance from the centre of the rivet (or bolt hole) to the edge of a plate or bar should not be less than 1½ times the diameter of the rivets in the case of planed or rolled edges. When practicable, the distance should be at least 2 diameters of the rivet, but not to exceed 8 times the thickness of the plate.

In chain riveting the distance between centre lines of adjacent rows should preferably not be less than 3 diameters of rivet, and in no case less than 2½ diameters.

In zigzag or staggered riveting the distance between centre lines of adjacent rows should preferably be not less than 2½ diameters, and never less than 2 diameters.

Diameters of Rivets.

There are various rules for obtaining the most suitable diameter of rivets, but they are not often used in structural steelwork.

In practice, a ¾ in. or ⅞ in. diameter rivet is adopted whenever possible, and it is best not to use any formula to obtain the diameter in terms of the thickness of the plate.

The following is an exceedingly good rule and is almost universally followed:—

¾ in. rivets for ⅜ in. plates.
⅞ in. ,, ,, ½ in. ,,
1 in. ,, ,, ⅝ in. ,, and over.

It is difficult to get rivets of larger diameter than 1 in. closed by hand.

SECTION XIV

PART II

PLATE GIRDER BRIDGES.

LIVE LOAD — WEBS AND STIFFENERS — JOINTS — CROSS GIRDERS—RAIL AND ROAD BEARERS—CURTAILMENT OF FLANGE PLATES — DEFLECTION — CANTILEVER PLATE GIRDER BRIDGES.

(Revised by J. D. W. Ball, A.M.I.C.E.)

LIVE LOAD.

Criterion for Maximum Bending Moment.

For a maximum bending moment under any load in a system of wheel loads on a girder, the load considered must be as far from one end of the girder as is the centre of gravity of all the wheel loads from the other end of the girder. Or, in other words, the load considered and the centre of gravity of all the loads in the girder must be equidistant from the ends of the girder. It is often possible to determine by inspection which load will produce the absolute maximum bending moment due to the passage of the live load.

Criterion for Maximum Shear.

The maximum end shear for a system of concentrated moving loads will occur when the reaction is a maximum.

The maximum positive shear in any section of a girder occurs when the foremost load is at the section, provided W is not greater than $\frac{P_1 L}{a}$. If W is greater than $\frac{P_1 L}{a}$, the greatest shear will occur when some succeeding load is at the point.

In this case—

W = the sum of the loads; P_1 = the foremost load; L = the length of the span; a = the distance apart of the foremost and next load in succession.

In plate girders with cross girders, to determine the position of wheels to give maximum shear in panel to left of any cross girder, one of the large wheels is always placed over the cross girder; the wheel so placed should satisfy the following conditions:—

(1) The load to left of the cross girder, and including the load over the cross girder, must be equal to or greater than the total load on bridge divided by number of panels.

(2) The load to left of the cross girder, and not including the load over the cross girder, must be less than the total load on bridge divided by number of panels.

It will sometimes be found that more than one arrangement of the loads will satisfy the above conditions. In such cases it is necessary to figure the shear for the different loadings and use the greater. (*American Bridge Co.*)

WEBS AND STIFFENERS.

Distance apart of Stiffeners.

The distance apart which stiffeners should be placed can be found by the following formula:—

$$L = \sqrt{\left(\frac{5}{K} - 1\right) 1500 t^2},$$

where

L = distance apart in ins.; t = thickness of web in ins.; K = stress per sq. in. to be provided for—*i.e.* $\frac{\text{total shear at a section}}{\text{net area of web}}$.

The position of stiffeners should be so arranged as to cause no irregularity in rivet pitches, *i.e.* the pitch should alter if it is necessary at a stiffener. All stiffeners should bear tightly at the top and bottom against the flange angles, and wherever possible, stiffeners should be placed at all web joints.

In British practice, stiffeners are seldom placed at distances wider apart than the depth of the girder, with 6 ft. as a maximum when the thickness of the web is less than $\frac{1}{60}$ of the unsupported distance between the flange angles.

Stiffener angles over the bearings should be designed to carry the entire shear at that point without exceeding the safe stress (packings should not be included in the area); they should be proportioned as struts with fixed ends, having a length equal to ¾ of the depth of the girder.

Compound Web Plates.

Web plates composed of two or more plates should not be used where the total thickness is less than 1 in. (For Riveting, see p. 230.)

Reduction of Web Thickness.

The following is a graphical method of obtaining the necessary lengths of web plate of given thickness or the thickness of web plates of given length. The thickness of web plate required at the abutments must first be calculated in each case.

Distances such as 'a' measured to ¾ in. scale in fig. 1, give the reduction in thickness permissible.

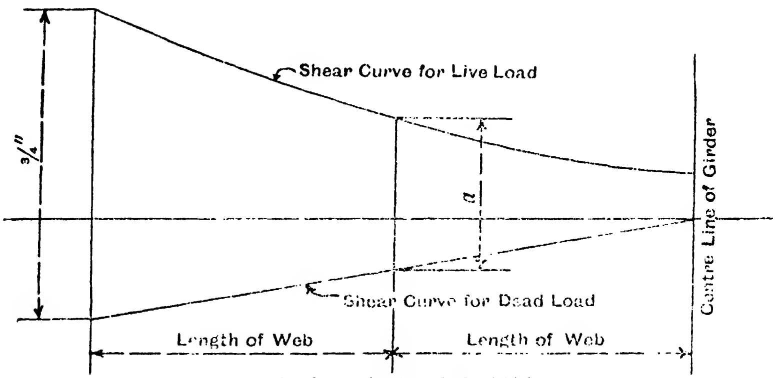


FIG. 1.—Reduction of Web Thickness.

JOINTS.

Thickness of Cover Plates.

The thickness of cover plates may be determined as follows:—

$$\text{Single covers} = t + \frac{t}{8}$$

$$\text{Sum of thickness of double covers} = t + \frac{t}{4}$$

where

t = thickness of plates joined.

Flange Angle Splices.

If a girder is so long that the angles cannot be obtained in one length, or if for any other reason it is necessary to splice the angles and the girder is to be shipped in one length, the splices in the angles should be as far apart as possible and on opposite sides of the web plate. Where the area of the flange angle is too great to be conveniently spliced by one angle, a splice angle with area equal to about ¾ of the area of the flange angle should be placed on each side of the girder.

Web Splices.

When part of the web is counted as flange area and the bending moment at the web joint actually requires this area of web to bring the flange stress down to the maximum allowed, there should be, in addition to the number of rivets required to take shear, sufficient rivets as near each flange angle as practicable to take the proportional part of the flange stress taken by the piece of web.

The net area of these splice plates must be made equal to area of web used as flange area multiplied by the depth of girder divided by the distance centre to centre of splice plates.

The number of rivets required to equal the net area of two splice plates, multiplied by the unit stress in flanges and divided by least value of one unit (shear or bearing) in web plates.

It is desirable to arrange the rivets so as not to cut out more than two rivet holes in the line next the joint, or at the ends of the plates; see fig. 2.

The vertical splice plates should have rivets enough on each side of splice to take the vertical shear at the position of the splice.

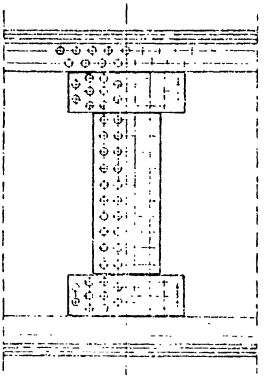


FIG. 2.—Web Joint.

Flange Splices.

If for any reason the flange plates have to be joined, a cover must be provided at each splice; the rivets connecting the cover must be such that their strength is equal to the plate spliced.

It is often possible to arrange the splices in a stepped manner with only one cover plate; such a joint is shown in fig. 3, and is suitable for a shipping joint.

The final cover must be at least equal in thickness to the thickest plate spliced, and, in the example, serves as a cover for the four plates.

The distance apart of the 'steps' must be such that the strength of the rivets between each is not less than will suffice to develop the full strength of each plate.

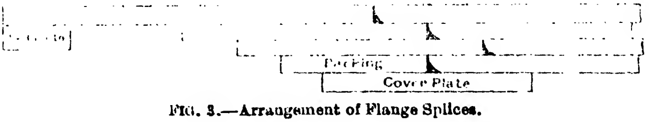


FIG. 3.—Arrangement of Flange Splices.

Rivets in Connection Plates.

When a single system of lateral bracing is used, the number of rivets connecting the plates to the girder should be sufficient to take the sum of components (in direction of girder) of both laterals connecting on the plate. (See fig. 4.) If AB requires 5 rivets and BC requires 4 rivets lay off in the direction of *ab* to any convenient scale a distance *by* of 5 and in the direction *bc* a distance *bw* of 4. Drop perpendiculars *wv* and *yx* and scale the distances *vb* and *bx*. The sum of these two distances represents the number of rivets required to connect the plate to the girder.

If a double system of bracing is used, they are proportioned for tension only, and the number of rivets required to connect the plate to the girder will be the number required to take the component of one lateral in the direction of the centre line of the girder.

There should be rivets enough connecting the plate to the cross girder or strut (if any) to take the component of one lateral in the direction of the cross girder or strut.

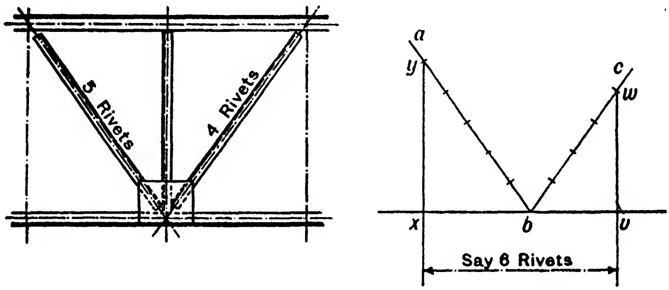


FIG. 4.—Rivets in Connection Plates.

CROSS GIRDERS.

Distance apart of Cross Girders.

. Generally speaking, it is not economical to place the cross girders in railway bridges closer together than 5 ft.; at intervals shorter than 5 ft. the load will usually consist of the heaviest axle load of the live load, while for intervals of 5 ft. and upwards it is necessary to consider the several positions that may be occupied by the axles in the heaviest portion of the live load.

The distance apart should be considered in relation to the centres of the heaviest wheel loads which will come upon them.

It is often found that increasing the spacing of the cross girders adds but little to the load which they will have to sustain, and it is usually economical to adopt a wide spacing, say from 6–10 ft.

In lattice girders the spacing of cross girders, depth of truss, and panel length must all be considered in relation to each other, for the cross girders will usually be placed at panel points to avoid secondary stresses in the booms.

The economy of placing cross girders far apart only applies in a modified degree to highway bridges, as in such cases the dead weight of the roadway forms a considerable part of the total load.

PROPORTIONS.

The depth span ratio of cross girders should be not less than 1 : 10, preferably 1 : 8, otherwise in bridges with buckled plate flooring considerable trouble is likely to occur with the rivets holding the plates down, working loose, or the plates fracturing at the seatings.

CONNECTIONS.

The connections of the main girders should be rigid, and 50 per cent. should be added to the number of rivets required to take the shear to resist the bending action.

In plate girders the web stiffeners may be made to considerably strengthen the end connections of cross girders, and in lattice girders a bracket plate of ample proportions should be fitted.

RAIL AND ROAD BEARERS.

Rail Bearers.

These should be in general placed 5 ft. apart, and must be proportioned to take the greatest wheel loads imposed by one, two, or more axles, as the case may be. Although they act as continuous girders, much depends upon the rigidity of the end connections and the deflection of the cross girders. Where they rest upon the cross girders they have a greater chance of acting continuously; but usually for the purpose of designing their section they will be treated as separate spans.

Road Bearers.

The distance apart of road bearers will be decided in relation to the width of the bridge and the centres of the wheels composing the maximum live load, and will be greatly decided by the type of bridge floor adopted. They should be designed as supported at the ends.

Proportions of Rail and Road Bearers.

The depth should not be less than $\frac{1}{12}$ of their span.

Curtailment of Flange Plates.

Unless this is done the girders will be stronger at the abutments than they need be. This can be remedied by either of the following methods:—

(1) Keeping the depth constant and varying the thickness of the flanges along the length.
(2) Varying the depth of the girder and keeping the flange thickness constant. By this method the girder theoretically becomes parabolic.

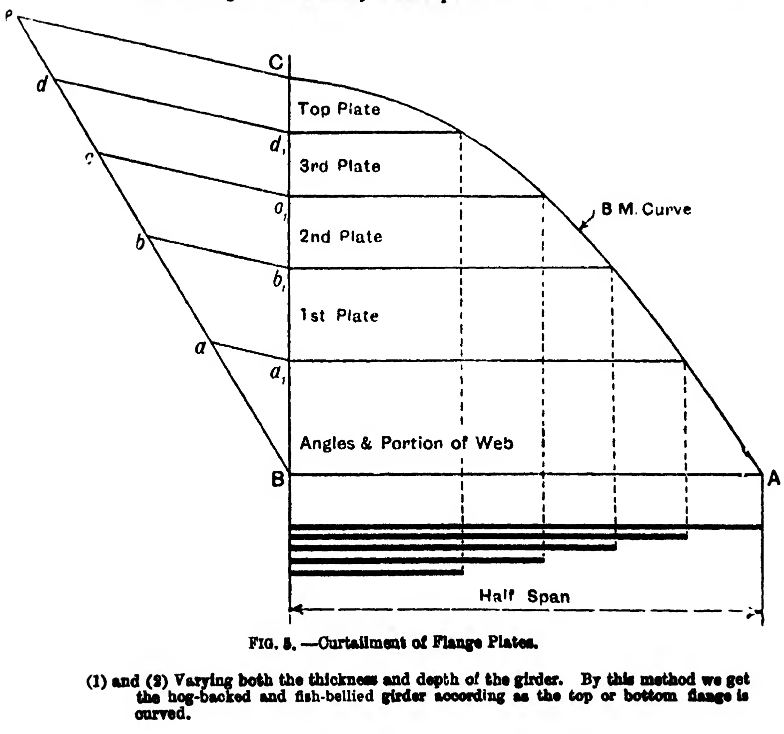


FIG. 5.—Curtailment of Flange Plates.

(1) and (2) Varying both the thickness and depth of the girder. By this method we get the hog-backed and fish-bellied girder according as the top or bottom flange is curved.

There are variations of this, for both top and bottom flanges are sometimes curved.

In most cases (1) is the most economical on account of there being less workmanship required in the manufacture.

The curtailment of flange plates in method (1) is carried out as follows :—

Referring to fig. 5, ABO is the bending moment curve on half the girder AB.

On any inclined line Be set out points a, b, c, d, as follows —

Be = Modulus of section of total flange at centre of girder (including portion of web if allowed for in calculations).
de = Modulus of section of top plate (net).
cd = " " 3rd " "
bc = " " 2nd " "
ab = " " 1st " "
Ba = " " bottom plate and angles (including portion of web if allowed for in calculations).

Join eO and draw parallels dividing the vertical Bc into similar spaces.

Lines drawn horizontally to meet the bending moment curve will now give the theoretical necessary lengths of the flange plates.

In practice the plates are continued for from two to four pitches of the rivets and the first plate over the main angles is always continued to the ends of the girder.

A close approximation is obtained by substituting the net area of each plate, and of the bottom plate and angles for the modulus of section referred to above.

Deflection.

Elastic Deformation.

Is one which disappears entirely on the removal of the external forces causing the stress.

The amount of deflection in girders depends mainly upon the following :—

(1) The length of the girder;

(2) The unit stress in the flanges.

The deflection arises from the top flange being compressed and the bottom flange being extended, and the amount of deflection is not materially affected by the kind of web in deep girders.

Mohr's Theorem.

A loaded beam takes up the same form as an imaginary cable of the same span, which is loaded with the bending moment curve on the beam and subjected to a horizontal pull equal to the flexural rigidity (EI). The deflected form of the beam is called the *elastic line* of the beam.

Graphical Construction for any Loading.

(Flanges of beam of uniform section.)

(1) Draw the bending moment diagram for the given loading of the beam.

(2) Treat the bending moment diagram as a load diagram and divide it up into narrow vertical strips; set down the mid-ordinates on a vector line.

(3) Draw the vector diagram with a polar distance equal to the flexural rigidity (EI).

(4) Draw the funicular polygon for this vector diagram and reduce it to a horizontal base, then the link polygon gives the elastic line.

The scale to which the deflections can be scaled is determined as follows :—

1 in. on the length of the bending moment diagram = x ft.
1 in. on the height of the bending moment diagram = y ft. tons.

Therefore in fig. 6 1 in. in height of any mid-ordinate of the bending moment diagram 1, 2, represents $d \times x \times y$ sq. ft. tons. Calculate the length of EI in sq. ft. tons to this scale $= \dfrac{EI}{144 \times d \times x \times y}$ inches), where E = modulus of elasticity in tons per sq. in., and I = moment of inertia in inch units); then the deflections will be to the scale of 1 in. = x ft.

It will be found convenient to reduce the length of EI for practical use, so the polar distance can be taken as $\frac{EI}{n}$, where n is some convenient divisor. The deflection scale will then be 1 in. = $\frac{x}{n}$ ft.

(*E. S. Andrews, B.Sc., M.I.C.E.*)

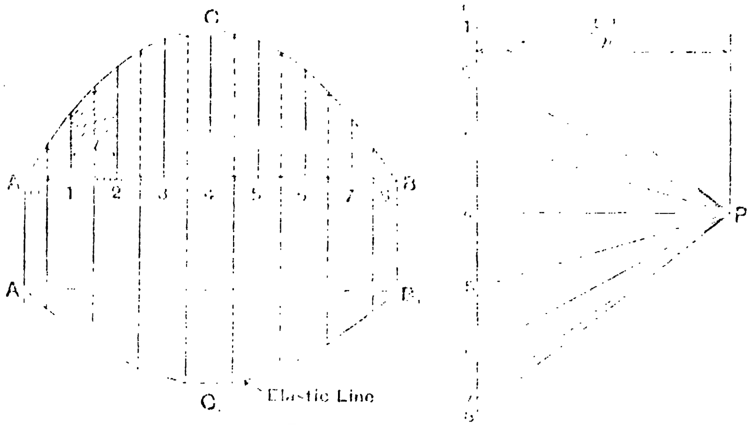


FIG. 6.—Graphical Determination of Deflection.

ACTUAL AND THEORETICAL DEFLECTION.

The difference between the actual and calculated deflection is due to various causes; in riveted girders the stiffness of riveted joints and covers, and the position and form of the floor, all tend to reduce the actual deflection, while poor quality of workmanship in the riveting and fitting together of parts tend to cause the actual deflection to exceed that calculated.

CANTILEVER PLATE GIRDER BRIDGES.

Cantilever plate girders, alternating with suspended spans, are being much used at the present time.

Kincardine-on-Forth Bridge.—On either side of the swing span there are seven 100 ft. spans with 50 ft. suspended girders in the second, fourth and sixth openings. The lower flanges are all arched to a radius of 290 ft. The web plates are 9 ft. deep at the supports and 5 ft. deep at the centre of each span. Across the 30 ft. width of roadway there are six of these girders, spaced at 6 ft. centres, the footpaths being carried on cantilever brackets in continuation of the diagonal bracing fitted between the girders at 10 ft centres. The bearings of the cantilever spans are alternately fixed and sliding. The projecting upper half of the ends of the suspended spans rest on socket and knuckle bearings on the projecting lower half of the cantilever ends, and these bearings at one end of each suspended span rest on a slide plate.

Storstrom Bridge.—Anchor spans are 189 ft. 7 ins. between bearings with 29 ft. 2 ins. cantilever arms. Suspended spans in alternate openings are 145 ft. 10 ins. Each span is carried on two main girders, 24 ft apart, with parallel flanges and 12 ft. built-up web plates. The anchor spans are supported on fixed and expansion cast steel bearings on alternate piers.

SECTION XIV

PART III

LATTICE GIRDER BRIDGES.

LIVE AND DEAD LOAD STRESSES—PROPORTIONS OF BOOMS AND STRUTS—PIN-JOINTED MEMBERS—VARIATION OF STRESSES—PORTAL BRACING—DEFORMATION AND DEFLECTION.

LIVE AND DEAD LOAD STRESSES.

Criterion for Maximum Bending Moment and Maximum Shear.

The maximum bending moment at any joint in a bridge loaded with wheel loads will occur when the average load on the left of the section is the same as the average load on the whole span.

The maximum shear in any panel in a bridge loaded with wheel loads will occur when the load on the panel is equal to the load on the bridge divided by the number of panels.

(*Ketchum.*)

Maximum Loads on Inclined Braces.

The maximum tension in any inclined brace occurs when the leading axle is at the foot of the brace, the live load covering the larger segment of the bridge from the foot of the brace to the farthest support.

The maximum compression in any brace occurs when the leading axle is under the head of the brace; the live load covering the shorter segment of the girder to the nearest support.

(*R. J. Woods.*)

Shears and Counterbracing.

(*a*) The maximum live load shear (positive) in any panel of a truss is equal to the left reaction when all the joints at the right of the panel are loaded and all joints at the left of it are unloaded.

(*b*) The dead load negative shear in any panel at the right of the centre is equal to the amount of dead load between the panel and the centre of the span (including a half panel load at the centre if the truss has an even number of panels).

(*c*) No counter is required in any panel beyond the centre of the truss in which the dead load negative shear exceeds the maximum live load positive shear; but in each panel beyond the centre in which the live load positive shear can exceed the dead load negative shear, a counter is required.

Stresses in Parallel-Flanged Girders with Uniform Rolling Load.

Divide span into a number of equal parts, one less than the number of bays (see fig. 1), obtaining points such as a, a_1. On the base A_1B_1 set up length $A_1O = \frac{PL}{2}\left(\frac{b-1}{b}\right)$, where P = intensity of load, L = span of girder, b = the number of bays.

Through O draw a parabola with vertex at B, and project points such as a, a_1, down to meet the parabola in points such as e, e_1, and project horizontally across bays as shown.

Then the maximum stresses in the diagonals for passage of the live load from right to left are obtained from this curve. Negative shears in a similar manner.

The position of the load to give maximum stresses in diagonals is obtained from

$$x = \left(\frac{n.y}{b-1}\right),$$

where

x = distance travelled by the load beyond end of nth bay, y = length of bay, and b = number of panels.

(*E. S. Andrews, B.Sc., M.I.C.E.*)

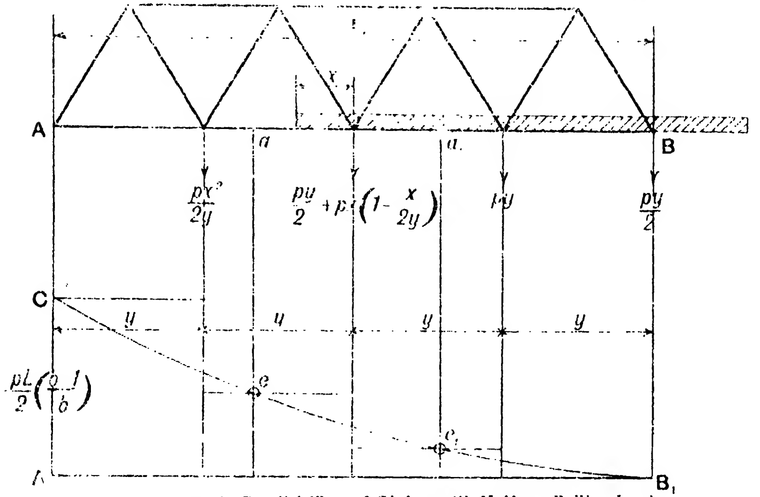


FIG. 1.—Stresses in Parallel-Flanged Girders with Uniform Rolling Load.

Distribution of Dead Load.

For spans under 200 ft. the total dead load shall be assumed to act at the loaded chord. For 200 ft. and over the total dead load will be distributed at top and bottom chords as follows:—

On Loaded Chords.

(*a*) Half the load resulting from trusses;
(*b*) Weight of horizontal wind bracing in plane of chords;
(*c*) Weight of floor system, permanent way, etc.;
(*d*) Half load resulting from weight of cross bracing in the case of deck bridges.

On Unloaded Chords.

(*a*) Half load resulting from trusses;
(*b*) Weight of horizontal wind bracing in plane of chords;
(*c*) Half the load resulting from weight of cross bracing in the case of deck bridges and the whole of it in the case of a through bridge. (*Sir Wm. Arrol & Co., Ltd.*)

Stress in Members.

In parallel flanged lattice girders the stresses in the diagonals

$$= \frac{\text{load on diagonal} \times \text{length of diagonal}}{\text{depth of girder}}$$

i.e. load $\times$ sec. ϕ. ϕ being the angle which the diagonal makes with a vertical line.

Force Diagram.

A very convenient and rapid method of determining the stresses in the members of a truss is by means of a force diagram as described on page 31.

If the truss contains relatively few bays, three or four diagrams will determine all the maximum stresses, having previously determined the total panel point loads which will produce the maximum stresses in the various members.

If the truss contains a fair number of bays it is more convenient to proceed by the method of influence lines or unit loads. A table is first prepared giving the stresses in each member, due to a unit load, applied in turn at each panel point. These stresses can be determined in one or two cases by drawing the corresponding force diagram, but when one or two have been done in this way, formulæ can be written down for determining the remainder and errors, if made, will be self-evident as the figures run in series. The headings for such a table are given below.

Member.	Stress due to a Unit Load at Panel Point.									Sum of Compression Stresses.	Sum of Tension Stresses.	Algebraic Sum of Stresses.
	4	3	2	1	0	1′	2′	3′	4′			

Generally speaking, the only variable panel point load will be that due to the weight of the truss and lateral bracing. The floor load will be constant for each panel, and it is also usual to consider the live load as constant at each of the loaded panel points, although the value of this constant panel point live load will generally depend on the length of the portion of the bridge assumed to be covered with live load.

The total dead load stress in each member in turn is determined by multiplying the variable panel point truss load by the corresponding unit load stress and summing these, by multiplying the panel point floor load by the algebraic sum of the unit load stresses, and adding the two results.

Live load chord stresses will be obtained in a similar way, but for web members, the live load stress may be tension or compression, according to which portion of the bridge is loaded, and these total tension or compression stresses will be obtained by multiplying the sum of the unit load tension stresses by the appropriate panel point live load, or the sum of the unit load compression stresses by the appropriate panel point live load.

Ritter's 'Method of Sections.'

This consists of drawing 'lines of section' cutting the bridge truss in not more than three of its members. These lines divide the truss into two parts, one of which is supposed to be removed, the external forces acting on the other portion alone being considered.

The stress in any one of the three members cut by the 'line of section' can be found by taking moments round the point of intersection of the other two.

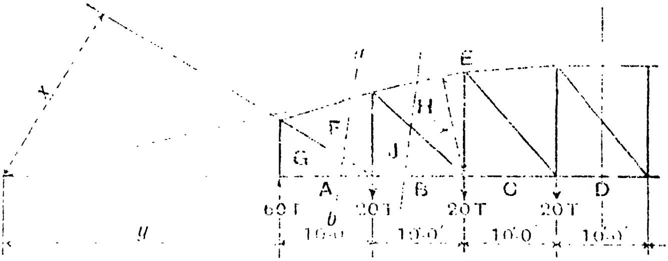


Fig. 2.—Ritter's Method of Sections.

This method is not applicable to finding the stresses in the flanges of parallel-flanged girders; for the flanges if produced would not meet.

In fig. 2 draw the sectional line ab, cutting FE, FG, GA; then,

Stress in FG $\times x = 60 \times y$, y being measured to the point of intersection of FE and AG.

It will be seen that the moments of the stresses on the members which meet at the point indicated become zero; and it is only necessary to equate the moment of the stress on the third member to the algebraic sum of the moments of all the external forces acting on the portion of the truss considered.

If it is required to find the stress in the member HE, with the live load in the position shown, then,

$$\text{Stress in HE} \times x' = (60 \times 20) - (20 \times 10).$$

Stresses in the vertical struts can be found in a similar manner.

Braced or Lattice Struts.

The least number of panels

$$= \frac{\text{Least radius of gyration of whole strut}}{\text{Least radius of gyration of unbraced portion}}$$

In the case of a strut composed of an angle at each corner, the least radius of gyration of unbraced portion must be taken as one of the angles.

Batten plates should be spaced from 2–3 times the overall width of the side braced, while the inclination of diagonals to the axis of the member should not be less than 45°.

The thickness of bars with single lacing should not be less than $\frac{1}{40}$ of their length; with double bars, not less than $\frac{1}{60}$ of their length.

Ties.

Ties should usually be designed to be of rigid construction, but may be of rolled flat bars except near the centre of the span, where they must be composed of rigid sections.

Counter bracing must be of similar construction to the centre ties.

Rolled bars forming long ties should be provided with distance pieces to reduce vibration.

Pin-jointed Members.

Pin Connections.

The tearing, shearing, and bearing strengths of the joint should be made as nearly as possible equal to each other, and to the tensile or compressive strength of the bar in which the joint occurs.

Usually the bearing stress on the pin is taken at somewhat less than for rivets, to allow for imperfect fitting of the pin.

The bending strength of pins is very important, for if the pin becomes distorted the distribution of the loads and the calculation of the stresses is very uncertain.

Secondary bending in the top booms and strut members of a bridge merit greater attention when they are connected with pin joints. In this country pin joints will not often occur anywhere other than in suspension bridges, but in America pin-jointed bridges are very common.

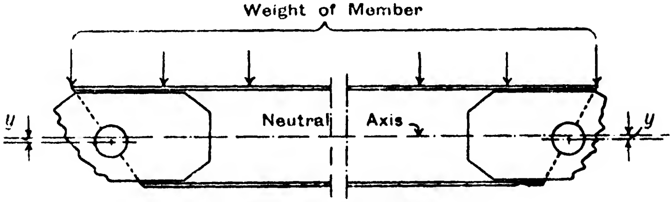


FIG. 3.—Pin-jointed Member with Eccentric Pins.

The position of pins may be so fixed that when the maximum stress (direct compression) comes upon the chord through the pins, it will produce an amount of bending moment equal to that produced by the weight of the member, but in the opposite direction.

In the ordinary type of top boom or end post section, this condition will be effected when the position of the pins is fixed at a distance y (see fig. 3), perpendicularly below the centre of gravity of the section, given by the formula

$$y = \frac{M}{Sc},$$

where M is the bending moment in inch pounds due to the weight of the member, and Sc is the total maximum direct compressive stress in the member in pounds.

Variation of Stresses.

Wöhler's Experiments.

When the elastic limit is not overpassed it is possible, under certain conditions, to fracture steel by repeating the stress a sufficient number of times. Wöhler showed that it was the range of stress, and not the maximum stress, that determined the number of applications before rupture.

The general conclusions are that:

(*a*) The greatest tensile stress a bar will bear for an indefinite number of times is for iron and steel about half its ultimate static breaking stress.

(*b*) The number of repetitions required to produce fracture is increased if the range through which the stress is varied is reduced.

(*c*) A bar will be strongest when exposed to varying stresses of the same kind, and weakest when exposed to stresses of opposite kinds, *i.e.* tension and compression.

The stress caused by a load suddenly reversed is three times the static stress, and a load suddenly applied causes a stress of twice the static stress. (Refer to 'Impact,' p. 500.)

Relation between Sudden Loading and Repetition of Stress.

The similarity between the results of experiments on variation of stress and sudden loading has given rise to many assumptions that Wöhler's results were really due to the effects of sudden loading, and that the two are really aspects of the same question.

Although there are many points which require to be decided in the consideration of these experiments, for practical reasons it seems best to allow for either one or the other, and not for both simultaneously in one member.

In the case of cross girders and vertical suspenders, allowance should be made for the full value of the elastic vibrations due to sudden loading; while rail bearers and main girders, both plated and latticed, of spans up to 100 ft., should be designed with an allowance for sudden loading or impact, using one of the impact formulæ evolved for the purpose.

In the flanges of girders and the principal members of arched and suspension bridges of large dimensions the stress variation takes place more gradually, and allowance for variation of stress can be made by adopting a reduced stress, based on the amount of variation of stress.

Launhardt-Weyrauch Method.

The working stress is varied according to the relative amounts of live and dead loads.

Members Subjected to Tension Only.

$$\text{Working stress} = \frac{f}{1\cdot 5}\left(1 + \frac{\text{minimum load}}{2 \times \text{maximum load}}\right) \text{ tons per sq. in.,}$$

being the static or dead load working stress.

Dynamic Method.

$$\text{Working stress} = \frac{f}{1 + \dfrac{\text{live load.}}{\text{total load}}};$$

$$\text{The sectional area of a bar in tension} = \frac{\text{maximum load}}{\text{working stress}}.$$

Members Subjected to Alternate Stresses.

Members of bridges subjected to alternate tension and compression should be proportioned as struts, to resist the greater stress added to ½ of the lesser stress, except in the case of wind bracing, where the members should be proportioned to resist the greater stress. The sum of the stresses should be used in designing the connections.

Portal Bracing.

Design.

There are many types of portal bracing which may be used; A to F, fig. 4, show some common forms.

The fixing of the lower ends may be assumed as either hinged or fixed.

They may be considered as fixed by virtue of the direct stress due to vertical loading, when $V \times \frac{1}{2}s$ is $>$ M max. at A or B with fixed ends; $M \text{ max.} = \frac{Rd}{4}$, where V = total stress in the column.

The following treatment for the calculation of the stresses is due to Mr. Milo S. Ketchum in 'Steel Mill Buildings,' in which book he deals with the stresses in many types, with both hinged and fixed ends.

Stresses in Portal (a). Columns Hinged.

The deflections in the columns are assumed to be equal, then,

$$H = H_1 = \frac{R}{2}.$$

Taking moments about the foot of the windward column,

$$V_1 = -V = \frac{Rh}{s},$$

Having found the external forces, the stresses in the members may be found either by algebraic or graphical methods.

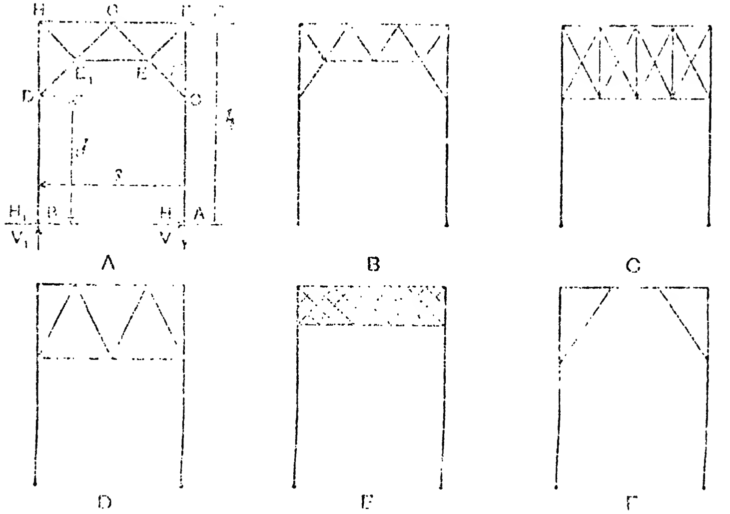


FIG. 4.—Types of Portal Bracing.

To obtain the stress in member GC, pass a section (Ritter's method of sections), cutting GF, EF, and GC, and take moments of the external forces to the right of the section about point F as centre.

$$GC = -\frac{Hh}{(h-d)\sin\phi}.$$

But $$H = \frac{R}{2}, \text{ and } (h-d)\sin\phi = \frac{s}{2} \times \cos\phi.$$

Substituting these values in the above expressions,

$$GC = -\frac{Rh}{s\cos\phi} = -V\sec\phi.$$

Resolving at C and F we have: stress in EF = 0, and also stresses in EE_1 and HE_1 = 0.

To obtain stress in GD, pass a section cutting HG, HE, and GD, and take moments of the external forces to the left of the section about point H as centre.

$$GD = \frac{Hh}{(h-d)\sin\phi} = +V\sec\phi;$$

To obtain stress in GF, pass a section cutting GF, EF and GC, and take moments of the external forces to the right of the section about point C as centre.

$$GF = + \frac{R(h - d) + Hd}{h - d}.$$

To obtain stress in HG, pass a section cutting HG, HE, and GD, and take moments of the external forces to the left of the section about point D as centre.

$$HG = - \frac{Hd}{h - d}.$$

The stress in the windward post AF is zero above and V below the foot of the knee brace C the stress in the leeward post is zero above and V_1 below the foot of the knee brace D.

The shear in the posts is H below the foot of the knee brace, and above the foot of the knee brace is given by the formula—

$$S = \frac{Hd}{h - d} = \text{stress in HG}.$$

The maximum moment in the posts occurs at the foot of the knee braces C and D, and is

$$M = Hd.$$

The stresses in portal (*b*) are found in a similar manner, but it will be found that all members are stressed in portals (*b*) and (*d*).

Deformation and Deflection.

Actual and Theoretical Deflection.

There is usually a difference between the actual and calculated deflection of framed structures. This is due to various causes, principally the rigidity and stiffness of the joints or otherwise, and the actual deflection is usually in excess of that calculated.

Contrary to statements that the amount of deflection is independent of any change of form which may take place in the web, and is not affected by the form of web, it is found that the deformation of the web members contributes largely to the total deflection.

Deflection of a Lattice Girder.

The theoretical deflection of a framed structure can be found both analytically and graphically when the elastic deformation of each member caused by the stress in it is known.

The following method is that developed by Maxwell and Mohr, and is taken from Sir W. Arrol & Co.'s Handbook :—

Example.—Find the deflection at the centre of a lattice girder 100 ft. span, 10 ft. deep, and loaded with a distributed load of 1 ton per ft. run (see fig. 5).

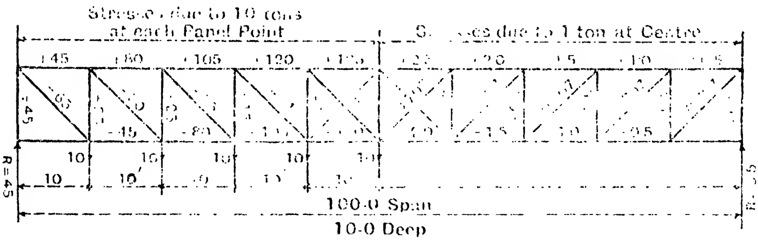


FIG. 5.—Deflection of Lattice Girders.

Let

L = length of any member in ft. ; l = length of any member in ins. ; S = total stress in any member due to 100 tons distributed load ; P = stress per sq. in. in any member due to 100 tons distributed load ; ϕ = total stress in any member due to 1 ton placed at the point at which it is required to find the deflection ; A = gross area of any member in sq. ins. plus 5 to 10 per cent. to allow for the effect of covers, joints, etc. ; E = modulus of elasticity of steel which may be taken as 12,000 tons per sq. in.

First calculate the stresses in each member due to the distributed load of 100 tons and obtain the sectional areas of the members; then ascertain the stress (U), in each member due to the load of 1 ton at the point where the deflection is required.

The elastic deformation of any member is $\frac{S}{EA}$ (or $\frac{LS}{1000\,A}$, if L is in ft.).

The results are then placed in tabular form as follows :—

Member.	L. Length	S. Total Stress	A. Area.	P. Stress per Square Inch.	$\frac{Pl}{E}$.	U.	Contribution to Deflection $\frac{PlU}{E}$	Summation.
Top Boom	Ft.	Tons.	Sq. Ins.	Tons.	In.	Tons	In.	In.
Bay 1 . .	10	+ 45	9	+5	+0·05	+0·5	+0·025	
,, 2 . .	10	+ 80	16	+5	+0·05	+1·0	+0·050	
,, 3 . .	10	+105	21	+5	+0·05	+1·5	+0·075	
,, 4 . .	10	+120	24	+5	+0·05	+2·0	+0·100	
,, 5 . .	10	+125	25	+5	+0·05	+2·5	+0·125	
								+0·375
Bottom Boom								
Bay 1 . .	10	Nil.	Nil.	Nil.	Nil.	Nil.	Nil.	
,, 2 . .	10	− 45	7·5	−6	−0·06	−0·5	+0·03	
,, 3 . .	10	− 80	13·4	−6	−0·06	−1·0	+0·06	
,, 4 . .	10	−105	17·5	−6	−0·06	−1·5	+0·09	
,, 5 . .	10	−120	20·0	−6	−0·06	−2·0	+0·12	
								+0·300
Web.								
Tie 1 . .	14·14	− 65	13	−5	−0·0707	−0·707	0·05	
,, 2 . .	14·14	− 50	10	−5	−0·0707	−0·707	0·05	
,, 3 . .	14·14	− 35	7	−5	−0·0707	−0·707	0·05	
,, 4 . .	14·14	− 21	4·5	−5	−0·0707	−0·707	0·05	
,, 5 . .	14·14	− 7	1·4	−5	−0·0707	−0·707	0·05	
								+0·250
End Post .	10	+ 45	11·25	+4	+0·04	+0·5	+0·02	
Strut 1 . .	10	+ 35	8·75	+4	+0·04	+0·5	+0·02	
,, 2 . .	10	+ 25	6·25	+4	+0·04	+0·5	+0·02	
,, 3 . .	10	+ 15	3·75	+4	+0·04	+0·5	+0·02	
,, 4 . .	10	+ 5	1·25	+4	+0·04	+0·5	+0·02	+0·100
							Half sum	+1·025 2
						Total	deflection	+2·050

The required deflection of the girder is thus found to be 2·05 in., the flanges contributing 1·35 in. and the web 0·70 in.

SECTION XIV

PART IV

SWING AND BASCULE BRIDGES.

PARTICULARS OF BRIDGES—WORKING STRESSES—FRICTION AND LUBRICATION—HYDRAULIC LIFTING AND SLEWING MACHINERY—CIRCULAR GIRDERS.

SWING BRIDGES.

The most usual types are either of the 'centre bearing' or 'rim bearing' type, and may be either 'single leaf' or 'double.'

The depth of a single leaf swing bridge is four times greater than that required for a double-leaf bridge of the same span, when the bridge is treated while in motion as a cantilever.

On the other hand, the length of a single-leaf swing bridge is less than the combined lengths of the two leaves for the same opening, but the double leaf type requires two sets of operating machinery.

The full advantages of swing bridges are only to be obtained when they span two openings; in such cases the necessity for counterpoises or kentledge does not exist and the effects of wind pressure are to a great extent neutralised.

A further classification of types is as follows:—

(a) Those which turn entirely upon rollers, *i.e.* the rim bearing type.

(b) Those where the weight is proportioned to be borne partly by rollers and in part by a centre pivot.

(c) Those swung entirely upon a pivot.

(d) Those which are lifted on a water centre by hydraulic power.

(e) Those which are lifted on a water centre by hydraulic power, but not sufficient to lift the whole weight.

In the latter type the tail is heavier than the toe, and the excess weight is borne by rollers running on a roller path, and adds stability to the bridge when swinging.

In type (e) of the above classification should also be included those swing bridges which are carried on a circular pontoon, the effect of which is practically a low-pressure hydraulic ram.

The general principle of these water-borne swing bridges is as follows:—A sealed circular pontoon or buoy is placed under the centre of gravity of the superstructure, and suitably and rigidly connected thereto in such manner as to be always submerged with a constant buoyancy and capable of being turned with the superstructure.

The upward pressure due to the buoyancy of the pontoon is less than the total weight of the bridge, the remainder of the weight resting upon roller paths which are carried on a gridiron or circular girder supported upon piles in the bed of the river as in several bridges over the river Weaver, Cheshire, designed by Mr. John A. Saner, M.I.C.E.

One of these bridges consists of two braced girders 112 ft. long, 17 ft. high over the centre of gravity, with a 19 ft. 6 ins. roadway, and weighs 303 tons, of which 255 tons is water borne.

In this case the consumption of electric power to turn the bridge is 0·24 of a unit for each swing, a swing including the withdrawal of the wedges, opening and closing of bridge and replacement of wedges.

It is stated that this type of swing bridge is more economical in operating expenses than any other swing bridge of equal size.

A swing bridge at the Spencer Dock, Dublin, has the pontoon working in an outer caisson, the 5 per cent. excess weight of the bridge being carried upon a centre pivot placed at the bottom of the pontoon. Lateral rollers with the roller path fixed to the underside of the bridge are provided in this design.

Lattice girders are preferable for all spans in excess of 100 ft. on account of their much less resistance to wind pressure.

In all cases where rollers at the tail end are provided, an excess of counter weight should be provided, so that the rollers are loaded when the bridge is swinging with not less than 10 tons when the bridge is covered with snow and allowance has been made for the increase in weight of the roadway and footpaths due to wet.

Hydraulic presses or wedges should be provided at heel and toe to relieve the rollers and pivot of weight when the bridge is in the fixed position for sustaining road traffic.

BASCULE BRIDGES.

Bascule bridges may be either single leaf or double, but are usually double-leafed.

The most modern types are as follows :—

(*a*) Those in which the pivot is overhead and the balance weight either overhead or below the level of the bridge floor.

(*b*) Those where the pivot is situated below the level of the bridge floor and the counterweight likewise.

(*c*) Those in which there is no fixed axis, but the tail of the bridge is in the form of a circular segment, upon which the bridge rolls backward and forward.

PARTICULARS OF BRIDGES.

SWING BRIDGES.

Name of Bridge.	Span (Opening).		Width of Road.		Length of Long Arm.		Length of Short Arm.		Depth of Main Girders at Centre over Angles.		No. of Rollers.	Diam. of Roller Circle.		Weight (less Kentledge).
	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.		Ft.	Ins.	Tons.
Manchester Ship Canal, Old Quay, Runcorn .	120	0	20	0	139	0	96	10	28	0	60	22	11	650
Moore Lane .	120	0	25	0	140	0	98	0	27	8½	64	27	10½	790
Stagg Inn. . .	120	0	25	0	140	0	98	0	27	8	64	27	10½	790
Northwich Road .	120	0	36	0	148	0	100	0	30	0	60	38	9	1350
Knutsford Road .	120	0	36	0	118	0	100	0	30	0	60	38	9	1350
Barton Bridge .	90	0	25	0	111	0	81	0	26	0	64	27	10½	610
East India Docks, London (Road and Railway) .	80	0	11	6	95	0	30	0	16	8	Centre	Pivot.		480
King George V. Dock, London, (Skew Bridge, 22°) . .	108	0	20 and 2 footpaths.	8	150	0	50	0	25	0	Centre	Pivot.		950
Kilbowie (Railway) (Skew Bridge, 24°) . .	22	3	—		152 133	7 3	36	4	17	6	—	19	4	416
Dry Docks, Marseilles, France (one road, one rail-track, 3 main girders).	92	0	52 and 1 footway	6	126	0	77	5	11	6	Centre	Pivot		575

BASCULE BRIDGES

Name of Bridge.	Span (Opening.)	Width of Road.	Depth of Main Girder at Cr. over Angles.	Type.	Weight.
	Ft.	Ft. Ins.	Ft. Ins.		Tons.
Southern Railway, over River Swale, Kent .	62	33 0 centre of girders.	—	Rolling lift.	520 (plus counter-weight.)
Cart Navigation Trust, Inchinnan . . .	90	20 0	20 5	Rolling lift	207
King George V. Dock, London (double leaf) .	50	17 6 (and 2 footways)	29 0	Pivot.	381
Taylor Street, Chicago (Scherzer double leaf) .	295	28 0	40 0	Rolling lift.	—

TRANSPORTER BRIDGES.

Another type of bridge which avoids high approaches is the Transporter Bridge. The first bridge of this type to be erected in this country was completed in 1905 and provides access between Widnes and Runcorn. The central span is 1,000 ft. between centres of towers and the height from high-water level to the underside of the stiffening girders of the cable suspension bridge, which also carry the trolley from which the car is hung, is 82 ft. The car is 55 ft. long by 24 ft. 6 ins. wide.

Working Stresses,

PERMISSIBLE WORKING STRESSES.

The maximum permissible unit stresses per sq. in. should be less for the main girders, cross girders, and rail bearers than those allowed for simply supported spans; about 10 per cent. is a usual figure. (Refer to ' Working Stresses, p. 493.)

ROLLERS.

Rollers supporting swing bridges should generally be of solid forged steel or cast steel, with suitable gun-metal liners and collars and provision for adjustment at the ends of the rollers.

The rollers and surfaces of both upper and lower paths should form parts of conical surfaces with a common vertex at the centre.

The size of the rollers should bear some relation to the diameter of the roller circle, and in general will not exceed 2 to 3 ft. diameter.

BEARING PRESSURE ON ROLLERS.

The following safe bearing pressures in pounds per lineal inch on rollers are taken from the *Transactions of the American Society of Civil Engineers*, vol. xxxiii.

Description.	Rollers at rest.	Rollers in Motion.
Cast Iron	400 d	200 d
Rolled or Cast Steel (28–32 tons)	800 d	400 d
Axle Steel (35–38 tons)	—	500 d
Tool Steel (60 tons)	—	800 d
Tool Steel hardened	—	1900 d

where d is the diameter in inches.

The values given are for rollers and bearing surfaces of the same material; if they are of different materials, lower values should be used.

Sir W. Arrol & Co., in their Handbook, give the safe load in pounds per lineal inch on rollers, as 560 d, where d = the diameter of the roller, the mean diameter being taken for conical rollers, the rollers and plates being of cast iron, wrought iron, or steel; for rollers in motion one-half the above value to be taken. It is stated that the working stress is one-fifth the elastic limit for cast iron, and one-third the limit for wrought iron or steel, the elastic limit being reached in steel plates and steel rollers of 1 in. to 16 ins. diameter at 880 d.

Bearing Pressures on Rotating and Sliding Surfaces.

MAXIMUM BEARING VALUES FOR ROTATING AND SLIDING SURFACES IN POUNDS PER SQUARE INCH.

Bearings on which the Speed is slow and intermittent.	
Pivots for swing bridges, hardened tool steel on special phosphor-bronze	3,500
Trunnion bearings for bascule bridges, axle steel on phosphor-bronze	2,000
Wedges, cast-iron on bronze	600
Wedges, cast-iron on cast-iron or structural steel	500
Screws which transmit motion on projected area of thread	200
Ordinary Cases, Parts moving at Moderate Speeds.	
Hardened steel on hardened steel	2,000
Hardened steel on bronze	1,500
Tool steel (not hardened) on bronze	900
Structural steel on bronze	600
Cast-iron on structural steel	400
Cast-iron on cast-iron	400
On crosshead slides, speed not exceeding 600 ft. per minute	50

In order to prevent heating and seizing at higher speeds, the pressure on pivots or foot-step bearings for vertical shafts and journals should not exceed :—

$$\text{On pivots, } p = \frac{160000}{n\,d}\text{ ; on journals, } p = \frac{300000}{n\,d},$$

where

n = number of revolutions per minute; d = diameter of journal or pivot in inches.

For crank pins and similar joints with alternating motion, the limit bearing values given in the above formula may be doubled.

Ball and Roller Bearings.

(See Section XX., Part I.)

Centre Pivots.

The construction of centre pivots for swing bridges requires careful consideration. They should be of ample proportions, and the pressure per square inch should not exceed 2 to 3 tons per square inch, or excessive friction will ensue.

The pivots should preferably consist of four parts; the lower casting and upper castings of steel, with an upper revolving pin of tool steel with a convex surface, and the lower fixed cup of phosphor bronze with a concave surface of slightly larger radius than the upper revolving pin, to assist lubrication.

Pivots have been designed in which a 'Tabloid' was placed between the upper revolving pin and the lower stationary cup. The 'Tabloid' was of phosphor bronze with spherical upper and lower surfaces, the upper pin on the bridge and the lower cup also having spherical surfaces. This type of pivot is, however, only suitable for bridges in which the slewing power is applied at two or more diametrically opposite points, for where the slewing force is applied at only one point, a slight displacement of the bridge on the seating has been found to occur, resulting in considerable eccentricity.

Pressures on Quadrants and Tracks of Rolling Lift Bascule Bridges.

The following data refer to four bridges constructed during recent years :—

Clear Span.		Radius of Quadrant.		Rolling Load.	Width of Bearing.	Pressure on Track.	Materials in Quadrants and Tracks
Ft.	Ins.	Ft.	Ins.	Tons.	Inches.	Tons per Lineal Inch.	
58	3	18	3	414	26·0	15·92	Mild steel 28–32 tons.
100	0	21	0	631	38·0	16·61	Quad. plate mild steel.
100	0	21	0	516	22·0	23·45	Track plate cast steel.
150	0	28	0	2,922	50·25	58·15 average.	Track plate cast steel 35 tons. Quad. plate mild steel 36-40 tons.

(*Selected Paper No.* 8, *Inst.C.E.*)

Friction and Lubrication.

Rolling Friction.

Let

P = force required to propel roller (lbs.) ; W = load on roller (lbs.) ; R = radius of roller (ins.) ; h = depth of depression caused in track (ins.) ; K = a constant ; then,

$$P = \frac{WK}{R - h},$$

or, as h is usually very small compared with R,

$$P = \frac{WK}{R} \text{ (nearly)}.$$

The value of K varies from 0·007 to 0·015.

Some Experiments on Friction.

The following experiment, undertaken by Sir Wm. Arrol & Co., Ltd., is particularly applicable to the friction of roller rings in turntable swing bridges :—

Two beams, each 175½ ins. long by 5¾ ins. broad, each weighing 1570 lbs., were planed to a smooth surface and accurately levelled. One beam was laid dry upon the other and afterwards two rollers, 2½ in. diameter, were placed 7 ft. apart between them. The force required to start motion was obtained for different loads.

In experiments with eleven American turntable swing bridges, the friction showed a maximum of 7·94 lbs. per 1,000 lbs. weight of moving structure ; but one bridge showed as low as 3·53 lbs.

Further experiments in America on new swing bridges, before the gearing, etc., had had an opportunity to wear smooth, were as follows ; for bridges which have been in operation for some time, the coefficients given will be less :—

Load.	Force to Start Motion.		Coefficient of Friction.	Percentage of Load.	Remarks.
	Total.	Pounds Per Ton.			
lbs.	lbs.	lbs.			
1570	450–500	650–716	0·29–0·32	29–32	Sliding Friction.
2130	475–560	500–590	0·224–0·263	22–26	,,
1570	2·73	3·9	0·00174	0·17	Rolling Friction.
2240	5·44	5·44	0·00243	0·24	,,

For rim-bearing swing bridges, the late C. Shaler Smith found the total frictional resistance in the livering to vary from 0·004 to 0·008 of the load on the rollers. Messrs. Boller & Schumacher found the coefficient to be 0·0035 for the Thames River Bridge, and Theodore Cooper found this to be 0·0038 for the Second Avenue Bridge.

For centre bearing bridges, Mr. C. Shaler Smith found the frictional resistance at the circumference of the pivot to be 0·09 of the weight turned. Mr. Schneider found the coefficient of frictional resistance at the circumference of the pivot, on hardened steel and phosphor bronze discs, with the usual working pressure of about 3,000 lbs. per sq. in., to be 0·067 at the start and 0·015 to keep the bridge moving at uniform speed. For the total frictional resistance, including that of the shafts and gearing required for hand operations, the highest coefficients observed were 0·115 for starting and 0·08 for keeping the bridge in motion.

(*Proceedings American Society of Civil Engineers.*)

Experiments by Beauchamp Tower show that a steel shaft in a lubricated gun-metal bearing seizes at about 600 lbs. per sq. in. as for steady running, while in a dry bearing they seized at about 80 lbs. per sq. in.

Professor Goodman found that the seizing pressure increases as the viscosity of the oil increases.

Horse-Power Absorbed in Revolving Bearings.

Let

W = total load on bearing in lbs.; D = diameter of bearing in ins.; N = number of revolutions per minute; ϕ = coefficient of friction.

For Cylindrical Bearings:

$$\text{Work done per minute in ft.-lbs.} = \frac{\phi W \pi D N}{12};$$

$$\text{Horse-power absorbed} = \frac{W \pi D N \phi}{12 \times 33{,}000} = \frac{\phi W D N}{126{,}000}.$$

For Flat Pivots:

$$\text{Horse-power absorbed} = \frac{\phi W D N}{189{,}000}.$$

Lubrication.

Oil should always be led to the point of least pressure in a bearing.

Oils for heavy pressure should be thick, that they may resist being forced out.

Lubricants for Various Cases.

Nature of Pressure, etc.	Type of Lubricant.
Heavy pressure with slow speed	Graphite, soapstone, tallow, and other greases.
Heavy pressure with high speed . .	Sperm oil, castor oil, and heavy mineral oils.
Light pressure and high speed . . .	Sperm oil, refined petroleum, olive, rape, and cotton-seed oils.
Ordinary machines	Lard oil, heavy mineral and vegetable oils.

Oils act chemically in a varying degree on most metals, and care must be exercised in selecting the most suitable kind.

HYDRAULIC LIFTING AND SLEWING MACHINERY.

Power required for Lifting Swing Bridges.

This is a simple calculation based on the total weight of the bridge lifted and the friction due to moving parts, leathers, and so on.

Power required for Slewing Swing Bridges.

$P = 0{\cdot}015$, for rim-bearing; $P = 0{\cdot}15\ W \frac{r}{R}$, for centre-bearing; $H.P. = \frac{Pv}{550}$; $v = \frac{R\pi}{2t}$,

where

P = Force required to turn the bridge, acting in centre line of track; W = total moving weight, in lbs.; H.P. = horse power; r = radius of pivot; R = radius of centre line of track; v = maximum linear velocity in centre line of track in feet per second. t = time, in secs., in which maximum velocity must be obtained, or one-half the number of seconds required to open the bridge.

The foregoing formulæ generally give results in excess of the power actually needed under ordinary conditions, and practically cover all contingencies.

If the bridge has to be operated by hand-power, the gearing has to be arranged either for the number of men available or for the time required to turn and operate the bridge. In calculating the strength of the gearing, etc., the power of one man should be taken at 125 lbs., as this is about the force a strong man, with a foothold, can exert for a short time.

Setting and Wedging Blocks.

It is usually not advisable to allow a bridge to rest upon the pivot when undergoing stress due to moving loads.

In some bridges the centre pivot is also the lifting press, bearing blocks being provided at suitable points to sustain the bridge when in position. In this case, however, there is difficulty in preventing the ram from turning in the cylinder.

With a solid pivot, the hydraulic lifting rams are placed at the tail of the bridge, where, when the rams have lifted the bridge off the pivot, a pair of sliding bearing blocks are inserted and the lifting power then withdrawn. In this type four or more rollers are provided to take a portion of the weight of the bridge while slewing, and to keep the bridge 'upon an even keel.'

The two leaves of a double swing bridge are often linked together at the meeting point, the object being to prevent any perceptible difference in level between the extremities of the bridge leaves when a heavy load is transferred from one to the other.

CIRCULAR GIRDERS.

Bending Moments in Circular Girders.

(Equally spaced Supports.)

No. of Points of Support.	Reaction at Point of Support (lbs.).	Maximum Shear (lbs.).	B.M. Over Point of support (in.-lbs.).	B.M. Midway between Supports (in.-lbs.).	Angular Distance Point of support to Point of Maximum Torsion. °	′	Maximum Torsional Moments (in.-lbs.).
4	$\frac{W}{4}$	$\frac{W}{8}$	−0·03415 Wr.	+0·01762 Wr.	19	12	0·0053 Wr.
6	$\frac{W}{6}$	$\frac{W}{12}$	−0·01482 Wr.	+0·00751 Wr.	12	44	0·00151 Wr.
8	$\frac{W}{8}$	$\frac{W}{16}$	−0·00827 Wr.	+0·00416 Wr.	9	33	0·00063 Wr.
12	$\frac{W}{12}$	$\frac{W}{24}$	−0·00365 Wr.	+0·00190 Wr.	6	21	0·000185 Wr.

W = Total uniform load supported by girder in lbs.; r = radius of girder in ins.

SECTION XIV

PART V

ARCH BRIDGES.

Types and Stresses.

ARCHED RIBS.

The arched rib or roadway girder may be in either of the following forms:—

(*a*) Rigidly fixed at each end.
(*b*) Hinged at each end.
(*c*) Hinged at centre and at each end.
(*d*) Hinged near one abutment.

STRESSES IN ARCHES.

The arch may be looked upon as an inverted suspension bridge, the links of the suspension bridge being in tension and those in the arch being in compression, but the latter would be in unstable equilibrium under any loading, and so in practice the arch is made capable of resisting bending stresses in addition to direct compression.

HORIZONTAL THRUST.

When once the horizontal thrust and the line of the resultant thrust or any given arch has been found, it is a simple matter to obtain the stresses.

The horizontal thrust and the line of the resultant thrust can be determined readily for parabolic arches of uniform section loaded with uniformly distributed load, or with concentrated loads, and the same rules can be taken as applying to flat segmental arches.

LINEAR ARCH AND LINE OF PRESSURE.

A linear arch is one which is theoretically supposed to be subject to direct compressive stress only and incapable of resisting flexure.

The line of pressure is the theoretical line, the ordinates of which vary as the bending moments for the load which the arch supports. It is synonymous with the curve of equilibrium.

Eddy's Theorem is thus:

'The Bending Moment at any point in an arch is equal to the product of the Horizonta Thrust into the vertical intercept between the centre line or neutral axis of the arch and the line of pressure.'

It is almost universally the practice with steel arches to hinge them at the abutments, and often at the crown also. By this means the alterations in form of the arch due to the passage of the load and variation of temperature more readily adapt themselves to the varying conditions. Moreover, the resulting thrusts pass through these points and the axis of the arch, or approximately so, and the bending stresses are thus reduced and can be determined with greater accuracy.

STRESSES IN PARABOLIC ARCH, HINGED AT THE ABUTMENTS.

Let

l = length of span of arch; v = versine or rise; w = load per unit of length, T = horizontal thrust at the crown and abutments; C = compressive stress at any point x from the centre, then

$$T = \frac{w\,l^2}{8v}$$

This expression represents the horizontal component of the pressure at any point in the arch

$$O = \sqrt{\left(\frac{wl^2}{8v}\right)^2 + (wx)^2}.$$

When loaded with a single weight W at the centre of the arch

$$T = \frac{Wl}{4v}.$$

If W rests at a point x from centre of arch, then,

$$T = \frac{W(l - 2x)}{4v}.$$

When loaded with a uniform load over half the span,

$$T = \frac{wl^2}{16v}.$$

Stresses in Three-Pinned Arch.

When loaded with a single weight W (see fig. 1),

$$T = \frac{W\,l}{4v}.$$

If the lines of pressure be inclined at an angle ϕ to the horizontal the direct compressive stress, s, at any section in the rib is

$$s = \frac{W\,l}{4v} \sec\phi \cos\delta,$$

where

δ = angle which the tangent at the section under consideration makes with the line of pressure.

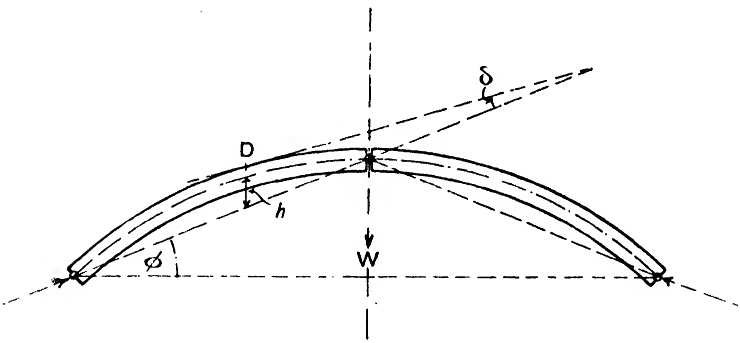


Fig. 1.—Three-Pinned Arch.

The bending moment at section D

$$= \frac{W}{4v} \times h,$$

(h being the vertical distance between the line of pressure and the neutral axis of the rib at the given section).

For other loadings the same rules apply, the line of pressure always passing through the pin joints.

STRESSES IN ARCHES WITH FIXED ENDS.

In the case of a parabolic arch with fixed ends, of uniform cross section throughout, the resultant thrust lines due to concentrated loads are readily determined from the following relations, which can also be taken as applying to flat segmental arches which can be considered as parabolic.

$$AP = \frac{2}{15} \times \frac{C - 5b}{C - b} \times v$$

$$MO = \frac{6}{5} \times v$$

$$CQ = \frac{2}{15} \times \frac{C + 5b}{C + b} \times v$$

where C is half the span, b the distance of the concentrated load from the centre of the span v the rise or versin of the centre line. AP (fig. 2) is measured below the springing line when the value is negative.

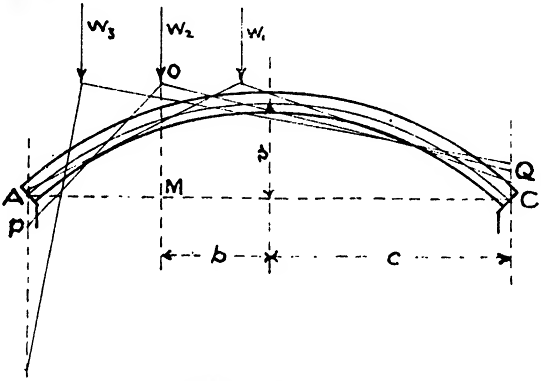


FIG. 2.—Fixed End Arch.

The horizontal component T of the thrust due to a concentrated load W is $15W(C^2 - b^2)^2 \div 32vC^3$, and the reactions R_1 and R_2 are obtained from the relations :—

$$\frac{R_1}{T} = \frac{MO - AP}{C - b}$$

$$\frac{R_2}{T} = \frac{MO - CQ}{C + b}$$

Any given load produces a maximum bending moment n the arched rib when $b = 0{\cdot}45C$, and to obtain the maximum bending moment from a system of concentrated loads these should be placed on half the span with this position as centre.

If the dead load is uniformly distributed and equal to w lbs. per ft. run, the resultant thrust line does not depart sensibly from the centre line of the arched rib, and the horizontal component of the thrust is equal to $wC^2 \div 2v$.

The bending moment due to any load is equal to the horizontal component of the thrust multiplied by the vertical intercept, at the section considered, between the resultant thrust line and the centre line of the arch, and the effect of a system of loads can be obtained by combining the bending moments due to the individual loads.

If T is the horizontal component of the thrust due to the uniformly distributed dead load, T_1, T_2 and T_3, the horizontal components of the thrust due to three concentrated loads W_1, W_2, and W_3, and y_1, y_2 and y_3, the vertical intercepts between the resultant thrust lines due to these loads, and the centre line of the arch at any vertical section, then the total bending moment at that section $= T_1y_1 + T_2y_2 + T_3y_3$, and the distance of the resultant thrust line from the centre line of the arch at the section considered $= \dfrac{T_1y_1 + T_2y_2 + T_3y_3}{T + T_1 + T_2 + T_3}$

Values of y below the centre line are negative quantities.

Combined Stresses in Arched Ribs.

In any arch where the line of pressure does not coincide with the neutral axis of the rib, there will be the following forces at any cross section :—

(1) A maximum compressive stress

$$= \frac{Q}{A} + \frac{M}{Z_c}.$$

(2) A maximum tensile stress

$$= \frac{M}{Z_t} - \frac{Q}{A}$$

(3) A mean shear stress over the section

$$= \frac{S}{A},$$

where

Q = the thrust; S = the shear (obtained by resolving the horizontal thrust along (or tangential) and perpendicular to the neutral axis of the section under consideration); A = the area of the cross section of the rib; M = the bending moment obtained by applying Eddy's Theorem; Z_c and Z_t = the moduli of section in compression and tension respectively.

Stresses in Braced Arches.

Having found the direction and magnitude of the abutment reactions for any loading, the stresses can be found in a similar manner, those in the bracing being conveniently determined by Ritter's Method of Moments, described under 'Lattice Girders,' page 519.

Effects of Temperature Variation.

A rise in temperature causes an increased thrust in two-pinned and rigid arches, but theoretically causes no such increase in three-pinned arches, although practically some slight difference may be occasioned by any stiffness of the pin joints.

For a short research on Temperature Stress, see *The Theory and Design of Structures*, by Ewart S. Andrews, B.Sc., M.I.C.E.

Ratio of Rise to Span.

This is usually taken as from 1 : 5 to 1 : 10, but varies greatly according to local conditions of headroom.

The three-pinned arch over the river Exe—a road-bridge carrying tram-cars, erected in 1905, has a ratio of rise to span of 1 : 13·2, and is probably the flattest steel arch in existence in this country.

SECTION XIV

PART VI

SUSPENSION BRIDGES.

Types and Stresses.

Advantages and Disadvantages of Suspension Bridges.

The principal advantage is their cheapness, and in large spans the advantage of the suspension bridge is so great that bridges of this type of 800 or 900 ft. span can be constructed at much less cost per foot run than girder bridges of half this span.

There are many disadvantages attending suspension bridges, the chief being lack of stiffness and rigidity, both vertically and laterally, which makes them unsuitable for carrying railways unless elaborate means are taken for stiffening them.

A suspension bridge constructed on 'Bridge's System,' in which the suspension rods are inclined, is much stiffer vertically than one in which the rods are vertical.

Types of Stiffened Suspension Bridges.

In order to lessen the changes of shape and oscillations set up by a moving load, the roadway girders may be constructed of rigid form and braced laterally. The principal varieties of this and other types of stiffened suspension bridges are:

(a) Roadway stiffening girders hinged at each end and at centre.
(b) Roadway stiffening girders hinged at each end.
(c) Roadway stiffening girders hinged in either of the above ways and bridge consisting of two cables at each side stiffened by bracing.

Examples of some of the above types are as follows:—

Types (a) and (b).—Williamsburg Bridge, over East River, New York, built 1903, span 1,600 ft. Double line of railway, four lines of tramways, two roadways and two footways and cycle tracks.

Manhattan Bridge over East River, New York, built 1909, span 1,470 ft., four lines of tramway, one roadway, and two footways.

Type (c).—The Hudson River Bridge, New York, clear span 2,880 ft., four lines of railway, four lines of tramway, two roadways, and two footways. A further example of this type is the Tower Bridge, London, in which the side spans are of this type.

An example of a bridge constructed with inclined and vertical suspenders is the Brooklyn Bridge, New York, built in 1884, span 1,595 feet, which carries a double line of tramway, two roadways, and one footway.

Stresses in Cable Uniformly Loaded.

The shape which a loaded cable takes up is the same as the link polygon for the given load system, drawn from one support to the other, with a polar distance equal to the horizontal component H of the pull or tension in the cable.

If

d = the dip of the cable; L = span; W = weight carried by the cable, then,

$$H = \frac{WL}{8d}.$$

A = area of the cable; f = safe tensile stress,

$$fA = \frac{W}{2}\sqrt{\left(1 + \frac{L^2}{16d^2}\right)}.$$

Stresses in Anchor Cables.

There are two chief methods by which the cables of suspension bridges are carried over the piers —

(1) Main and anchor cables passing continuously over rollers fixed to the top of the piers,

(2) Main and anchor cables passing continuously over saddles mounted on rollers on the tops of the piers.

Let

ϕ = angle which bridge cable makes with a vertical at point of support; δ = angle which anchor cable makes with a vertical at point of support; T = tension in bridge cable; T_1 = tension in bridge anchor cable.

Case (1).—Tension in anchor cable is equal to tension in bridge cable, whatever the inclinations,

$$= T_1 = T;$$

Horizontal force acting on piers $= H\left(1 - \dfrac{\sin\delta}{\sin\phi}\right);$

Vertical pressure on piers $= \dfrac{W}{2}\left(1 + \dfrac{\cos\delta}{\cos\phi}\right).$

Case (2).—Tensions in anchor and bridge cables not always the same, but there is no horizontal force on the piers,

$$T_1 = \frac{T\sin\phi}{\sin\delta};$$

Vertical pressure on piers $= \dfrac{W}{2}(1 + \tan\phi\cot\delta).$

Stresses in Suspension Bridges with Pin-Jointed Stiffening Girders. Type (*a*).

Uniform Rolling Load.

With a uniform load over the whole span there is no bending moment on the girders, the whole behaving as an unstiffened suspension bridge.

When the load covers half the span there will be a reverse or negative bending moment on the unloaded girder equal to the positive bending moment on the loaded girder.

$$\text{Maximum negative B.M.} = \frac{wL^2}{64};$$

and occurs at $\frac{1}{4}$-span.

The maximum positive bending moment at any point distant z from the centre pin, the head of the load being distant $x = \dfrac{l^2}{2l + z}$ from ditto

$$= \frac{wz\,(l^2 - z^2)}{2\,(l + z)}.$$

. The absolute maximum positive bending moment = $0\cdot0753\,wl^2$ (or $\dfrac{wL^2}{53}$ approximately), and occurs when $z = 0\cdot53l$, the head of the load being distant x from the centre pin.

Isolated Rolling Load.

With a concentrated rolling load passing over the girders :—

$$\text{Maximum negative B.M.} = \frac{WL}{16}.$$

and occurs at $\frac{1}{4}$ span.

When the load is on one of the girders the bending moment is a maximum at any point when the load just reaches it.

Maximum B.M. at any point distant y from the centre pin

$$= \frac{Wy}{2L^2}(L^2 - 4y^2).$$

Absolute maximum positive B.M. = $0\cdot096$ WL, and occurs when load is $0\cdot289$L from the centre pin.

STRESSES WITH STIFFENING GIRDERS HINGED AT ENDS ONLY. TYPE (b).

In this case the stiffening girders will have to bear considerable stresses due to the increase in dip due to increase in temperature, which may amount to as much as half the safe stresses.

With a uniform rolling load passing over the span—

$$\text{Absolute maximum bending moment} = \frac{wL^2}{54},$$

and occurs at ⅙-span, when the load covers ⅔ of span.

The head of the load is always a point of contraflexure of the girder.

Assuming that the positive bending moment is a maximum at the mid point, then:

$$\text{Maximum bending moment in the length covered by the load} = \frac{wx^2 (L - x)}{8L};$$

when head of load is distant x from the nearest pin.

In the foregoing:—

w = load in tons per foot run; W = concentrated rolling load in tons; L = length of span in feet; l = length of half span in feet.

CHANGES OF TEMPERATURE.

In England the observed expansion of steel structures appears to be about $\frac{1}{2400}$ of their original length from their coldest temperatures. This corresponds to a range of temperatures of 60° F. In suspension bridges, with movable saddles at the tops of the piers, the augmented dip of the central cable is due to increase in length of the anchor cables and consequent movement of the saddles as well as the increase in length of the central cable itself.

Let

l = the half span of central cable; D = central dip; D_1 = augmented dip due to rise in temperature of 60° F.; L = length of semi-parabola measured along curve; then by the properties of the parabola,

$$L = \sqrt{l^2 + \frac{4}{3} D^2};$$

and

$$D = 0 \cdot 866 \sqrt{L^2 - l^2};$$

therefore

$$D_1 = 0 \cdot 866 \sqrt{\left(L + \frac{L}{2400}\right)^2 - l^2}.$$

To this must be added the increased dip due to inward movement of the saddles.

ECONOMIC PROPORTIONS.

Economy in the weight of a cable can be effected by adopting a liberal depth or dip, but it is impossible in practice to increase the depth beyond a certain amount, because the greater the dip the greater will be the distortion and oscillation caused by a moving load.

In most of the existing examples of suspension bridges of various types, the dip does not exceed $\frac{1}{10}$ of the span, and in many cases $\frac{1}{12}$ to $\frac{1}{15}$ has been adopted.

SELF-ANCHORED SUSPENSION BRIDGE.

The new Chelsea Bridge provides an example, claimed to be the first of its kind in this country, of a self-anchored suspension bridge. The stiffening girders, in addition to distributing the effect of local live load over all hangers of the loaded span, also takes the horizontal component of the anchorage pull. These girders are hinged near each tower and, for erection purposes at the centre, but this last hinge was afterwards riveted up. The main dimensions, decided principally by site conditions, are as follows: centre span 352 ft.; side spans each 173 ft.; height of cables at centre 13 ft. above road level; height of towers about 52 ft. above road level; depth of dual stiffening girders 8 ft. 10 ins. over flange angles; width of roadway between towers 40 ft.; width of footpaths outside towers each 12 ft. The towers are hinged at the bases, and independent with no portal bracing.

SECTION XV

HIGHWAY ENGINEERING

ADMINISTRATION — HIGHWAY STATISTICS — FEATURES AND COSTS OF TYPICAL WORKS — COSTS OF CARRIAGE-WAY CONSTRUCTION — COSTS OF ROAD MAINTENANCE — RESEARCH AND EXPERIMENT — ROAD MATERIALS — THE VEHICLE AND THE ROAD — FUNDAMENTALS OF ROAD PLANNING—DESIGN IN PLAN AND PROFILE—CARRIAGE-WAY CONSTRUCTION—SPECIFICATIONS, DESCRIPTIONS, COSTS — SURFACE TREATMENT — FOOTPATH CONSTRUCTION — ROAD MACHINERY — ROADSIDE TREES — ROAD TUNNELS—RECENT LITERATURE

(pp. 543–574)

(Contributed and Revised by Reginald Ryves, M.Cons.E.)

(*Author of 'The King's Highway; or the Nature, Purpose, and Development of Roads and Road Systems'; 'Preliminary Studies in Bridge Design'; 'Road Safety: the Speed Factor as Affecting the Frequency and Severity of Road Accidents'; 'The Channel Tunnel Project.'*)

SECTION XV

HIGHWAY ENGINEERING

(Contributed by Reginald Ryves, M.Cons.E.)

For the purpose of maintaining the year-book character of this section a considerable portion of the older matter, mostly relating to fundamentals of highway engineering, has been omitted. As in the case of the 1932 edition, in which administrative provisions were set forth at considerable length, the 1937 edition, or an earlier one, may serve as a volume of reference in regard to the portions no longer to be reproduced annually.

REFERENCES TO THE 1932 EDITION.

The Local Government Act, 1929: England and Wales, p. 712; Scotland, p. 714.
Under that Act (England and Wales): powers delegated, powers claimed, appeals, p. 713.
Ministry of Transport, Roads Department, p. 714.
The Roads Improvement Act, 1925, p. 715.
The Public Health Act, 1925, p. 716.
The Road Traffic Act, 1930, p. 762.
Traffic Statistics, p. 761.
Working Costs Data: steam wagons, electric vehicles, petrol vehicles, pp. 759–760.

REFERENCES TO THE 1937 EDITION.

The Bridges Act, 1929, p. 619.
Legal Points for Road Engineers, p. 620.
Road Construction, pp. 625–632.
Resistance to Traction, pp. 633–634.
Tractive Effort on Different Pavements, p. 634.
Reinforced Concrete Roads; Principles of Design, p. 642.

ADMINISTRATION: GREAT BRITAIN.

Trunk Roads Act, 1936.—The Act makes the Minister of Transport the highway authority for some 4,500 miles of roads, substantially and with some additions, the trunk road system in being, 1936. The Act came into force on April 1, 1937, in England and Wales, and on May 16 in Scotland. The Trunk Roads (Delegation of Powers) Order, 1937, imposed certain functions upon local authorities, in respect of these roads, for a period of two years, in order to secure continuity of administration.

The Trunk Roads Act, 1946.—Provided for the addition, April 1946, of 3,685 miles of existing roads to the previous mileage of 4,500, raising the percentage mileages; England and Wales, from 2·13 to 4·03; Scotland, from 4·6 to 7·5.

The Special Roads Bill.—Presented to the House of Commons, November 1948, it is an enabling Bill, providing powers for the construction of motorways and roads for the exclusive use of cyclists, or pedestrians. Special roads may be trunk, classified or unclassified, and may include parts of existing roads. Orders under the Trunk Roads Act, on which procedure had commenced, or which are made within one year of the Bill becoming law, will take effect as if they were special road schemes.

Restriction of Ribbon Development Acts.—A Ministry of Transport Circular of October 1948 explains how the Town and Country Planning Act of 1947 repeals certain of the provisions of the Restriction of Ribbon Development Act of 1935 and repeals entirely that of 1943.

HIGHWAY LAW.

Speed Limit.—The speed limit in 'lighted areas,' these being mostly built-up areas, is 30 m.p.h. Evidence given on behalf of the Pedestrians' Association before the House of Lords Select Committee on Road Accidents, included, that 30 m.p.h. is too high a speed; that 25 m.p.h. would give a much better margin of safety, and that experiments in America pointed to 23½ m.p.h. as the most effective speed with respect to traffic flow in built-up areas.

HIGHWAY STATISTICS.

Mileages of Public Roads in Great Britain, 1937–38.

(M., Metropolitan Cities and Boroughs; C., County Boroughs.)

England.

***Trunk Roads.*—Urban, 762, rural, 2,092. Total, 2,854.**

***Class I.*—County: urban, 3,941, rural, 9,850; M., 292; C., 1,508.** Total, 15,591.

***Class II.*—County: urban, 2,446, rural, 8,555; M., 125; C. 725. Total, 11,851.**

Unclassified.—County: urban, 955, rural, 73,364. 'District roads' in non-county boroughs and urban districts, 17,425. M., 1933; C., 10,864. Total, 104,541.

***All Classes.*—Urban, 40,976, rural, 93,861. Total, 134,837.**

Wales.

***Trunk Roads.*—Urban, 97, rural, 320. Total, 417.**

***Class I.*—County: urban, 422, rural, 1,264; C., 79. Total, 1,765.**

***Class II.*—County: urban, 248, rural, 936; C., 35. Total, 1,219.**

Unclassified.—County: urban, 163, rural, 12,934; 'district roads,' 1,882; C. 540. Total, 15,519.

***All Classes.*—Urban, 3,466, rural, 15,454. Total, 18,920.**

Scotland (B., the 24 large burghs, population exceeding 20,000; b., other burghs.

***Trunk Roads.*—All under 'counties,' 1,188.**

***Class I.*—County, 5,124; B., 320; Total, 5,444.**

***Class II.*—County, 3,866; B., 101. Total, 3,967.**

Unclassified.—County, 12,908; B., 1,495; b., 871. Total, 15,274.

***All Classes.*—Urban, 2,787, rural, 23,086. Total, 25,873.**

Great Britain.

Trunk Roads, 4,459. Classes: I, 22,800, with trunk roads, 15·2 p.c.; II, 17,037, or 9·5 p.c.; Unclassified, 135,334 or 75·3 p.c. Total, 179,630, an increase of 726 miles in the twelve months.

Northern Ireland.

Class I., 1,273; Class II., 1,245; Unclassified, 10,479. Total, 12,997.

Class III. Roads.—In 1946, the Ministry of Transport called for returns from Highway authorities; lists of such of the unclassified roads as might be designated as in Class III., qualifying for a grant towards costs of maintenance and repair. The new scale of grants for the three classes is: Class I., 75, Class II., 60, and Class III., 50, per cent.

As finally approved by the Ministry, the mileages of Class III. roads, in round figures, are: English Counties, 35,500; Welsh Counties, 6,000; Scottish Counties, 5,900; aggregate of mileages exceeding 30, in County Boroughs of England, and Wales (Glamorgan only), 866; aggregate of mileages, exceeding 10, the 6 large Burghs of Scotland, 209.

Road Mileages in Relation to Area and Population.

Road Mileages per Square Mile.

England and Wales.—2·633. Classes: I., 0·295; II., 0·224; Unclassified, 2·115.

Scotland.—0·869. Classes: I., 0·210; II, 0·133; Unclassified, 0·513.

Great Britain.—2·039. Trunk, 0·051. Classes: I., 0·259; II., 0·193; Unclassified, 1·536.

***Canada*, by Provinces: Quebec, 0·047; improved roads, 0·02, including gravelled, 0·016. Ontario, 0·18; improved roads, gravel, 0·10, hard surface, 0·02. Nova Scotia, 0·87, including graded and surfaced, nearly all gravel, 0·225; graded only, 0·202, unimproved, 0·441. Prince Edward Island, 1·71. Saskatchewan, 0·6. Manitoba, 0·084; including improved roads, more than half the mileage gravelled, 0·02. British Columbia, 0·051; including gravelled, 0·014; earthen, 0·036. Alberta, 0·24; including improved roads, 0·066 (gravelled, about 0·004, earthen, 0·062); unimproved, 0·176. Canada, surfaced roads, 0·112.**

***Africa*, British possessions and protectorates: Northern Rhodesia, nearly all earthen, 0·025. Southern Rhodesia, 0·042. Kenya, 0·043; comprising main and district roads, 0·021, fair-weather roads and tracks, 0·022. Tanganyika, 0·046; comprising main and township roads, 0·007; district roads: grade (A), 0·002, grade (B), 0·026; local roads and tracks, 0·011 Uganda, 0·039; comprising first-class, heavily metalled, 0·011; second-class, gravelled, 0·002; third-class, 0·025.**

Australasia.—Victoria, the system includes State highways, 0·017; developmental roads, 0·05. South Australia: main roads, 0·017; all metalled and gravelled roads, 0·036; remade roads, 0·121; all roads, 0·157. New Zealand, 0·47, including 0·32 paved or surfaced.

France.—State roads, 0·118; departmental, 0·826; communal, 0·962; all classes, 1·906.

[To reduce to km. per sq. km., multiply by $\frac{87}{140}$, or 0·6214.]

Belgium.—In 1934: state roads (*s*), 0·465; provincial roads (*p*), 0·083; local roads (*l*), 1·818. All paved and metalled roads (*t*), 2·366. In km. per sq. km.: (*s*), 0·287; (*p*) 0·052; (*l*), 1·130; (*t*), 1·417.

Persons per Mile of Road: Counties (based on mileages, March 31, 1927, and census, 1921).—*Eleven*, below 61; Radnor 23, Montgomery 32, Cardigan 44, Merioneth 48, Brecon 54, Hereford 56, Devon 57, Westmoreland 57, Rutland 58, Pembroke 60, Norfolk 60. *Sixteen*, below 90; Lincolnshire, Holland, 67; Anglesey 70, Shropshire 70, Lincolnshire, Kesteven, 71; Cornwall 73, East Riding of Yorkshire 76, Carmarthen 77, Lincolnshire, Lindsey, 78; Oxfordshire 79, Denbigh 81, Suffolk, West, 83; Somerset 85, Cumberland 86, Suffolk, East, 87; Wiltshire 87, North Riding of Yorkshire 89. *Twelve*, below 131; Gloucestershire 93, Huntingdon 97, Dorset 98, Carnarvon 101, Hampshire 105, Flint 107, Northamptonshire 114, Berkshire 118, Isle of Ely 120, Leicestershire 124, Buckinghamshire 130, Northumberland 130. *Eight*, below 200; Worcestershire 139, Sussex, West, 143; Warwickshire 145, Sussex, East, 151; Cambridgeshire 153, Huntingdonshire 194, Bedfordshire 199, Nottinghamshire 199. *Nine*, 200 to 300; Staffordshire 201, Derbyshire 209, Monmouthshire 217, Cheshire 223, Soke of Peterborough 223, Isle of Wight 230, Kent 230, Essex 241, Yorkshire West Riding, 260. *Five*, over 300; Surrey 324, Lancashire 384, Durham 424, Glamorgan 450, Middlesex 927.

Great Britain, 248; Canada, 21; U.S.A., 196; Germany, 292; France, 106; Italy: national roads, 3,393; provincial, 1,696; other, 565; all roads, 377.

Australia, 13·8; New South Wales, 22·2; Victoria, 17·3; Queensland, 8·2; South Australia, 9·7; Western Australia, 6·0; Tasmania, 21·1; Nyasaland Protectorate, 475; Northern Rhodesia, 196; Southern Rhodesia, 183.

Mileages per 10,000 *persons.*—See *The Surveyor*, April 21 and 28, 1933, for other figures and discussion.

Examples.—Towns having populations 98,000 to 991,300: Birmingham, 7·24; Liverpool, 7·68; Manchester, 8·71; Sheffield, 8·71; Leeds, 12·3; Halifax, 22·4; Nottingham, 8·3; Tottenham, 4·93; West Ham, 4·33. Population, 40,000 to 98,000: Port Talbot, 16·0; Merthyr Tydfil, 17·5; Bournemouth, 15·3; Lancaster, 11·2; Dudley, 9·34; Grimsby, 7·0; Gellygaer, 5·37. Population 20,000 to 40,000: Scunthorpe and Frodingham, 16·5; Coulsdon and Purley, 15·8; Erith, also Chorley, 10·4; Wednesbury, 8·3; Gravesend, 8·24. Metropolitan Cities and Boroughs: Finsbury, 8·51; Woolwich, Greenwich, Lewisham, nearly 7·3; Kensington, 5·36; Poplar, 4·05; Bethnal Green, 3·7.

Miles of Roads per Square Mile.—Birmingham, 9·9; Liverpool, 17·3; Manchester, 19·5; Sheffield, 8·4; Leeds, 10·4.

ROAD SYSTEMS OVERSEAS.

Australia.—(i) December 31, 1934: Population, 6,691,000; area, 2,450,021 sq. miles; road mileage, 485,931; number of registered motor vehicles, 720,509, or 107·7 per 1,000 of population; per mile of road, persons, 13·8; vehicles, 1·5. (ii) June 30, 1935: State highways, 9,802 (none so classed in South Australia or Western Australia); other main roads, 36,500 (none so classed in Tasmania); developmental roads, 12,907 (none so classed in South Australia or Tasmania); total, 59,209 miles of roads so classified, or 12·18 per cent. of the total road mileage. Expenditure for 1934–35: on classified roads, £8,846,800; on unclassified roads, £6,025,394.

Union of South Africa.—It was anticipated that 5,396 miles of national roads would have been completed by the end of the fiscal year 1940–41, at a cost of about £11,000,000, including about £8,500,000 construction cost, and £2,170,000 interest and redemption on loans.

Features and Costs of Construction.

Many works classed as road-making involve alterations to the fences, ditches, and roadbeds of existing roads or tracks. Costs are not, therefore, generally comparable, the earlier work being, in some cases, largely to the good, in others involving expense additional to that which would have been incurred on virgin ground. In Great Britain, the distinguishing term 'new construction,' or some equivalent, is usually employed in respect of works on a new alignment.

Western Australia.—Works costing £666,431 (1934–35) comprised: cleared, 848 miles; formed, 897 miles; gravelled 339 miles; metalled, 15 miles; sanded, 10 miles; reconditioned, 173 miles; side drains formed, 213 drain miles; bituminous treatment, 95 miles.

Similar work, on comparable mileages, has been continued year by year.

Western Highlands of Scotland.—Programme of reconstruction or improvement on 1,500 miles of roads, including bridge works, £4,500,000, estimate.

Crawley By-pass.—About 2½ miles. Estimate £133,400, for a 100-ft. road, with two 20-ft. carriageways, two 6-ft. cycleways and two 5-ft. footways; and including a bridge over the railway. Opened July 1939. Revised estimate about £180,000.

Winchester By-pass.—Opened February, 1940; nearly 7 miles; cost, £420,000. Normal width, 72 ft.; carriageways, 20 ft., divided by a 10-ft. middle strip; three fly-overs, and 7 other bridges, of spans 26, 42, 44, 45, 50 (two) and 82 ft. Total of excavation, about 600,000 cubic yds.; of embankment, slightly less.

Gloucester By-pass.—A section 1¾ miles long; width, 120 ft.; two 22-ft. carriageways separated by a 26-ft. strip; cycleways (not yet constructed) and footways; allows for future widening of the carriageways to 30 ft.; ultimate width, including service roads, 172 ft.; building lines 20 ft. behind the service roads. Curves 2,000 ft. or more radii; excepting one of 500 ft.; all super-elevated; traffic roundabouts (one in this section) at crossings of main roads; trees spaced 60 ft. apart in both verges between the footways and sites of cycleways, and a line of trees in the middle reservation. Cost about £56,000, of which £45,500 is for road works and £10,500 for land; cost of roundabout, £2,360; rates, per cubic yd.: bulk excavation, 3*s.* 11*d.*; extra for filling, 1*s.* 4*d.*; concrete under kerb, 27*s.* 2*d.*; sewer excavation: not exceeding 5 ft. deep, 5*s.* 1½*d.*, between 5 ft. and 12 ft. deep, 6*s.* 10*d.*; per sq. yd.: 3 ins. consolidated clinker or gravel, 0*s.* 7*d.*; 8 ins. concrete slab, 6*s.* 0¾*d.*; preparing and sowing verges, 0*s.* 5*d.*; per lineal yd.: 12 ins. by 6 ins. precast concrete kerb, 3*s.* 11*d.*; concrete pipes, from 4*s.* 10*d.*, for 9-in., to 22*s.* 11*d.*, for 24-ins.; wire mesh and oak post fencings, 3*s.* 9*d.*; apiece: precast manholes, 7 ft. to 12 ft. deep, £9 to £14; trees, stakes, and wire guards, 9*s.* 11*d.*

Coventry By-pass.—Completed 1940. Length, 6 miles 7 furlongs; cost, including service roads, bridges and incidental works, £350,000; comprising, main road, £259,000; land, £30,000; service roads, £42,000 (recoupment of 30*s.* per foot of frontage); Tile Hill Lane section, £19,000; width, 150 ft.; least radius of a curve, 1,500 ft.; concrete carriageways, chiefly; some of stone pitchings and a wearing course of tar-bitumen macadam; cycleways of tarred limestone.

Schemes in Abeyance.—Schemes estimated to cost over £10,000 and not to be put in hand unless to meet war requirements or urgent needs—as ordered May 1, 1940—totalled £3,133,000 estimated cost.

Haifa-Baghdad Road.—Described in the *Journal Inst. C.E.* January and October issues, 1940, the former containing a paper, by Lieut.-Colonel R. Briggs, describing the works in the sections, Irbid to the Lava Belt, 41 miles; through the Lava Belt, 106 miles, and thence to the Transjordan-Iraq frontier, 56 miles. The length of the road from Haifa to the Iraq frontier is 275 miles and thence to Baghdad, 350 miles; the latter portion including 50 miles of previously existing road.

Rangoon-Chungking Road.—Known as the Burma Road and nearly 200 miles in length, it includes a section of 350 miles, from Shaknoan to the China-Burma frontier, having a minimum formation width of 18 ft., increased to 24 ft. in favourable ground, and a road crust, initially 10 ft. wide, having a base course, 4½ ins. thick, of packed stone and a wearing course, 3 ins. thick, of gravel or broken stone.

Costs of Carriageway Construction.

(Per mile, and maintenance per mile per annum.)

Victoria.—Forming an asphaltic (tar-bound gravel) wearing course on an earthen road, £540, its maintenance cost £50; a gravel crust, £1,000, maintenance, £30; a higher type (for economic comparison, 4 per cent. loan repayment), £4,000, maintenance, £20; cement concrete, 9-7-9 ins., 20 ft. wide, £11,800; crushed rock (presumably crusher-run broken stone), 20 ft. wide, 7 ins. thick, £1,570; 8 ins., £2,000; mixed-in-place, tar-bound gravel wearing course, loose depth, 2½ to 3 ins.,; the tar applied at the second of the three stages of mixing; a sealing coat of bitumen, covered with fine material: cost, £680 per mile for 18-ft. width.

New South Wales.—Analysis: In an expenditure of £247,630; percentages of cost of construction—formation, 34·10; culverts, 7·14; paving, 34·34; bridges, 9·99; fencing, 1·54; maintenance, during construction, 1·71; to the end of the year, 1·09; camp, water-supplies, sundries, 3·88; survey, designing and supervision, 6·21.

Newcastle-Tynemouth Road.—Nearly 4·7 miles; about £43,000 per mile, or, not including the cost of four large bridges, about £31,700 per mile. Half the 102 ft. width made; 30-ft. carriageway; reinforced concrete 7½ ins. at sides, 10½ ins. at centre, on flat bed of ashes; top 2 ins. blue whinstone chippings; single reinforcement.

Reconstruction.—Cardiganshire, including foundations and surfacing with tar macadam, £3,167 per mile. Lincolnshire (Holland), £4,000. Dorset, Wimborne, and Cranbourne R.D.C., £2,791. A moorland road, £5,000.

Costs of Road Maintenance, 1936–37.

(Per mile, per annum.)

Maintenance, Repair, and Minor Improvement.

(Classes I, II. and III.)

'Class III.' here means the then wholly unclassified roads. See p. 541.

Northamptonshire.—1942–43. Ordinary Maintenance.—Classified roads, £255. Unclassified roads, £82, comparing with £451 and £129 in 1938–39.

Holland.—1944–45. Trunk road A16, £195; Class I., £247; Class II., £158; Unclassified, £82; average, £102.

Leicestershire. 1945–46. Trunk (42·5 miles), £333; Class I. (168 miles), £268; Class II. (205 miles), £284; Unclassified (1,280 miles), £119.

ENGLAND.—I., £485; II., £329; III., £135; average, £201.
Counties.—I., £447; II., £309; III., £100; average, £179.
County Boroughs.—I., £592; II., £454; III., £211; average, £269.
London.—County, City and Metropolitan Boroughs: I., £2,076; II., £1,341; III., £533.
Boroughs (non-county) and Urban Districts: unclassified, £196.

WALES.—I., £293; II., £235; III., £69; average, £106.
Counties.—I., £289; II., £227; III., £57; average, £99.
County Boroughs.—I., £416; II., £486; III., £166; average, £213.
Boroughs (non-county) and Urban Districts.—Unclassified, £125.

SCOTLAND.—Classes I., £169; II., £133; III., £92; average, £118.
Counties.—Classes I., £153; II., £129; III., £58; average, £96.
Large Burghs (24).—Classes I., £483; II., £289; III., £341; average, £362.
Small Burghs.—Unclassified, £156.

GREAT BRITAIN.—I., £392; II., £276; III., £123.

Overseas Costs of Maintenance.

(Per mile, per annum.)

Jamaica.—Main roads, 1934: (*a*) £96 14*s.*, including, in the parish of Kingston, £397; (*b*) including special works and the maintenance and construction of bridges, £105. Flood damage in 1933, £163,000, and in 1934, £20,000, compares with the expenditure, under (*a*) of £227,000

Tasmania.—State roads, 1,170 miles, £54 in 1935; estimate for 1936, £56.

Victoria, Australia.—The cost, £35, of maintaining gravel roads by horse-drawn drags or small graders, has been reduced to £20 by using pneumatic-tyred power graders.

Relation of Cost to Traffic.—New Zealand: on the basis of low cost construction at £350 per mile, one coat, for traffic up to 200 vehicles per day, or 2-coat work for 800 to 1,000 vehicles per day. Costs of maintenance, including interest on the capital, short-term repayment, sinking fund, and general maintenance of the road. Vehicles per day and £ per annum: with 1,000 motor vehicles per day, £350; 750 vehicles, £275; 500 vehicles, £223; 250 vehicles, £194.

Analyses of Costs of Maintenance.

Northamptonshire.—Class I. roads, 301·7 miles; expenditure in one year, £115,217. Analysis, percentages of the total: resurfacing, 30·5; strengthening foundations, 22·6; kerbing, 6·41; surface dressing, 19·7; patching, 2·08; sanding and gritting, 1·06; siding and ditching, 5·11; drainage and culverts, 1·9; guide posts and fences, 0·71; footpaths, 7·15; scavenging, 1·52; white lines and caution signals, 0·86.

Cardiganshire (1935).—In a total of £13,265: labour, £4,750; material, £1,990; haulage, £1,245; steam rolling, £460; tarring, £4,020; bridges, £60; various items, £470.

Canada.—Percentage costs: ditching, 1·8; grading, 1·9; resurfacing, 14·6; culverts, 1·6; dragging, 0·1; oiling and patching, 61·3; snow removal, 8·7; cutting weeds, 8·6; bridges, 1·2; drain assessment, 0·2.

Costs per Mile: Directly Maintained Roads.

Maintenance and Minor Improvement.

(*Essex*, 1937–38 : *Report of County Surveyor, R. H. Buckley.*)

Road classes	I.	II.	III.	I.	II.	III.
Mileages	266·5	351·7	1931	—	—	—
		Costs.			Percentage costs.	
	£	£	£	p.c.	p.c.	p.c.
Scavenging, gulley emptying, snow clearing	21·35	13·79	1·41	4·85	3·90	1·13
Resurfacing and strengthening	69·71	120·01	24·27	16·62	34·2	20·92
Patching and gritting	37·89	39·95	17·76	9·01	11·2	15·41
Surface dressing	155·0	56·65	30·3	36·97	16·1	26·20
Maintenance of drains, footways, fencing, warning and direction signs, etc.	61·89	46·50	7·28	14·78	13·2	6·30
Siding, clearing ditches, maintenance of trees, shrubs, etc.	48·60	42·52	28·46	11·56	12·3	24·61
Provision of kerbing and channelling	6·69	13·26	1·15	1·59	3·73	1·0
Construction of footpaths	6·93	5·17	0·34	1·65	1·49	0·3
Provision of gullies, catchpits, drains, and culverts	9·46	12·84	4·60	2·25	3·63	3·98
Provision of warning and direction signs, etc.	2·78	0·89	0·17	0·66	0·25	0·15
Provision and planting of trees and shrubs	0·24	0·02	—	0·06	—	—

RESEARCH AND EXPERIMENT.

The Laboratory.

Tests of Road Stones.

(*See Report to the Third International Road Congress, T. E. Stanton and R. G. Batson.*)

The equipment of the National Physical Laboratory, Teddington, enables the following test of road stones to be made :—

(1) A dry attrition test (resistance to wear); (2) a wet attrition test; (3) an abrasion test (for hardness); (4) a repeated impact test (toughness); (5) a test of the cementation value; (6) an absorption test.

Attrition Test.—The machine is of the 4-cylinder Deval type, as used in France and in the United States. The cylinders are 7½ ins. in diameter and 14 ins. long, set at 30° to the axis of rotation. About fifty pieces of the rock, weighing eleven pounds, are placed in one of the cylinders, and the machine revolved 10,000 times at about 30 revolutions per minute. Only the material which will pass a sieve of $\frac{1}{16}$ mesh is counted in measuring the amount of wear, which is recorded both as a percentage and by the French coefficient $\frac{40}{\text{percentage of wear}}$.

In the wet test 1·1 gallon of water is used with 11 lbs. of stone. The wear is usually greater in the wet than it is in the dry test ; but not always.

Repeated Impact Test.—This test is made with the Page impact machine, which has a hammer weighing 4·4 lbs., delivering the blows through a plunger, the surface of which, in contact with the specimen, is spherical to a radius of 0·4 in. The test piece is a cylinder, 1 in. by 1 in., ground to the true diameter, and cut to length by a diamond saw. The test consists of a series of blows, beginning with a fall of 0·4 in., which is increased by 0·4 in. with every successive blow until the test piece is broken. The toughness is measured by the number of blows. Tests can also be made with a constant height of fall.

Abrasion Test.—The test piece is prepared in the same way as for the impact test and is of the same size. The machine is of the Dorry type, and consists of a cast steel disc revolving in a horizontal plane. The test piece is held, with its axis vertical and with a load equal to 3·5 lbs. per square inch, against the surface of the disc 26 cms. from the centre. Standard sand of 30 to 40 mesh is fed continuously through funnels at a definite rate, the same in each case. After about 1,000 revolutions of the disc, at the rate of 28 per minute, the test piece is weighed and the loss of weight noted. The test is repeated on the other end of the test piece, and the average loss of weight by the two runs is recorded. The hardness is measured by $(20 - \frac{1}{3}W)$ where W is the loss of weight in grammes per 1,000 revolutions.

Cementation Test.—Coarsely crushed rock, weighing 1·1 lb., is ground in a ball mill with 0·02 gallon of water, the balls being steel shots (two) 5·1 ins. in diameter. The material is ground for two and a half hours at the rate of about 2,000 revolutions per hour. The resulting paste is moulded into briquettes, 1 in. diameter and 1 in. long, the maximum pressure being 1,880 lbs. per square inch. After twenty-four hours, the briquette is dried for twenty hours in the air, and for four hours in a hot air bath at 200° F., cooled for twenty minutes in a dessicator, and then tested. It is placed on the anvil of the machine, with its axis vertical, and struck through a plunger by a small hammer with an effective drop of 0·4 in., the rebound being measured by the movement of a lever attached to the hammer, and holding a pencil against a paper-covered revolving drum. This records the number of blows required to destroy the resiliency of the test piece, after which there is no rebound. The mean of six tests is taken as giving the cementing value of the material.

Crushing Strength.—The specimen is a cylinder 1 in. in diameter by 1 in. long, ground out of a rough sample by emery-wheels. The cracking stress and crushing stress are measured.

Absorption of Water.—A stone of about 13 lbs. is weighed in air and then in water. It is kept in water for three days and is then weighed in air. If A = weight in air, B = weight in water immediately after immersion, and C = weight in water after immersion for three days, the number of pounds of water absorbed per cubic foot of rock is nearly

$$\frac{C - B}{A - B} \times 62\cdot5$$

Petrological Analysis.—The classification of the rock, in the terms of geological nomenclature and as regards its constituents, is made in the Geological Museum, London, S.W.

Ring Track Testing Machines.—No. 1 and No. 2, described in Road Research Board Report, 1933–35, mean diameter of track, No. 1, 5 ft. 6 in.; No. 2, 38 ft.; No. 3, 110 ft., described in a lecture to the Road and Building Materials Section of the Society of Chemical Industry, by R. H. Stradling. (Published with illustrations in *Roads and Road Construction*, April 1 and May 1, 1935.)

No. 3 was installed in April 1936. Loaded weight of vehicle, 12 tons; maximum speed, 40 m.p.h., effecting a centrifugal pull of 25 tons on the centre post. Traversing gear to give track wear on 2 ft. 6 ins.

Tests of Roadbed Materials.

Unconfined Compression Test.—A convenient and rapid method of measuring the shear strength of clay soils. The apparatus consists of a metal framework in which cylindrical soil specimen, 3 in. long and 1½ in. diameter, is compressed by a helical spring, the load being applied by rotating the handle at the top of the apparatus. An autographic device enables the load-deformation curve to be drawn directly on a chart mounted on the plate in front of the apparatus. For purely cohesive soils the shear strength is equal to half the unconfined compressive strength.

Shear Box Test.—Used to investigate the shear properties of all types of soil, both cohesive and non-cohesive. A soil sample of 6 c.m. square cross section contained in a metal box is constrained to shear along a definite plane, the shear strength being determined for various compressive loads applied normally to the plane of shear. The shear strength is related to the normal pressure by Coulomb's formula: $S = c + p \tan$ 'phi' where S = shear strength; c = cohesion; and p = normal pressure, all in lb. per sq. in. and 'phi' = angle of internal friction.

Triaxial Compression Test.—The test is made on a cylindrical specimen, 3 in. long and 1½ in. diameter. Its cylindrical face is covered with a rubber membrane and then subjected to a known hydraulic pressure, while an increasing axial load is applied until the specimen fails.

Plate-bearing Test.—A circular-plate is loaded on the actual sub-grade and a load/deflection curve obtained for increments of load. The modulus of sub-grade reaction is taken as the pressure required to produce a deflection of 0·05 in. measured in lb. per sq. in.

California Bearing Ratio Test.—A test (C.B.R.) devised in the United States. Related empirically to wheel loads and required thicknesses of road construction. The C.B.R. value of a soil is obtained by relating the load/penetration curve for a circular plunger, 3 sq. in. in cross section, driven into soil compacted into a special cylinder, to the corresponding curve for a standard highly stable soil.

TESTS OF BITUMINOUS MATERIALS.

Penetration Test.—The material to be tested is usually brought to a definite temperature, which for standard comparisons is 77° F. (25° C.). The penetrometer consists essentially of a weighted needle attached to the 'hand' of a dial by a simple mechanism so that the hand on the dial registers the distance moved by the needle. The point of the needle is brought to the surface of the specimen, the dial reading noted, the needle released for one second or sometimes longer, and then held. The difference between the two readings on the dial measures the penetration. It is desirable to make tests with different weights on the needle and at different temperatures.

Solubility Test.—Solubility in carbon disulphide. A small quantity is placed in a weighted flask containing 100 c.c. of carbon disulphide, and afterwards decanted through an asbestos mat and the precipitate dried and weighed.

Other Apparatus and Tests.

Hardness Number Test.—The hardness number of a sample of mastic asphalt is defined as the depth in hundredths of a cm., to which a steel rod of 6·85 mm. diameter, with the end cut off square, will penetrate the sample under a load of 100 kg. per sq. cm. (Illustration and description by D. M. Wilson, Soc. of Chem. Ind. Paper; Ref. (a), 'Roads and Road Construction,' April 1, 1932.)

Deformation Test.—On a test piece 2 ins. in diameter and 2 ins. long, cut from the pavement, or moulded. The test piece, placed as an upright cylinder, is pressed by a steel disc of somewhat larger diameter, to which, centrally, by means of a $\frac{5}{16}$ in. diameter steel ball, an initial load of 20 kg. is applied by a weight of 1,750 grms. and leverage, the load at the lever end being increased by running shot into a can hung from it and the load on the specimen thereby increased by 22·9 kg. per min., until the height of the cylinder (test piece) is reduced by twenty-hundredths of a cm., shown by 20 divisions on a dial. The result is expressed as the kg. load, on the end of the beam required to cause this shortening (ref. as (a) above).

Hutchinson Automatic Penetrometer.—In this new instrument there is no delicate or complicated timing apparatus, nor any part subject to rapid wear. The vernier system of reading penetrations eliminates the rack and pinion operated pointer and dial. A lifting gear for the sample plate provides for rapid and precise adjustment. (Description and illustrations in 'Roads and Road Construction,' October 1, 1932.)

Stability Test.—(1) A machine devised by O. R. Stokes and J. Zapata, for testing the stability of cold emulsions, is described in 'Proceedings of the (American) Association of Asphalt Paving Technologists,' 1932.

(2) (The same ref.) The Hubbard-Field apparatus, of the piston and cylinder type, is an adaptation directed to provide for the testing of coarse aggregate mixture.

(1) *Metro Plastimeter.*—For measuring the consistency of pitch-like substances. A steel sphere is forced, by the application of a weight, into a mould containing the material, the rate of penetration being measured. The sphere is of much less diameter than the mould.

(2) *Metro Trough Viscometer.*—A trough resembling the spirit level of a surveying instrument is so mounted on trunnions that it can be steeply tilted. The tar is placed in the well, or reservoir, of the trough, which is brought to the required temperature and tilted to a given angle. The consistency of the tar is expressed as the time of flow between two marks.

(3) *Capillary Tube Viscometer.*—An upper and a lower vessel, each of about 50 millilitres capacity, are connected by a capillary tube (perforation through a solid mounting) 5 cm. in length. The lower vessel, containing the tar, is put under mercury pressure applied at a hole in the bottom; from the upper vessel, containing a suitable liquid, that liquid, displaced as the tar flows upwards, measures the rate and amount of the flow.

Tar Viscometer.—A viscometer used in investigations into the use of Vertical Retort Tars, at Johannesburg, is Dr. E. H. Selvey's modification of the Metro Trough Viscometer. It was found that with this instrument the tar temperature could be adjusted to within 0·05 ° C. of the surrounding water within 15 secs.

Equiviscous-Temperature Test. The temperature in °C. at which the tar has a time of flow of 50 seconds measured by the tar viscometer. Its use enables the viscosity of any road tar, however fluid or however viscous, to be expressed on a single scale. The value becomes progressively higher as the viscosity increases.

Experimental Determinations.

Coefficient of Slab-on-Road-bed Friction.—(Ministry of Transport Technical Circular. Average values: moist earth, 0·5 to 1·3; clays, 1·5 to 2·0; very rough subsoil (? road-bed), 2·0 and upwards.

Effect of Heat on Aggregates (B. H. Knight).—It was deduced from experiments with Bonawe granite, heated to 250 and 300° F., and subjected to sudden and slow cooling, that no ill results need be feared when such an aggregate is heated as in ordinary practice.

Moisture Content of Soils.—Measured *in situ* by burying a series of electrodes. The method is based on the relation between the electrical 'restivity' (*r*) and moisture content, also the relation between (*r*) and the temperature of the soil. (Report for 1933, National Physical Laboratory.)

Wheel Impact.—The first main section of the 'Wheel Impact Research,' in which a heavy six-wheeled lorry was employed, was nearly completed in 1936. The other vehicles are: a heavy four-wheeler, a medium four-wheeler, and a motor car. Experiments are made to determine the influence of the following factors: speed, unsprung weight, load, type and size of tyre, inflation pressure, shock absorber, driven or trailed axle, and middle or rear axle. (See Report for the year 1936 of the National Physical Laboratory.)

Road Surface Resistance to Skidding.—The subject of Road Research Technical Paper No. 1 (Studies in Road Friction, I.), December 1936, in which experiments are described and the results discussed. In another publication, Road Research Bulletin No. 1, December 1936, information is given as regards the construction and operation of the special motor-cycle and sidecar, and its measuring apparatus; also guidance in respect of the interpretation of the results obtained. The Road Research Laboratory is prepared to give advice on the selection and purchase of a suitable machine, and assistance in training personnel in its proper use. (It is clearly indicated that the co-operation of county and municipal highway engineers, in this important matter, is regarded as desirable.)

Control of Moisture Content of Aggregates.—The title of Road Research Technical Paper No. 4, January 1937, is: 'The Control of the Moisture Content of Aggregates for Concrete, Introducing a new Vibration Method.' In the opinion of the Road Research Board (prefatory note) '. . . the vibration system of control should make possible a considerable improvement in the quality of concrete laid in roads.' The inundation method, devised and employed in the United States, has been fully investigated by the Road Research Board, the experiments being described and the results discussed in the paper. It was found that the method is not generally applicable in Great Britain. *The Vibration Method* is based on experiments made by Dr. G. Glanville at the Building Research Station. The vibration is applied by mechanical means, under full control, and the proportion of water in the aggregate is brought very nearly to a constant value.

Surface Area of Aggregate.—(D. Torrance and R. B. S. Gilmour.) The relationship between surface area and weight of spheres or cubes, as given by Neumann's expression, $S = 6/p.d.$ square centimetres per gramme, gives values somewhat too low. A more accurate relation, assuming that the aggregate contains, in approximately equal proportions, spheres, cubes, pyramids and triangular prisms, is given by $S.A. = 3590/p.d.$ sq. ft. per ton; where p = sp. gr.; d = gauge of stone in inches. An example, taking 65 per cent. of ¾-in. gauge and 35 per cent. ⅛ in., relates 10 gallons of bitumen per ton of aggregate to a film thickness 0·0043 in., considered to be suitable for grading ¾ in. down.

Texture Prints of Road Surfaces.—The earliest prints were taken with a pneumatic tyre, replaced—for convenience in constant use of the method—by a rubber roller giving impressions as nearly as possible the same as those obtained with the tyre. The road surface is carefully inked, over an area about 8 ins. square, with a half-tone black printing ink diluted with about half its volume of paraffin. The operator holds the roller at the nearer edge of the inked patch and rolls forward, allowing the weight of his body to maintain a steady pressure on the roller. The impression is then transferred to a sheet of paper. (*Road Research Bulletin*, No. 3, 1939.)

Lateral Pressure of Clay.—Apparatus described in paper No. 5242, *Inst.C.E.*, by G. M. Binnie and J. A. Price. The principle of the machine is similar to that of Dr. von Terzaghi's tape, in that it depends upon the measurement of friction between metal parts under load; but whereas the tape is based upon sliding friction, the machine depends upon the friction of a rotating shaft between two bearing surfaces. (*Journal Inst. C.E.*, February 1941; discussion in the issue for October 1941.)

RESEARCH.

Chemical Research Laboratory, Teddington.

Drying Time of Tar.—The admixture of 1 or 2 per cent. of calcium soap to tar reduced the drying time of road tar by about 50 per cent.

Rubber-Tar.—Chlorinated rubber, an amorphous powder, was found to be readily mixable with certain kinds of tars. It increased the viscosity of high-temperature tars. The rubber-tar adhered to granite and to glass, under water.

Gypsum Cement.—A new hydraulic gypsum cement of great strength, resisting weathering, with less expansion and contraction than Portland cement, and highly resistant to mechanical wear. (See *The Engineer*, March 4, 1938.)

ROAD MATERIALS.

BRITISH STANDARD SPECIFICATIONS.

(To be obtained from the Publications Department of the British Standards Institution. 28 Victoria Street, London, S.W. 1; price 2*s*., or with postage, 2*s*. 2*d*., unless otherwise stated.)

Granite and Whinstone Kerbs, channels, quadrants and setts (No. 435). Kerbs in a series 4 in. by 9 in. to 8 in. by 12 in.; setts in eight standard sizes.

Asphaltic Bitumen Road Emulsion.- No. 434 1935. Introduces the 'Liability' test. Emulsions for Penetration (Grouting and Semi-grouting) Method and Surface Dressing. Price 3*s*.

Cold Asphalt Macadam (No. 434).—Provides a specification, using the emulsion (No. 433). Examples of suitable grading of aggregates and rates of application of emulsion and blinding materials are given in the appendix.

'*Precast Concrete Flags.*' -No. 368—1936. A revision of the 1929 specification. The loading in the transverse tests has been increased and the specified permissible amount of wear reduced. With an amendment, August 1947.

Precast Concrete Kerbs, channels, quadrants and setts. A revision (September 1936) of B.S.S. 340. The load to be supported in the cross-breaking test has been increased. With an amendment, October 1938.

Sandstone Kerbs, channels, quadrants and setts, No. 706—1936. There are requirements as regards structure, texture, water absorbtion and density.

Hydrated Lime and Mortar.—BS/ARP No. 24—1939; relates to hydrated lime for use in making a cement-lime mortar for bonding brickwork and masonry; provides for hydrated lime produced from high-calcium quicklime or a greystone quicklime.

BS/ARP No. 25—1939, provides for a lime-cement mortar for bonding brickwork, natural or cast stone and other structural units.

Road Stone and Chippings.—No. 63—1939; sizes of road stone and chippings. Specifies square mesh sieves. (B.S. 410.)

Furnace Slag for Concrete Reinforcement.—No. 877—1939. Foamed blast-furnace slag for concrete reinforcement. The standard is restricted to material having a lime content within 50 per cent.

High-alumina Cement. (No. 915—1940.) The compression test is included, without the alternative of a tensile test.

Compressed Natural Rock Asphalt.—No. 348—1948.

Cold Asphalt Macadam.—Penetration (Grouting and Semi-grouting) Method, using Road Emulsion.—No. 433—1931.

Single Coat Asphalt (cold process).—No. 510—1933. Price 3*s*.

Two-Coat Asphalt (cold process).—No. 511—1933. Price 3*s*.

Rolled Asphalt. Asphaltic Bitumen and Fluxed Lake Asphalt (hot process).—No. 594—1945. Price 2*s*. 6*d*.

Rolled Asphalt. Fluxed Natural Asphalt and Asphaltic Bitumen (hot process).—No. 595—1935. Price 5*s*.

Mastic Asphalt Surfacing. Fluxed Natural Asphalt and Asphaltic Bitumen (hot process).—No. 597—1935.

Emulsions of Road Tar and of Road Tar-Asphaltic Bitumen Mixtures for Penetration (Grouting, Semi-grouting) and Surface Dressings.—No. 618—1935. Price 3*s*.

Tarmacadam and Tar Carpets, Granite Limestone or Slag Aggregate.—No. 802—1945. Price 2*s*. 6*d*.

Sampling and Testing of Mineral Aggregates, Sands and Fillers.—No. 812—1943. With an amendment, 1946. Price 5*s*.

Tarmacadam and Tar Carpets, Gravel Aggregate.—No. 1241—1945.

Methods of Test for Soil Classification and Compaction.—No. 1377—1948. Price 7*s*. 6*d*.

Mastic Asphalt (natural rock asphalt aggregate) for roads and footways.—No. 1446—1948.

Mastic Asphalt (limestone aggregate) for roads and footways.—No. 1447—1948.

Foamed Blast Furnace Slag for Concrete Aggregate.—No. 877—1939. With an amendment, April 1947.

Portland Cement. Ordinary and rapid hardening.—No. 12—1947. With an amendment, May 1948. Price 3*s*. 6*d*.

Portland Blastfurnace Cement.—No. 146—1947. With an amendment, May 1948. Price 3*s*. 6*d*.

Girder Bridges. Tables of British Standard Unit Loadings for railway girder bridges and highway girder bridges, No. 153, Appendix No. 1 (1925). With an amendment, May 1930. Price 2*s*. 6*d*.

Girder Bridges.—No. 153, Part 3, 1937. 3—Loads and Stresses. Price 3*s*.

High Tensile Steel for Bridges, etc., and General Building Construction.—No. 548—1934. Amendments, May 1936, February 1938, and June 1942. Price 2*s*. 6*d*.

Roadstone.

When a single word is employed as the name of a rock used for road stone or for paving setts the word may be definite in itself, denoting a rock of a particular composition which does not, as a rule, vary much from the average. In other cases, the general word, or word denoting the normal material, is compounded with another word, usually the name of a mineral which is present in an unusually large proportion, or which is an unusual constituent or one of several, any of which may be present. Thus, a diabase is usually composed chiefly of labradorite and augite, and sometimes olivine; and dolerite, though different in texture, is similar in composition. An olivine dolerite is a dolerite in which olivine is present in a considerable proportion, and a quartz-enstatite diabase is a diabase in which both quartz and enstatite (minerals, not rock names) are present in noteworthy proportions. (See 1941 and earlier issues for a classification of roadstones.)

Roadstone: the Relation of Size to Toughness.—(B. H. Knight): The results of physical tests carried out on relatively large pieces of material are apt to be very misleading if the latter is intended for use in sizes which are much smaller than those actually tested.

In a sett, the number of cracks reaching the outer surfaces will not be very great unless the stone be a particularly bad example, while the number of fissures which reach from one surface to another will be quite negligible. When, however, the same stone is crushed to small sizes, the

SCHEDULE OF REQUIREMENTS FOR ROAD TARS.

BRITISH STANDARD, No. 76—1943.

Properties.	Type A.						Type B.				Type C.	
	Viscosity Range.						Viscosity Range.				Viscosity Range.	
Equiviscous Temperature, degs. C.	13–20	20–27	27–34	34–41	41–48	48–55	27–34	34–41	41–48	48–55	34–41	41–48
Viscosity. Seconds	17–50	10–30	30–100	43–140	26–83	36–120	30–100	43–140	26–83	36–120	43–140	41–48
Temperature of test, degs. C.	20	30	30	35	45	50	30	35	45	50	35	45
Water, max. per cent. by weight	0·5	0·5	0·5	0·5	0·5	0·5	0·5	0·5	0·5	0·5	0·5	0·5
Distillation—												
a. Oils below 200 degs. C., max. per cent. by weight	1·0	1·0	1·0	1·0	0·5	0·5	0·5	0·5	0·5	0·5	0·5	0·5
b. 200–270 degs. C. per cent. by weight	12–17	9–15	6–13	4–11	3–7	0·5–4	4–11	1–8	1–5	0·5–3	1–5	0·5–3
c. 270–300 degs. C. per cent. by weight	5–10	4–10	3–9	3–8	2–7	2–7	4–9	4–9	3–7	2–7	3–7	1–5
b. + *c*. max. per cent. by weight	27	23	20	17	13	10	16	13	11	9	10	8
Softening point of residue												
Ring and ball test, max. degs. C.	52	52	52	52	54	56	46	46	48	50	37·5	37·5
Kraemer & Sarnow test, max. degs. C.	44	44	44	44	46	48	38	38	40	42	29·5	29·5
Phenols, max. per cent. by volume	5·0	4·0	3·5	3·0	2·5	2·0	3·0	2·5	2·0	1·5	1·5	1·5
Naphthalene, max. per cent. by weight	6·0	5·0	4·5	4·0	3·5	2·5	1·0	3·5	3·5	2·5	1·0	1·0
Matter insoluble in toluene, max. per cent. by weight	21	22	23	24	25	26	23	24	25	26	20	20
Specific gravity at 15·5 C./15·5 C.—												
Minimum	1·090	1·100	1·110	1·120	1·125	1·135	1·110	1·120	1·125	1·135	1·120	1·125
Maximum	1·230	1·240	1·250	1·260	1·265	1·275	1·250	1·260	1·270	1·280	1·260	1·270

ratio of fissures reaching the surfaces to the total volume of stone becomes very much increased, and hence the tendency for the stone to fracture under traffic is very much greater. This is the reason why some granites which are perfectly satisfactory as setts fail badly as chippings.

EXPERIMENTAL MATERIALS.

RUBBER PAVINGS.

For descriptions and dates of laying, see the 1937 edition, p. 646. Further information was given in subsequent editions, up to that of 1941. It relates to pavings laid in London, Glasgow, Edinburgh, and the Mersey Road Tunnel; also to rubber latex carpets and to rubber-bitumen mixtures.

In view of the small extent to which rubber can economically be employed for paving roads, and the low coefficients of friction of the surfaces, repetition of the information so far given does not seem to be justified.

RUGOSITY OF RUBBER PAVINGS.

Skidding Tests: surfaces wet. Rubber block surfaces in London: (1) E.C. District, January: Old paving; at 15. 20, 25 and 30 m.p.h., respective coefficients, 0·29, 0·27, 0·26, 0·25; New paving, at 15, 20, and 25 m.p.h., respective sideway force coefficients, 0·25, 0·23, 0·21. (2) S.W. District, September; at 15, 20, and 23 m.p.h.; respective coefficients, 0·23, 0·18, 0·17. (Road Research Technical Paper No. 1.)

Rubber Latex.—(*a*) Latex, 10 per cent. vulcanised, rolled to a carpet ¼ in. thick on open tar-macadam; (*b*) the same latex, trowelled into open tar-macadam. Skidding tests, surfaces wet. February, gave, for 15, 20, 25 and 30 m.p.h., the respective coefficients, (*a*) 0·20, 0·17, 0·16, 0·16; (*b*) 0·63, 0·59, 0·56, 0·50. Later, in August, both surfaces having been dressed with bituminous emulsion and sand, the corresponding coefficients for both surfaces, wet, were 0·40, 0·34, 0·31, 0·29.

CAST IRON PAVING.

Experimental lengths of carriageways have been paved with cast iron blocks (proprietory) at Nottingham, Worcester, Accrington, Stratford-by-Bow, Islington, Birmingham, Burnley, Rochdale, Ilford, Leyton, Hampstead, Stockport, and Folkestone.

Each block is an equilateral triangle of about 11½-in. sides and weighs about 10 lb. It is dished and ribbed to give a depth of 2 in. At each corner a steel stud projects about ⅛ in. on the underside, these preventing the block from rocking. The wearing surface is grooved to about ⅛ in. depth. The blocks are bedded on a bituminous mastic, upon concrete, and the vee joints, ⅜ in. wide at the top, are filled with a special jointing material.

Cardiff.—The city engineer has reported unfavourably on an experimental length of cast iron paving.

Mersey Tunnel.—At the contract price of 16*s*. 9*d*. per sq. yd., 46,000 sq. yds. of cast iron paving was provided for the Mersey Tunnel.

Rugosity.—On some types, the coefficient of sideway force is very low. With a new type of cast iron paving, having a surface pattern of small studs ½ in. square, the coefficients of sideway force ranged from 0·6 at 5 m.p.h. to 0·54 at 25 to 27 m.p.h.

EXPERIMENTAL BINDING MATERIALS.

Salt.—As distinct from its employment as a hygroscopic surface binder, common salt has been used as a road crust, or wearing course, binder on a considerable mileage of roads in the United States and Canada. Recent information relates to the use of a binder consisting of clay, calcium chloride, and common salt, the proportions not being given. It is reported that salt-bound roads become very hard and tough.

Fabric Binders.—Used in bituminous carpeting, or heavy surface dressing. (*a*) Cotton fabric: recent reports from the United States are not optimistic as to the future of this method of reinforcing the surface of a road. The fabrics employed cost, in different districts and according to the weight of the fabrics, $450, $600 and $750, per mile of 18-ft. road. (*b*) Gunny cloth: In experiments recently initiated in India, gunny bags were slit open and trodden into a layer of bituminous road binder, and were covered with chippings at 5 cub. ft. per 100 sq. ft., rolled in. The cost of the gunny cloth was R. 1, An. 8½ per 100 sq. ft., and that of the whole of the surfacing work was Rs. 8, An. 2 per 100 sq. ft.

Sira.—*Kankar* (soft limestone) roads grouted with *sira* (or molasses), a by-product in the manufacture of sugar from sugar cane. The old surface was scarified to a depth of 4 in. and the material screened, ½ in. up being kept on the road and ½ in. down reserved for top dressing. The *sira* was spread at 15 tons per mile of 10-ft. road, which was then sprinkled with water and rammed. The fine material was spread and treated with a diluted mixture of sira and water (sira at 1 ton per mile). It was then rolled, left for 3 days without traffic, and kept wet for a further 4 days. Costs: per mile, 10 ft. wide: sira, 16 tons at Rs. 22 per ton, Rs. 352; tools and plant, Rs. 48; labour, Rs. 238; consolidation, Rs. 106; total, Rs. 744.

The Vehicle and the Road.

REGULATIONS UNDER THE ROAD TRAFFIC ACT, 1930.

Motor vehicles are defined in terms including the following: *Heavy Locomotive*, a vehicle not constructed itself to carry any load and the weight of which, excluding water, fuel, accumulators and tools, exceeds 11½ tons. *Light Locomotive*, similarly, exceeds 7¼ but does not exceed 11½ tons. *Motor Tractor*, similarly, does not exceed 7¼ tons. *Heavy Motor Car*, constructed to carry a load or passengers, unladen weight exceeding 2½ tons. *Motor Car*, similarly, but unladen weight under 3 tons if (1) constructed solely for the carriage of persons and their effects, (2) adapted to carry not more than 7 passengers exclusive of the driver, (3) fitted with tyres of a prescribed type, or unladen weight does not exceed 2½ tons. *Public Service Vehicles*, include three classes, stage, express and contract carriages.

The respective limitations of weight are subject to alteration, as are other of the regulations. The following, which have significance with respect to road design are herein referred to in that aspect only. Information as to the regulations in force at any given date can be obtained from H.M. Stationery Office.

Public Service Vehicles.—All-wheel load (*a*) and load of 2 wheels in a tranverse line (*t*). Four-wheelers: double-deckers, (*a*) 10 tons, (*t*) 6⅜ tons; single-deckers, (*a*) 9 tons, (*t*) 6 tons. More than 4 wheels; (*a*) 12 tons, (*t*) 4½ tons.

Abnormal Indivisible Loads.—(Authorisation of Special Types Orders.) Only one such load may be carried at one time, unless within the normal weight limits. Vehicles or trailers (S.T.), used only for the conveyance of such loads are limited by: all-wheels' load not exceeding 15 cwt. per inch width of tyre in contact with the road. Four days' notice of the transport of such loads must be given, containing an indemnity in respect of any damage which may be caused.

Flexibility of Springs.—In respect of (S.T.) above, the following is ordered: every such trailer constructed after January 15, 1931, which has more than 4 wheels in contact with the road surface must be so constructed that, under any condition of loading, when it is at rest upon a level surface, all wheels shall be in contact with the road surface and, if any wheel is lifted and supported at a distance of 3 ins. above such surface, the weight transmitted to the road surface by any wheel must not be increased by more than 10 per cent.

TYRE WIDTHS IN PROPORTION TO LOADS.

Even when superseded, or no longer applicable to existing conditions, regulations relating to tyre widths in proportion to wheel loads have a bearing upon the design of carriageways. The following paragraphs are, therefore, retained.

Information in regard to the tyre widths which are the least allowed for certain wheel loads, to the unit width per unit of loading and the variations in accordance with the character of the tyre, will be found in reports to the S.I.R.C. (D. A. Crawford, Report No. 80). 6th Question. For a summary relating to eight reports, see the *Surveyor*, September 21, 1934, p. 288. The following information is from a comprehensive table for Australia, relating to the regulations in :—

Queensland.—(1) Regulations of 1933, main roads: solid rubber, single tyres, graduated rom 18 cwt. per wheel on 4 ins. to 73 cwt. on 14 ins.; twin tyres, from 36 cwt. per wheel on (two) 4 ins. to 73 cwt. on (two) 8 ins.

Crawler Tracks: 10 cwt. per in. width of track of flat steel, wood or rubber pads in contact with the ground. To ascertain the maximum weight load in cwt. multiply the sum of the widths by 10.

Tasmania.—Per wheel, 3½ cwt. per half inch width of tyre.

India.—Loads and Tyres of Bullock Carts.—In an article by T. S. Pipe, executive engineer, Bombay Presidency (*Indian Roads*, March 1933), the actual loads carried per inch width of tyre by wheels of bullock or buffalo carts in India are compared with those given on p. 633 (1937 edition). In twenty-seven areas, such as circles (P.W.D.) or districts (administrative) the lowest loadings in lb. per in. width of tyre include only five between 410 and 500, and the others include twelve from 1,000 to 1,800.

Assam.—Tyre widths were prescribed under the Highways Act of 1928. The maximum loadings is 275 lbs. per in. width of tyre.

Burma.—No vehicle with untyred wheels nor any with wheels having less than 2½ ins. bearing width of tyre may be used. For 2-wheeled vehicles the minimum widths of tyre are, for half a ton, 2½ ins.; for 1 ton, 3½ ins. for 1½ ton, 4½ ins.; for heavy loads transport, from 2 tons to 8 tons, 5 ins. to 12 ins.

WHEEL DIAMETER AND LOADINGS.

Queensland.—(1) Four-wheeled vehicles; smooth metal; wheel diameters, inches, and permissible loads in cwt. per in. width of tyre per wheel. Under 15 ins., 3 cwt.; 15 to 24 ins., 4 cwt.; 24 to 36 ins., 5 cwt.; 36 to 48 ins., 6 cwt.; 48 to 62 ins., 7 cwt.; 62 to 76 ins., 8 cwt.; 76 ins. and over, 8½ cwt.; for grooved steel tyres, half, and for peneumatic tyres, one and a half times those loadings. (2) Two-wheeled vehicles, for the same range of diameters, half one cwt. less in every case; grooved steel tyres and pneumatic tyres in the same proportions as above.

South Australia.—Wheel dia. 30 ins. or less, 7 cwt. per in.; more than 30 ins. in dia., 8 cwt.

TRACTIVE EFFORT.

In order to establish, in the economic aspect, relative values for power called for on the level and power called for on gradients (both gradients and power being related to costs), it is necessary to know, approximately, what is the tractive effort required on the level. The investigations, particular in every case, correlate capital expenditure on road construction with running costs of vehicles.

Tractive Effort, Germany.—Report to the S.I.R.C. (1) Not including air resistance, kg. per tonne, with high-pressure pneumatic tyres: small stone setts (Gabbro), 15; gravel-bound, broken-stone roads, 18; broken-stone and asphalt, 15; large stone setts, 16; paving stones, 21. (2) At about 19 miles per hour, including air resistance and that (20 per cent.) due to curvature of the track of 196 yards radius; relative resistances: stone setts, 100; concrete, 103; gravel-bound broken-stone, 106; asphalt, 110; tar surfaces, 111. (3) Torque measurements showed that moments of starting and braking couples are from 2·3 to 6·2 times that when the vehicle is running at 19 miles per hour, it being deduced that wear of the road is in similar proportion.

Tractive Effort, France.—Report to the S.I.R.C. (citing data, M. Castaing, 1932). For Paris motor omnibuses, kg. per tonne: on asphalt, 14·50; on wood, 14·75; on small sett paving, 14·80; on sandstone 'setts,' 16·25. When the flexibility of the springs was about doubled, the resistance to traction increased by about 14 per cent.

Tractive Effort and Speed.—Report from Japan to the S.I.R.C. Low-pressure pneumatic tyres, large sett paving, kilms. per hour, kilogs. per tonne: 10–12·4; 20–14·2; 30–15·7; 40–17·2; 50–18·9.

ROAD ACCIDENTS.

Ministry of Transport Analysis of 1935–37 *Statistics. Result Classification.*—Numbers of accidents: in built-up areas, fatal (f.), 3,999, or 2·6 p.c.; serious injury (s.), 34,894, or 22·6 p.c.; slight injury (l.), 115,507, or 74·8 p.c.; in areas not built up: f., 2,338, or 5·2 p.c.; s., 15,825, or 35·5 p.c.; l., 26,499, or 59·3 p.c.

Responsibility Assigned To.—In the percentages of cases, fatal: pedestrians, 40; cyclists, 17; non-fatal: pedestrians, 30; cyclists, 23. Responsibility, all accidents (percentages): drivers, 33·6; pedal cyclists, 22·8; pedestrians, 30·5; other persons, 2·8; vehicles or equipment, 3·6; miscellaneous causes, 5·8; not traceable, 0·9. Those figures include 14,500 accidents (in a total of nearly 200,000) attributed to the fact that children under seven years of age were unaccompanied or inadequately supervised.

Conditions.—Percentages (fatal, f., non-fatal, n.):—

Road.—At junctions: f., 31·7; n., 41·7; straights, or open bends: f., 60·5; n., 52·6; in built-up areas: f., 63·1; n., 78·0.

Weather.—Clear: f., 81·6; n., 82·7; rain and hail: f., 14·2; n., 14·0; fog or mist: f., 3·3; n., 2·2.

Light.—Daylight: f., 57·4; n., 69·9; dusk: f., 4·7; n., 4·4; dark: f., 37·9; n., 25·8.

Carriageway Widths.—Not more than 20 ft.; f., 22·7, n., 21·1; 20 to 30 ft.: f., 47·8; n., 46·8; 30 to 40 ft.: f., 20·3; n., 21·6.

War-time Road Accidents.—In the year ended August 31, 1940, the number of persons killed was 8,347, comparing with 6,624 in the last 12 months of peace. Pedestrians killed numbered 4,932, an increase of 1,878 on the figure for the preceding 12 months, though the increase for all users was 1,719. There was, therefore, a decrease among other classes of users. The greatest increase in accidents since the war began has been in daylight hours.

FUNDAMENTALS OF ROAD PLANNING.

SEGREGATION OF TRAFFIC CLASSES.

In the planning, or planned development of the road system of Great Britain, it is now realised to be urgently necessary to decide the extent to which the separation of fast through traffic is to be provided for, and in what manner. It is almost generally accepted that two additional one-way carriageways would be required, and majority opinion seems to be that to provide these alongside existing trunk roads would cost more than construction on new locations, or, in another aspect, would make impossible economic capital expenditure in securing shortening of distances and reduction of gradients. Roads for complete separation of fast through traffic from slow and mixed traffic may best be called 'speedways,' since the main separation is no longer that of motor vehicle traffic from horsed vehicle traffic.

SPEEDWAYS: POLICY AND RECOMMENDED PROVISIONS.

The Institution of Highway Engineers.—The council of the Institution of Highway Engineers has approved a report of a committee whose recommendations included: (1) Fast through traffic should be separated from mixed traffic. (2) Speed routes should be provided exclusively for motor traffic. (3) Such speed routes should be designed, constructed and maintained by the Minister of Transport through his own staff.

Other Opinions.—Opinions favourable to the policy of providing special roads for through traffic have been expressed by: Mr. D. Edwards, in his presidential address to the Institution of Municipal and County Engineers; by Mr. T. G. Wilkins, in his presidential address to the Institution of Civil Engineers Northern Ireland Association; and by a considerable number of county surveyors. Such provision seems to be contemplated by highway engineers in the United States, as a development of the near future. It was accepted, in principle, by the Government of the Netherlands, in which country the construction of a speedway, without level crossings, between Arnhem and Nijmegen was approved before the war.

Main Features of Speedways as Contemplated.

Great Britain.—(J. E. Swindlehurst, Institution of Highway Engineers). A series of new trunk roads, the traffic on which is limited to defined classes of vehicles. Of the twin carriageways type, of width to suffice for expected traffic for many years to come. By-passing large centres of population. Connected, at defined points, with main trunk roads, by means of link or radial roads. Accommodating only high-speed motor traffic of the pleasure-car class, and motor transport vehicles. Additional traffic lanes would be provided where the route links two busy centres of population. No road junctions or crossings on the level.

Many other descriptions of the speedway as proposed for Great Britain have been furnished. Substantially, they tally with the provisions made in respect of Italian and German motorways, as regards main features and lay-out and connections with existing roads.

Location.—Opinion seems to be, on the whole, favourable to such location of the speedways and such excavation and embankening as would reduce expropriation costs, but would involve high construction costs in order that distances travelled might be reduced and gradients be made very easy, compensation being found in reduced running costs of vehicles.

Proposed Speedway in Lancashire.

In a report of the Highways and Bridges Committee of Lancashire County Council, the construction of an entirely new road was recommended, as preferable to the widening of an existing route and the construction of many by-passes. The length of the road from the Preston boundary to the Cheshire boundary would be about 23 miles, with crossings or junctions at 10 classified roads, 16 minor roads, 20 farm or occupation roads, and 35 public footpaths. The proposed lay-out is shown as providing: two outer verges, each 9 ft. 6 ins. wide; two 22-ft. carriageways, and a 22-ft. middle reservation.

Features of German Speedways.—The system aggregates about 5,000 miles. Usual provisions: dual carriageways, each 7·5 (24·6 ft.) wide; a middle strip (along which a hedge is sometimes planted) 5 m. (16·4 ft.) wide; flanking strips of bituminous macadam, 1 m. (3·28 ft.) wide; outer grass margins of 1 m. Minimum radii of curves; in level country, 2,000 m. (6,562 ft.); in irregular country, 1,000 m. (3,281 ft.); in mountain country, 400 m. (1,312 ft.).

Designing in Respect of Speed.

Speed versus Safety on Curves.—The title of an article in *Civil Engineering* (U.S.A.), February 1937, presenting the second part of Prof. R. A. Moyer's summary of the results of investigations for determining uniform speed-control standards, published in Bulletin No. 120 of the Iowa Experimental Station. Points from the article:—

(1) A significant feature of the slippage tests is the magnitude of the slippage as the speed of the car is increased. The trend indicates that the slippage at speeds of 80 m.p.h. or more is so large on curves sharper than 3 deg. that the most skilful driver will have difficulty in steering the car and holding it within a 10-ft. traffic lane.

(2) Even on a good road surface with excellent brakes, an expert driver operating at 80 m.p.h. cannot stop the car safely in less than 500 ft. after he first recognises the danger, or if he desires to overtake a car travelling 70 m.p.h. in the face of another car approaching at 80 m.p.h., a clear space of at least one-half mile is necessary.

(3) Engineers are accustomed to build structures with a factor of safety consistent with the risk of failure involved? Yet in highway safety work engineers are very prone to determine safe driving speeds without any regard for a factor of safety. Records of automobile accidents indicate that driving cars with little or no factor of safety is certain to lead to dire consequences. Traffic engineers and drivers should recognise the fact that the law of averages holds true and is as certain to be enforced as the law of gravity.

(4) There are many factors which indicate that it may prove far cheaper and possibly equally safe to take to the air when speeds above 60 m.p.h. are desired.

In reference to (3), and to the growing opinion amongst American engineers that there should be factors of safety in road designing, it was observed in an editorial note, *The Surveyor*, April 16, 1937: 'This logically involves the stultification of nearly all recent dispositions in respect of sighting distance and other features of lay-out.'

Road Surface Rugosity

For convenience of reference and memorising, the stopping distances in the following table are, when not multiples of 5 ft., brought to the next higher multiple of 5 ft. Except for that modification, they substantially represent the values in Major C. Cook's graph (Paper No. 5105, *Proc. Inst. C.E.*, December 1936), based on a reaction time of one second. Major Cook's values are given as related to braking efficiences expressed as percentages, which, divided by 100, correspond to road rugosity coefficients available with full mechanical efficiency of the brakes.

Approximate Stopping Distances in Feet.

Speed m.p.h.	Coefficients of Road Rugosity.					
	1·00	0·60	0·50	0·40	0·30	0·20
20	45	55	65	70	75	100
30	75	95	110	120	145	195
40	115	150	170	190	240	325
50	160	215	240	280	350	490
60	210	285	325	380	490	—

Sideway Force Coefficents.

Road Research Technical Paper No. 1.—The following values of coefficients of sideway force are from the text or graphs of that publication. The road surfaces were wet, in all cases. (Extrapolations by the section editor.)

(*a*) Surfaces laid in 1930 and tests made in 1933. Single-coat hot asphalt, as left by roller: At 20 m.p.h., 0·65; at 30 m.p.h., 0·40 (as judged by extrapolation: at 40 m.p h., 0·26; at 48 m.p.h., 0·23).

Mastic asphalt, B.S.S. 346: ½-in. coated chippings spread over and rolled in with ¾-cwt. hand roller. Fine sand then sprinkled on, and surface rolled with 3-ton petrol roller. At 20 m.p.h., 0·75; at 30 m.p.h., 0·53. (By extrapolation: at 45 m.p.h., 0·33.)

Single-coat, hot process tar macadam: granite aggregate; sprayed with tar and covered with ¼-in. to ⅜-in. chippings. Rolled with 8-ton roller. At 20 m.p.h., 0·76; at 30 m.p.h., 0·57.

Single-coat hot process tar macadam: slag aggregate; nine weeks after laying, ⅜-in. granite chippings spread over tar-sprayed surface and rolled with 7-ton roller. At 20 m.p.h., 0·80; at 30 m.p.h., 0·50. (By extrapolation: at 40 m.p.h., 0·28; at 48 m.p.h., 0·23.)

Two-coat cold asphalt: sealing coat of ⅛-in. to $\frac{1}{32}$-in. granite grit, treated with flux oil and asphalt cement. Rolled with 8-ton roller. At 20 m.p.h., 0·81; at 30 m.p.h., 0·52. (By extrapolation: at 40 m.p.h., 0·34.)

Two-coat cold asphalt, left as finished: at 20 m.p.h., 0·53; at 30 m.p.h., 0·37 at 35 m.p.h., 0·32. (By extrapolation: at 45 m.p.h., 0·28.)

Cold asphaltic concrete, painted with bitumen emulsion and spread with ⅜-in. chippings: (1) At 20 m.p.h., 0·72; at 30 m.p.h., 0·45. (By extrapolation: at 35 m.p.h., 0·30; at 40 m.p.h., 0·20; at 45 m.p.h., 0·15). (2) At 20 m.p.h., 0·77; at 30 m.p.h., 0·55. (By extrapolation: at 35 m.p.h., 0·42; at 40 m.p.h., 0·32; at 45 m.p.h., 0·24.)

(*b*) Other cases. Asphalt, at 30 m.p.h.: 0·48 in June and 0·58 in October; 0·42 in June and 0·51 in October; 0·24 in November and 0·38 in December.

(*c*) Granite and gravel surfaces gave sideway force coefficients of 0·59 to 0·78 at 30 m.p.h. about three years after surface treatment. (See also figs. 50 to 56 in Technical Paper No. 1, noting the small decreases in the values of coefficients with increase in speed, in the light of the information as to methods of preparation and treatment.)

Coefficients of Rugosity.—In Paris. Coefficients ranged from, wet asphalt, 0·2 to 0·33 to, dry wood, 1·05. 'Non-skid' surfaces, wet, 0·84; stone, dry, average 0·60 in the range 0·5 to 0·8. Cement concrete, wet, 0·25 to 0·80, average about 0·55; dry, consistently approximating to 0·75.

DESIGN IN PLAN AND PROFILE.

Visibility at Summits.

The Ministry of Transport Memorandum of August 1930 advises that curves of parabolic form shall be provided at all changes of gradient and be such as will provide a clear view of 50 yds. on each side of a summit to drivers of mutually approaching vehicles. In many States (U.S.A.) it is required that a 500-ft. line of sight must be provided 5 ft. above the crown of the road.

New Zealand.—The Main Highways Board of New Zealand recommend for the classes: I., 300 ft.; II., 200 ft.; III., 100 ft. view. The height of the driver's eye is usually taken as 4 ft. At summits a vertical curve of 1,250 ft. radius is provided. (This gives 100 ft. between two drivers approaching from opposite directions.)

Steepest Gradients.

New Zealand.—Standards aimed at: road classes: I., 1 in 15 to 1 in 20; II., 1 in 12; III., 1 in 10.

Radius at Bends.

The Ministry of Transport Memorandum of August 1930 advises that where conditions are favourable, a minimum radius of 1,000 ft. should be aimed at; failing which, the width of the carriageway at the bend should be increased. Normal practice in open country in the United States is to provide 5 degs. (1,146 ft. radius) or 6 degs. (955 ft.) curves. The German reporters to the Sixth International Road Congress prescribe a minimum radius of 300 metres in level and rolling country.

The Main Highways Board of New Zealand recommend, in easy country, 6, 4½ and 3 chains. (100 ft.) radius, for first, second and third class roads; in difficult country, 2, 1½ and 1 chain. Additional widths, on the inside of the bend, are given, up to 3 chains, 5 ft.; 4 chains, 4 ft.; 5 chains, 3 ft.; 10 chains, 2 ft.

Superelevation at Bends.

A convenient approximate formula for balance of centrifugal and gravity accelerations, or of centrifugal force and component of weight is:

$$\frac{h}{l} = \frac{v^2}{15 \times r}$$

where h is the height of tilt, l the horizontal width, in the same units; v the speed in miles per hour, r the radius in feet.

The reporters to the Sixth International Road Congress, October 1930, recommend the following superelevations: for radius 50 to 100 ft., 1 in 10; 200 to 500 ft., 1 in 12; 500 to 800 ft., 1 in 16; 800 to 1,000 ft., 1 in 24. The German reporters indicate 4 in 100 as normal, 5 in 100 on sharp curves and 6 in 100 where the curve is on a steep gradient. The French reporters give, for roads in Morocco, a limit of 1 in 10 on curves of short radius, and on mountain roads, where the radius r corresponds to superelevation 1 in 10 the superelevation of a curve of radius R is

$$d = \frac{r}{R \times 10}$$

The Main Highways Board of New Zealand employ a formula

$$O = \frac{6 \times S^2}{1000\ R}$$

in which O is tilt in inches per foot, S, speed in miles per hour and R, radius in 100 ft. chains. It gives superelevations three-quarters of those for balance, and is a modification of h in inches $= l$ in feet $\times \frac{S^2}{R} \times \frac{4}{5}$ in which R is in feet.

The Board recommends tilts in inches per foot: 2 chains, 1¼; 3 to 7 chains, 1; 8 to 15 chains, ¾; 16 to 40 chains, ½; high-class pavements not more than 1 in.

Ayrshire (D. Torrance).—A crossfall steeper than 1 in 12 should not be adopted if conditions safe both for fast traffic and slow, horse-drawn traffic are to be obtained. Example: radius 400 ft., superelevation for 30 m.p.h. gives a crossfall of 1 in 6·66, very dangerous for a horse-drawn cart loaded high with hay. The transition length is often 100 ft. If it be impracticable to use a transition of this length a shorter one may be used, and a small vertical curve inserted at each end of the transition. (The consideration in respect of the hay-cart applies to high-loaded motor vehicles, when obliged to slow down or to stop on a bend.)

Winchester By-Pass.—Dual 20-ft. carriageways; minimum radius of bends, 1,000 ft.; steepest gradiant, 1 in 25; minimum vertical sighting distance at 3 ft. 9 in. above road surface, 400 ft.

Roads designed for High Speeds.—Oregon. Designed speeds (S) 80 per cent. of critical speeds (attainable with safety only by highly skilled drivers). Relations of speeds to some limiting factors: Curves of 2 and 3 deg.; vertical sight distance (*v.s.d.*), 1,500 ft.; S, 80 *m.p.h.*; stopping distance (*p.u.*) in feet, dry pavement, 490, wet 740; sight distance (*s.d.*) on curves, 2 deg., 960; 3 deg. 785 ft. Curves of 4 deg.; *v.s.d.* 1,500 ft.; S, 75 *m.p.h.*: *p.u.*, dry, 290, wet, 430 ft.; *s.d.* on curves, 675 ft. Curves of 6 deg.: *v.s.d.*, 1,000 ft.; S, 60 *m.p.h.*; *p.u.*, dry, 190; wet, 290; *s.d.* on curves, 450 ft. Curves of 10 deg.: S, 45 *m.p.h.*; *p.u.*, dry, 120; wet, 175; *s.d.* on curves, 355. On tangents: S, 80 *m.p.h.*; *p.u.*. dry, 330; wet, 490. Length of transition curve, in feet:

$$l = (1 \cdot 0517 \, V^3)/r.$$

where V = speed in *m.p.h.*, r = rad. of curve, in feet. (R. H. Baldock, *Engineering News-Record*, May 23, 1935. Abstract, *The Surveyor*, June 14, 1935.)

Overtaking and Passing.—For speeds, S_1 of the passing vehicle, S_2 of the vehicle overtaken and passed, S_3 of a vehicle approaching from ahead, the sight distances, x, are:

$$x = (S_1 + S_3)\left(\frac{A}{S_1 - S_2} + t\right)$$

in which A is the distance in feet from the back of the car S_1, to the front of the car S_2, at the moment when the decision to overtake and pass is made, and t is the time required (about 3 sec.) for the passing vehicle to return to the left-hand (in G.B.) or right-hand (U.S.A.) side of the road. (Oregon. R. H. Baldock.)

Hedges in Relation to Snowfalls and Glare. (*The Surveyor*, March 19, 1937.)—The essentials of the hedge provision are, first, that it must be thick enough to check the wind effectively, so that a drift shall form on the windward side and, if it is a low hedge, on the lee side also; next, that it must either be high enough to hold back nearly all the snow, or be far enough from the carriageway or footway to allow space, on the verge, for a drift on the lee side; thirdly, that where it must be kept low to allow of sighting over it, its width should be increased beyond the normal for a hedge, in order that a body of snow may be formed between the windward and the lee drifts.

Where it is desired to afford protection from the dazzle of vehicle lamps, for pedestrians, by planting between the footway and carriageway, or for drivers of vehicles, by planting in the middle strip, the proper provision is a series of clumps of shrubs or bushes, not more than a few feet in width, in the direction along the road, but of considerable breadth transversely. Their spacing is a detail of surveying, in respect of the lines of sight and positions of oncoming vehicles; the wider the clumps are, transversely to the road, the greater can be the spacing. On bends, the spacing will be inversely proportional to the radius. Should hedges be provided, snowdrifts may form against them, and, unless they are on wide dividing strips, block the carriageway or footways.

Standard Road Widths.

A Ministry of Transport Circular of March 1936, No. 454 (Roads), prescribes as minimum standard widths 'appropriate in ordinary circumstances':—

Single carriageway, not exceeding 30 ft.; with footpaths, 60 ft.: with cycle tracks, also, 80 ft. Dual carriageways, each of two traffic lanes; with footpaths, 80 ft.; with cycle tracks, also, 100 ft. Dual carriageways each exceeding two traffic lanes; with footpaths, 100 ft.; with cycle tracks, also, 120 ft. Where further provision is required, for wider cycle tracks, additional width of verges for improved visibility or for equestrian traffic, greater space for services or improved amenities, 140 ft. 'Extended standard widths' include increments of 20 ft. in cutting or on embankment; otherwise, the width should not exceed 160 ft. The traffic lane unit is 10 ft., the cycle track minimum is 6 ft. and increments 3 ft. The standard widths adopted should provide for such future widenings of carriage-, cycle-, and foot-ways, as may be necessary. There should, therefore, be ample central reservations, margins and verges.

Dual Carriageways.—Will be desirable where 'an existing road carries or a proposed road is expected to carry' 400 vehicles in the peak hour.

Road Widths: Typical Provisions.

Widths including Embankments and Cuttings.—It is usual to specify that the widths of embankments and cuttings shall, for widths between improvement lines, be added to the normal widths for roads on the flat.

Worcestershire.—Highways Committee recommend as least widths: between buildings, Class I. roads, 120 ft.; Class II., 100 ft.; highway widths, Class I., 60 ft.; Class II., 50 ft. *West Riding*: Highways Committee recommends, for widenings, when necessary and economically practicable, Class A, 80 ft.; B, 60 ft.; C, 45 ft.

Carriageway Widths.—Nottinghamshire, approximate verge width, by calculation, of 361 miles of tarred and other surface-treated roads (in a total of 561 miles of main roads), the treated width, 18 ft. Similarly, Isle of Ely, average treated width of tarred roads, 15 ft. 8½ ins. Berkshire, Bradfield rural district, 196 miles, approximate metalled width, by calculation, 13 ft.

New Zealand.—Normal widths of carriageways, or surfaced widths: Classes I., 18 to 20 ft.; II., 14 to 18 ft.; III., 10 to 14 ft. Normal widths of formation, exclusive of water tables: I., 24 to 30 ft.; II., 18 to 24 ft.; III., 14 to 18 ft.

Nigeria.—For Classes I., II. and III. respectively. In normal ground: between inner edges of ditches, I., II. and III., 40 ft.; at top of embankment, 24, 20 and 16 ft.; in cuttings, between edges of side ditches, 22, 18 and 14 ft.

Acceleration and Deceleration Lanes.

Calculations (*a*) of the lengths and widths necessary to allow for acceleration to traffic-flow speed, with a further pulling up distance (in a blind alley), in case there should be no opportunity to weave into the traffic flow, and (*b*) of the length allowing for deceleration, to a stop or to a gateway or turn-in speed, are given in a paper by A. Mitchell, senior traffic engineer, State Highway and Public Works Commission, North Carolina. (*Proc. Am. Soc. C.E.*, March 1941 : Abstract, *The Surveyor*, June 20, 1941.)

Settlement of Embankments.

(*A. H. D. Markwick and G. C. Wilson, of the Road Research Laboratory.*)

The embankments, on the Mickleham and Caterham by-pass roads, Surrey, are of chalk and rest on solid chalk at shallow depth. A main result is that where settlement of less than about ½ in. occurred, there was no appreciable cracking in the concrete carriageway. More or less serious cracking occurred where there were settlements of 1 in. or more, and serious damage where differential settlements on the slabs exceeded 1½ ins. in 25 ft. The settlement/time curves are of the same type in all cases. Among the conclusions are, that on an embankment on a curve of small radius, greater settlement occurs on the outside of the curve ; that the movements were continuing, more than two years from the initiation of the measurements ; and that wide variations occur in the settlements observed even over quite small distances. The work as a whole shows that, if damage to rigid surfacings is to be avoided, only very slight settlements can be permitted after the construction of the final surfacing.

[The last conclusion is a fact well known, for many years past, to all experienced road engineers. It is one of the basic facts.—*Section Editor.*]

CARRIAGEWAY CONSTRUCTIONS.

Reinforced Concrete Slab Carriageways.

Depth of Slab.—Much thinner slabs than those usually employed would be theoretically sufficient in most cases, and usually more economical of materials, but practically it is found to be better to have thicker slabs for the following reasons : (*a*) With a deeper spacing between the upper and lower grilles, less steel is needed, the mesh can be larger, larger stones may be used in the aggregate, and the filling in of the concrete is done more easily. (*b*) Any fairly large piece of road metal, left accidentally on the foundation or in the aggregate, may lead to the formation of a tension crack on the under side of the slab. (*c*) In some cases the depth which must somehow be filled is not sufficient to make it worth while to provide a firmly rolled hard-core foundation, the cheaper solution being to employ a relatively low-stressed slab, with a somewhat reduced proportion of cement and relatively light reinforcement. (*d*) Lastly, 7 ins. of fairly tough concrete, with large-mesh reinforcement, is more easily cut through, for excavations, than 4 ins. of very tough concrete with a smaller mesh reinforcement.

Coefficient of Expansion.—(1) *Milan* (A. di Renzo), 53 millionths per deg. F., in the temperature range 76 deg. F. (2) *Harmondsworth*, 78 millionths, in the temperature range 46 deg. F.

Concrete Carriageways : Normal Thicknesses.

(Usually reinforced.)

Great Britain.—A large mileage has been constructed with the thickness of 8 ins., but, for comparable conditions, 6 ins. has often been adopted. It may be considered that thicknesses greater than 8 ins. are excessive. It should be possible, everywhere, so to design and construct that 6 ins. or less will suffice, unless it be more economical to provide 8 ins. depth of poor concrete. For estate roads with very little traffic, 6 ins. is usual in Greater London.

Germany.—Usually about 8 ins., or on old, consolidated crust, 4 to 6 ins. ; usually not reinforced, and seldom double reinforcement.

France.—On good foundations, and for light traffic, 4 to 5 ins. ; on normal foundations, 6 to 7 ins. ; on poor foundations, 8 ins. or more.

Sweden.—Usually 8-6-8 ins. ; also on main roads in Spain.

Switzerland.—About 6½ ins., usually reinforced.

Southern Ireland.—Normal practice, 6 ins.

Reinforced Concrete Slabs : Computations.

Length.

(1) *Ministry of Transport* (Technical Circular).—Where L is the length of the slab in ft. ; f the tensile resistance of the concrete ; F the coefficient of friction (see under ' Experimental Determinations,' p. 552) : w the weight of the slab in lb. per sq. ft. ; d the slab thickness in inches ;

$$L = 12df \div wf.$$

(2) *Empirical Formula* (*R. A. Ryves*).—Where F is the range of air temperature in deg. F., C the range in deg. C., and K a coefficient, 1·0 for full exposure; 1·1 to 1·2 for situations such as hillsides facing south, where the range of ground surface temperature is high in proportion to the range of air temperature; 0·9 to 0·6 for protected situations (0·6 where very well protected from sun and from radiation).

$$\text{Length in feet} = \frac{2500}{KF} = \frac{1389}{KC}$$

$$\text{Length in metres} = \frac{423}{KC}$$

Slab Lengths and Widths.—In Belgium and in Spain the usual length is 10 m.; In Germany, 10 to 15 m.; in Switzerland, 6 to 10 m., the range of temperature in some situations being very great; in France, up to 30 m., but now not usually exceeding 10 m.; in Sweden, usually 18 to 25 m.; in North Dakota, where the range of temperature is 153° F., very short lengths; in Benares, India, 50 ft. was adopted, after 66 ft. had proved satisfactory, the range of temperature being only 50° F. For a normal temperature range of 85° F. in England the corresponding slab length is 29·4 ft., for North Dakota, 16·3 ft.

Lengthwise joints are usually provided for widths exceeding: in France, 7 to 9 m.; in The Netherlands, 6 m.; in cold-winter regions of the United States, 10 ft. In Switzerland it is considered (*R. Dutron*) that it is the anchoring effect of thickened edges which makes lengthwise joints necessary, though these are still usually provided, the individual slabs being 3 to 5 m. (say, 10 to 16 ft.) wide.

Transverse Joints.—Owing to the anchoring effect when there is more than one slab in the width of the carriageway, the transverse joints should be, as in Switzerland, in line or not more than some 16 ins. out of line.

Kingston By-pass Road.—Construction joints at 30 ft. intervals; expansion joints at every 120 ft.

A Swiss Construction.—(H. S. L. Knight.) Bottom course, gravel, 5-mm. to 35-mm. gauge, 58·82 per cent.; sand, 0·5 mm. gauge, 41·18 per cent., by volume; cement, 250 kg. per cub. in. of concrete (421 lb. per cub. yd.). Top course, broken limestone, 20-mm. to 35-mm. gauge, 66 per cent.; sand, 0·5 mm. gauge, 34 per cent., by volume; cement, 350 kg. per cub. m. of concrete (590 lb. per cub. yd.).

German Motorways.—Concrete carriageways are provided on a large proportion of the mileage. The slabs are usually 20 cm. (7·9 ins.) thick, and 6·0 m. to 20 m. (19·7 ft. to 65·6 ft.) long, according to the degree of friction of slab on road bed. Transverse joints at right angles to centre line. Estimate, for the German climate, maximum movement of a slab of 10 m. (32·8 ft.), ± 3 mm. (0·12 in.). Slabs usually in two courses, the lower, about 5¼ in. deep, of gravel concrete; the upper 2¾ in. deep with an aggregate of hard stone chippings. The slabs are usually reinforced at the junction of the two layers, to prevent cracking (Ref. *Bautechnik*, vol. 14, pp. 579 and 586).

Vibration of Concrete.—Investigations carried out by a research committee (Inst. C.E. and Inst. Struct. E.) pointed to the following provisional conclusions, which, it was pointed out, may have to be revised for larger masses of concrete, different types of vibration, or mixtures other than that employed, 1 : 1·8 · 4·2. (*a*) The use of vibration allows of consolidation with a reduced water/cement ratio. (*b*) The acceleration should be above some critical value (with the mixture used, $4g$ for a water/cement ratio 0·40, decreasing to $1·5g$ for ratio 0·60). (*c*) The frequency of vibration is not usually of such importance as the acceleration. (*d*) The time of vibration is, for a particular acceleration, greater for high than for low frequencies, particularly with dry mixes. As regards (*a*) it may be observed that one of the objects of hand vibration is to reduce the water content—in effect, to reduce the water/cement ratio.

Vibration of Aggregate.—From a lecture (Society of Arts) by Dr. R. E. Stradling. On a road construction job, the *plus* or *minus* variation of water content from the mean was: without vibration, 17 per cent.; when the sand was vibrated, 5 per cent.; when both gravel and sand were vibrated, 2 per cent. The variations in the crushing strength of the concrete from the mean were *plus* or *minus*, 50, 18 and 18 per cent. in the respective cases.

Brick Paving Data.

Great Britain.—Brick paving has been tested in Birmingham, on a Staffordshire main road, in Denbighshire, on a Lancashire main road, in Ayrshire, in Doncaster. The British reporters to the Sixth International Road Congress conclude: that brick roads have *advantages* in respect of freedom from undulations, negligible expansion, contraction, and absorption factors, no tendency to lift or float as the result of heavy or prolonged rain, the, as nearly as reasonably possible, non-skid surface; *disadvantages*, in respect of not being so easy to repair and renew as some other block pavements. In respect of essentials the reporters noted important advantages of brick paving and not one disadvantage.

Brick Paving, experimental. Wolverhampton-Walsall main road, concrete foundation, blue brick on edge. North Midlands, (1) Birmingham, wire-cut blue brick on concrete strength crust, floated ½-in. sand-cement: joints grouted 1½ to 1 mortar; laid Sept. 1922; (2) Stone-Newcastle, Class I. main road, very heavy traffic 7-in. to 8-in. 6 to 1 concrete strength crust; 9 ins. by 4½ ins. by 3 ins., common blue wire-cut bricks on edge, laid 1922; (3) Wolverhampton-Stourbridge, Class I. main road, two test lengths laid 1923 and 1925; (4) Walsall-Wolverhampton, Class I. main road, 3,000 sq. yds. blue brick grouted with cement, on concrete strength crust; (5) West Bromwich, Class II. road; exceptionally heavy steel-tyred traffic. From a report by J. E. Swindlehurst, M.Inst.C.E., on inspection in the summer of 1926. Relating to (1) little signs of wear under heavy traffic: (2) speaking broadly, surface good after 3½ years; (3) generally in a good state; (5) indications of wear after 3 years.

Birmingham (H. Humphries).—Laid in 1932. On a 10-in. reinforced concrete strength crust, with reinforcement at 2 in. and 8 in. from the bottom. A ½-in. bed of well damped 1 : 2 cement-sand mortar. Himley bricks laid on edge and 'double grouted' with 1 : 1½ grout. The bricks were laid as close as possible, the joints being about 1/16 in. Traffic includes motor omnibus services and iron-tyred, horsed vehicles.

SPECIFICATIONS : DESCRIPTIONS : COSTS.

With Portland Cement as the Binder.

Concrete Roads, Great Britain.—Replies to a questionnaire of the Technical Advisory Committee. 132 replies, 102 relating to two-course work and 30 to single-course work. *Thickness*, 6 to 8 ins. in 90 per cent. of the cases. *Costs*: in 82 per cent. of the cases between 1*s.* and 1*s.* 9*d.* per sq. yd., per inch of thickness, or, taking 1*s.* 3*d.* and 8 ins., 10*s.* per sq. yd.; 1*s.* 6*d.* and 8 ins., 12*s.* per sq. yd.; 1*s.* 6*d.* and 6 ins., 9*s.* per sq. yd.

Lincolnshire.—A 10-in. slab with upper and lower reinforcement: the lower 8 ins. crushed quartzite aggregate, the upper 2 ins. granite, 12*s.* 4*d.* per sq. yd.

London Area.—Six-inch reinforced slabs, including a layer of ashes, 7*s.* 5*d.* per sq. yd.

Deptford.—Old metalling removed, surface regulated with 3 ins. of rubble; an 8-in. reinforced concrete slab, 1 : 6, Colne River ballast, with a 1-in granolithic surface; 9 ins. total thickness, 10*s.* per sq. yd.

Concrete Road Crust.—Southwark; lower course, contains 18 per cent. of stone ¾-in. to 2-in. gauge; upper course, 3 of crushed gravel to 1 of cement; the gravel, 8 of ¾ in. to ½ in., 17 of ½ in, to ¼ in., 75 of ¼ in. to dust, per cent. Such a crust, on roads with 22,000 tons of traffic per week, is treated with silicate of soda, with good results.

Concrete Road Crust.—Epsom R.D.C., Class B road, 2,752 tons per day; reinforcement to have ultimate resistance across the road of 12,500 lbs. per foot; slab, 7 ins. of shingle concrete finished with 2 ins. of granite concrete; shingle portion: aggregate, by weight, 54 per cent. passing 1½ to ⅜-in. square mesh, 24 per cent. passing 3/16 square mesh, 22 per cent. passing 3/16 mesh; sand, all passing 3/16, not more than 5 per cent. passing 100 mesh sieve; cement, rapid-hardening ('ferrocrete'), 370 lbs. to each wetted and mixed cubic yard of concrete; surface layer, 1½ in. dolerite containing sufficient ¾-in. to ⅛-in. chippings to reduce the voids to 30 per cent. by volume, of this 3 parts; sand 1 to 1½ parts; rapid-hardening cement, 550 lbs. per cubic yard. This cost 9*s.* 8½*d.* per square yard for carriage way 20 ft. wide. Cost, including excavation and surface treatment with bitumen, £9,086. per mile.

Cement-bound Macadam.

Southern Ireland (J. Caffery and E. J. Duffy, Trans. Inst.C.E.I., vol. lviii.).—Grouted crusts, fully satisfactory, can be made at costs 5*s.* per sq. yd. less than those of pre-mixed concrete. Normal consolidated thickness, 7 ins.; transverse joints, 30 ft. apart.

Quantities and Costs.—Great Britain. A crust 4 ins. thick, made by the sandwich method. Per 100 sq. yds.: stone, 2-in. B.S.S. gauge, 15 tons; sand, 4 tons; cement, 1¼ tons. Typical cost: per 1,000 sq. yds., stone, 150 tons at 12*s.*, £90; sand, 4 tons at 6*s.*, £12; cement, 18¾ tons at 50*s.*, £46 17*s.* 6*d.*; total for materials, £148 17*s.* 6*d.* Labour, including use of roller and mortar mixer, and water supply, at 7*d.* per sq. yd., £29 3*s.* 4*d.*; incidentals, such as timber, fencing, and lighting, at 2*d.* per sq. yd., £8 6*s.* 8*d.* Total cost, £186 7*s.* 6*d.*, or 3*s.* 9*d.* per sq. yd., with a trained gang of 16 men, laying 400 sq. yds. per day. The estimate does not include any preparation (Brit. Portland Cement Assoc.).

Cement Grouting, Australia (W. T. Sunderland).—(1) Transverse joints are sometimes provided, the spacing being 50 or 60 ft. Stone, 95 per cent. of it 2½-in. gauge; grout, 1 : 2½ with 1½ of water; depth before rolling about 8 ins. Steel reinforcement, laid on the top of the first layer, is sometimes employed. The cost has been about 7*s.* to 9*s.* 2*d.* per sq. yd. The ratio is 1 : 2½ : 12, comparing with 6-in. 1 : 2 : 4 concrete in Melbourne, costing 14*s* per sq. yd.

(2) *Items of Cost in £.*—Relating to (1), 2,141 sq. yds. on a sand foundation: formation and metalling, 323·3; rolling, three days, 10·5; laying and fixing reinforcement, 4·45; No. 14 B.R.C. fabric, 2,231 sq. yds., 130·15; sand, 2,495 cu. ft. bags, at 5*d.*, 62·35; cement, 998 cu. ft, bags, at 4*s.* 6*d.*, 224·55; labour in grouting, 3 days, 22·55; petrol and oil, 0·8; interest and

depreciation on the plant, at 4·0 per week, 2·0 : supervision, 5 per cent., 39·55 ; royalty on process, at 1½*d.* per sq. yd., 13·4 ; total, £833·55, or 7*s.* 9·4*d.* per sq. yd. The labour cost (wages 14*s.* to 16*s.*) was nearly 2*s.* per sq. yd.

The Sandwich System (*R. D. Jackson*).—(1) Practice in Jersey. Timber forms with iron-plated tops were used to give 5 ins. thickness in the middle and 3 ins. at the sides of the road. The stone is rhyolite, or a close-grained granite, of 2-in. gauge. Costs : about 4*s.* 8*d.* per sq. yd.

(2) *At Kirkcaldy* (*A. Walker*).—The road is scarified to 3 ins. if necessary ; rolled and formed to 1 in 32 ; a 2½-in. layer of whinstone, 1½-to 2½-in. gauge, is rolled, the roller passing twice over every part ; a mortar, 2 parts of clean sand to 1 part of rapid-hardening cement, spread to about 1¼ in. thick ; a 2½-in. layer of stone spread over it ; the whole rolled until the mortar comes to the top ; surplus slurry brushed forward. This forms a good holding surface for horses. For 200 sq. yds., 26 tons of cubical stone, 1¼ to 2½ ins., 5 cu. yd. sand, 52 cwt. rapid-hardening cement. Cost of the surfacing only, 3*s.* to 4*s.* 6*d.* per sq. yd.

Bituminous-Bound Constructions.

Top Courses and Carpets.—British report to Fifth International Road Congress : bituminous and asphaltic work ; ordinary costs, with five years' maintenance included : 2-coat work, 4 ins., 11*s.* 6*d.* to 13*s.* ; single-coat, stone and sand aggregates, 3 ins., 9*s.* to 10*s.* ; 2½ ins., 8*s.* to 9*s.* 3*d.* ; 2 ins., 6*s.* 6*d.* to 7*s.* 6*d.* ; clinker, 2-coat, 3 ins., 7*s.* 6*d.* to 8*s*. 8*d.* ; single coat, 2 ins., 6*s.* 3*d.* to 7*s.* 3*d.* ; mastic, 2 ins., 8*s.* 9*d.* to 10*s.* 6*d.* ; compressed rock asphalt carpet, 12*s.* to 15*s.*

Coarse-graded Bituminous Concrete (*T. W. Allen*).—United States practice. Aggregate : passing 1½ in. and retained on ½-in. screen, 30 to 60 ; passing ½ in. and retained on 4-mesh screen (*sic*), 15 to 25 ; passing 4-mesh and retained on 10-mesh screen, 5 to 15 ; passing 10-mesh screen, 20 to 25. The part passing the 10-mesh sieve (*sic*) when tested separately has the grading : 10–40, 15 to 40 per cent. ; 40–80, 22 to 50 per cent. ; 80–200, 15 to 20 per cent. ; passing 200, 10 to 15 per cent. Asphaltic cement, 8 to 12 per cent.

Asphalt.

Two-coat Asphalt.—Binder course, 2-ins. thick ; stone, ½-in. to 1½-in., 69 per cent. ; sand, 23 per cent. ; fluxed Cuban asphalt, penetration 60/65, 8 per cent. Wearing course, 1 in. ; stone, ⅜-in. to 1½-in., 30 per cent. ; filler, 10 per cent. ; sand, 48 per cent. ; fluxed Cuban asphalt, penetration 40/45, 12 per cent. (See next item.)

Single-coat Asphalt.—Three inches thick ; ¾ in. to 1 in., 45 per cent. ; sand, 36 per cent. ; filler, 9 per cent., fluxed Cuban asphalt, 60/65 penetration, 10 per cent.

In both cases the specification includes : the final operation is rolling in ½-in. or ¾-in. chippings, coated with 2 or 3 per cent. of 70/77 penetration bitumen.

Clinker Asphalt Paving (Sheffield : *Mr. W. J. Hadfield*).—Hard-burnt destructor clinker is used, crushed and screened ½-in. mesh for the bottom coat and ¼-in. mesh down for the top coat. The rejections are used for ordinary tar macadam. The material is heated to 300° C. There is about 10 per cent. bitumen in the bottom coat and 17 per cent. in the top coat. The bottom coat is spread to rather more than 1 in. in thickness and slightly rolled. The top coat has a thickness of rather less than 1 in when rolled, giving a total of 2 ins. Rolling is continued till no further impression can be made on the material. A 12-ton roller is preferred.

Granite Asphalt.—(1) London-Folkestone Road : consolidated thickness, 3 ins. Stone, graded granite, 1½ in. to ½ in., 54 per cent. ; graded silica sand, all passing ½-in. mesh sieve, 28 per cent. ; filler (80 per cent. passing 200 mesh sieve), 8 per cent. ; Mexican bitumen to British Standard Specification and requisite penetration 10 per cent. (2) Kingston By-pass Road : Length No. 10. Cold asphalt. Percentages : binder course, 1½ in. thick ; granite, ¾ in. to ½ in., 35·8, ⅜ in. to ½ in., 34·5 ; coarse sand, 19·2 ; bitumen emulsion, 10·5. Wearing carpet, ¾-in. thick ; granite, ¾-in. to ½ in., 48 ; coarse sand, 17·8 ; fine sand, 15·0 ; filler, 5·5, bitumen emulsion, 13·7.

Tarmacadam.

Tarmacadam—Quantities of Tar (*J. C. Mann*).—Gallons per ton : for stone of 2-in. to 2½-in. gauge, 7 to 8 gals. ; 1½-in. gauge, 7½ to 9 gals. ; ½-in. gauge, 10 galls. ; ¾-in. gauge, 11 gals.

Laying Tarmacadam (*J. Young*).—Bituminous tarred macadam : stone gauge 1½ to 2 ins., usually a course 2½ to 3 ins. thick ; the tarred stone, after being laid, is covered with about 1 in. of ¾-in. tarred chippings, the road being then rolled, after which the surface is brushed over. The road is then sprayed with a hot composition of tar and bitumen, 60–40, or a cold coat of bituminous mixture is brushed in, and the surface coated with a mixture of dry limestone, slag, and whinstone chippings, ½ in. to ¾ in., and lightly rolled.

Tar-grouted Macadam (*E. W. Cone*, verbatim).—A grouted macadam road is constructed by laying approved mineral aggregate upon a prepared foundation to a sufficient depth to provide a consolidated thickness of 3 ins. after dry rolling. Tar is then applied by hand or pressure, the Surface being finally covered by chippings. The requirements as to the permissible size of the aggregates are not very rigid, but the objective should be to secure a structure of uniform moderate void content after dry rolling. If the surface is too dense, difficulty will be experienced in getting the tar to penetrate into it, and if it is too open, the tar will collect at the bottom. As it is desirable to have as great mechanical stability as can be obtained, it is better practice to use coarse aggregate (2½ ins. and 1½ ins.), and if the voids are large, they can be partially filled with chippings during the dry rolling.

The tar should comply with the British Standard Specification for Tar, No. 3, and should have a viscosity of between 180 and 240 secs. It should be heated to a temperature between 250° and 300° F., and applied evenly and steadily to the rolled dry aggregate, 1 gal. being used to treat each sq. yd.

After the application of the tar the surface should be covered immediately with hard, dry chippings and should be rolled until there is no further movement. The construction of a grouted macadam road requires the utmost care in every detail, if the best results are to be obtained.

Recent Costs of Tarmacadam.—(1) Per sq. yd.: 3½ ins., 3*s.* to 3*s.* 6*d*; for moderately heavy traffic, the surface so finished that periodical spraying is not necessary, 4*s* 6*d.* to 5*s.*; in London, 3 ins., 6*s.* 8*d.*, 3½ ins. limestone tarmacadam, 5*s.* 7*d.* (2) Midlothian: Mixed at the quarry and taken direct to the roads: wearing courses, 2*s.* 6*d.* to 3*s.* 3*d.*, according to the distance from the quarry.

Grouted Work.

Hot Grouting (G. S. Barry).—With tar No. 2, or with a compound of tar and bitumen. The metal is spread over the old road surface at 9 sq. yds. to the ton, and rolled until there is very little movement of the stones. Chippings of ¾-in. size are brushed into the interstices and rolling continued until the coat is fairly rigid. The tar No. 2, or compound, heated to 250° F., is poured over the surface by means of cans with flat nozzles.

The application is best done in strips first in one direction and then in the opposite direction. This action gives the hot material a better chance to penetrate into the coat. It is important that the metal is dry, and that the work is not done during damp weather. The tar, or compound, is applied at from 1 to 1½ gals. per sq. yd. Thereafter the surface is spread with more chippings, which are brushed into the interstices and then rolled.

The surface may now be left for a few weeks, after which the sealing is done. This consists of brushing off all loose chippings and applying the heated tar, or compound, at or about half a gallon per sq. yd., and spreading ½-in. or ¾-in. chippings, well rolled. The cost per sq. yd. ranges from 2*s.* 9*d.* to 3*s.* 3*d.* (Scotland), and the method is suitable for roads carrying 2,000 to 3,000 tons per day.

Cold Emulsion Grouting.—The methods, as regards the laying of the stone and procedure generally, are very similar to those employed in hot grouting.

Penetration Work with Emulsion.—(Ayrshire). The metal, of 2½-in gauge, was lightly rolled and sprayed with slow-setting emulsion at 1½ sq. yd. per gal. Next day it was thoroughly rolled. After a week it was sprayed with ordinary emulsion at 2½ sq. yd. per gal., covered with ¾-in. whin chippings and again rolled. The cost, including the scarifying and shaping of the old worn road, was £85 11*s.* 5*d.* for 1,956 sq. yds., or 10½*d.* per sq. yd. Unit costs were: emulsion, slow-setting, 7½*d.*, ordinary, 5½*d.* per gal.: 2½-in. road metal, 5*s.* 2*d.* per ton, ¾-in., 10*s.* 1*d.* per ton. Items: roller, £2 0*s.* 10*d.*: haulage, £7 10*s.* 0*d.*; labour (155 hours at 1*s.* 3*d.*), £9 13*s.* 9*d.* Of each gauge of road metal, 24½ tons were used.

Quantities of Cold Emulsion (H. J. Prentice).—Grouting work. Gallons per sq. yd. Full grouting: for stone of 2-in. gauge, 1 to 1¼ gal.; 2½-in., 1¼ to 1¾ gal.; 3-in., 1½ to 2 gals. Semi-grouting: for stone of 2-in. gauge, ½ to 1 gal.; 2½-in., ¾ to 1¼ gal.; 3-in., 1 to 1½ gal.

Analyses of Costs.

Oxfordshire (C. E. Dolphin).—Cold emulsion work. Surfacing main roads with field flints and gravel, the flints from the Chiltern Hills, the gravel from the Thames Valley; flints, 3-in. downwards, hand-picked from fields, cost 5*s.* per cu. yd., laid to depth of 3 ins. on the scarified road crust, all shapes together; watered and rolled; bound by grouting with cold emulsion, ¾ gal. per sq. yd., or 1 gal. per 1⅓ sq. yd.; covered with ⅜-in. shingle, 160 sq. yds. per cu. yd.; traffic for a few days, then a sealing coat of emulsion, ¼ gal. per sq. yd.; blinded with fine sand and rolled. Costs, per sq. yd.: flints 8·5*d.*, shingle 0·75*d.*, sand 0·54*d.*, roller and horse hire, 2·41*d.*, labour 0·3*d.*, grout 8·25*d.*, sealing coat 2·75*d.*: total, 2*s.* 2·2*d.* A similar but cheaper process is also employed. For roads carrying up to 600 tons per day, a gravel, shingle, sand, grout, sealing coat road costs 1*s.* 11·3*d.* per sq. yd.

Road Crusts for Steep Gradients.

Bituminous-bound Slag.—Has been provided on gradients of considerable length, 1 in 14 and 1 in 12.

Bituminous-grouted Stone.—Cornish granite, no sealing coat, has been provided on a gradient of 1 in 6.

***Bituminous-bound Destructor Clinker.*—Muswell Hill: a long gradient of 1 in 10; varied traffic, including London omnibus services.**

Tarmacadam.—Aberdulais-Resolven Road, 1 in 11. Has been laid on gradients of 1 in 6.

Cement-bound Crusts.—For (*a*) considerable lengths of gradient exceeding 1 in 10; (*b*) any gradient on which parts of some length are 1 in 8 or steeper; (*c*) any gradient parts of which, however short, are 1 in 6 or steeper: most surveyors prefer cement-bound crusts, the trend of practice being in favour of the sandwich method. On a gradient mostly 1 in 6, a portion 1 in 4 has been consolidated by tamping.

Cement-grouted Wearing Course.—A wearing course 2½ ins. consolidated thickness, grouted with a slurry of 1 : 2 cement and sand, well swept in. The surface was then brushed to leave the stones standing proud, and was tarsprayed. Cost, 2*s.* 6¾*d.* per sq. yd.

Concrete (*O. Barker*).—Brow Hill, Haworth: 2-course concrete, 1 : 2 : 4; bottom course graded 1½ ins. down; top course 2 ins. B.S.S. gauge. When tamping was completed, the surface was brushed with stiff brooms, leaving the stones exposed and slightly proud.

NOTES FROM BRITISH AND OVERSEAS PRACTICE.

Dry-bound Macadam (*G. S. Barry.*)—In Scotland. The success of this method may depend upon the formation of stone powder in the period before rain falls, a powder fine enough to form, with water, a paste which hardens into a cement. (*The Surveyor.*)

Cement-Lime-Concrete.—Hyderabad (Report No. 9, S.I.R.C.). Laid on each side of a 19 ft. middle strip of the carriageway made of plain cement concrete. Proportions, cement, lime, sand, road metal: 1 : 1 : 5 : 8.

Clay-bound Gravel.—Michigan (W. O. Dow, 'Good Roads,' May 1934). A mixture of clay and gravel laid to a depth such as 4 ins. to give 3 ins. consolidated wearing course. A *fully wet process* is described, found effective in remedying the soft and unstable condition of the course when awaiting rain after it had been laid.

Bamboo-Reinforced Concrete.—China (C. H. Chang and H. Tsai, Report No. 4, S.I.R.C.). In slabs 18 cm. thick, bamboo reinforcement was placed in various ways. Transversely, the rods were 20 cm. apart and, when longitudinal rods also were placed, these were 8 cm. apart. The rods were 1 to 1½ cm. square, placed 5 cm. from the bottom of the slab. The following values were found by experiment.

Bamboo.—Ultimate strengths in lbs. per sq. in., compressive, 5,500; tensile, 14,000; bending, 13,000; in shear, 450. Modulus of elasticity, 1,660,000. The outer layer of the stem of a bamboo has a tensile strength of at least 25,000 lbs. per sq. in. ('Ligno-concrete' was described by G. O. Case in *The Engineer* many years ago.)

Strip Roads.—Southern Rhodesia: Concrete strips 2 ft. wide and 2 ft. 9 in. apart between inside edges; thickness, 6 in.; mixture, 1 : 2½ : 5, somewhat wet, the trench being previously soaked with water. In lengths of 30 ft. with ½-in. felt joints. Hardened by two coats of silicate of soda. Cost, 800*l.* to 1,000*l.* per mile. Traffic capacity 250 to 300 vehicles per day. Described in a paper by S. Chandler, chief engineer, Roads and Bridges, South Rhodesia Government (Extracts, *The Surveyor*, June 14, 1935).

Heating Blacksoil Road-beds (*L. H. R. Irvine*).—Described in a paper, Inst. of Engineers, Australia (see 'Roads and Road Construction,' October 1, 1931). The 'blacksoil' seems to be similar to the 'black cotton soils' of India, which contain a very large proportion of colloid clay and a relatively large content of humus. A trial of the method is being made in Queensland, on the Brisbane-Toowoomba Road. The process is, essentially, one of heating the road-bed until the soil cracks, forming interlocking paving elements of brick, the joints between which are then filled with bitumen.

Silicated Road Crust.—French practice (Jacques-Thomas): aggregate hard or semi-hard Jurassic limestone or similar porous limestone; gauge, passing a ring of 1½ in. to 2 in. diam.; coarse sand (the finer kinds, having a tendency to form pasty mortars, are to be avoided). About 40 litres of silicate and ½ cub. m. of sand to each cub. m. of broken stone; say, 6¾ gals., ½ cub. yd., 1 cub. yd. Maximum, of silicate per cub. m., 55 litres, say 9½ gals. per cub. yd. Proportion of silica to soda in the solution, 3·0 to 3·5, the lower figure for porous materials. A 4-in. layer of broken sandstone mixed with silicated limestone contains, per cub. m., 45 litres of silicate of soda and 300 litres of sand (by volume, 4½ : 30 : 100). Hydraulic lime is added to the sand in wet weather. This layer is rolled until the viscid mud exuding from it is ready for the top course of broken sandstone, 1½ to 2 ins. thick. After a month, the surface is treated with emulsion. Such road crusts are provided on roads in very damp situations.

Gloucester By-pass Road.—A bottoming of ashes, 3 ins. deep; a bare, unreinforced concrete slab, laid for full width, on the alternate bay system, bays 18 ft. 6 ins. long laid first, followed 2 or 3 days later by the 9-ft. bays. The slab is in two layers, the bottom layer 5 ins. thick, 6 parts Frampton gravel, one part rapid-hardening cement; the top layer 3 ins. thick, 1 : 1½ : 3, rapid-hardening cement, Bideford sand, ¾-in. to ⅛-in. Tytherington limestone. Butt transverse joints at 60° to the kerb, except that the 2 ft. nearest the kerb is at 90° to it, to avoid a corner of less than 30°. Camber, 1 in 40. A ⅜-in. bituminous joint between the slab and the kerb.

Mixed Roadstones.—(1) Stroud Rural District: granite and limestone in equal proportions for all work when chippings are required for road surfaces. (2) Stalybridge · pink granite (such as Shap granite) 3 parts; Scotch granite (milky-white or grey) 1 part; 'blue granite' (Derbyshire or Welsh basalt) 2 parts; Derbyshire grits, 1 part.

Rolled Concrete Road Crusts.—(Victoria Country Roads Board, Australia; Mr. L. F. Loder, chief engineer.) Mixtures of 1 : 2½ : 10 or 1 : 2½ : 12 can be employed, the voids to be filled by the mortar being in a proportion less than in ordinary concrete. It seems that vibration is employed as an alternative to rolling.

SURFACE TREATMENT.

Tar for First Treatment.—Successfully followed for some years past in France, the practice of using tar alone for the first dressing of a water-bound or dry-bound broken-stone crust is now normal practice in Scottish and many English counties.

Hot Tarring and Gritting.—Relating to 375 miles of roads in Hampshire. Six sq. yds. per gal. on water-bound surfaces, about 16 sq yds. per gal. on tarmacadam. Partly on tarmacadam and partly on other surfaces, some previously tarred, the general average cost, including gravel and granite chips, was 3·16*d*. per sq. yd.

Spraying Material (J. C. Mann).—The viscosity of the spraying material used must vary with the size of the chips. Where a 15/25 viscosity tar would make a satisfactory job with ¼-in. or ⅜-in. chips, it would fail if ¾-in. to 1-in. chips were used. The recommended scale is:

Viscosity and Chip Size.—Chips ¼-in. to ⅜-in., 15/25 seconds tar; ½-in. to ⅝-in., 25/35 secs.; ¾-in. to 1-in., 40/60 secs. The high viscosity tars give results amply repaying some trouble experienced with the barrels. It is recommended:

Overheating.—A temperature of 180° F. should not be exceeded with tar, and it is best to keep it between 150° and 160° F.

Cold Emulsion and Gritting.—Hampshire: 17½ miles of roads, average cost 3·73*d*., or, not including sealing coats on semi-grouted surfaces, 3·77*d*. per sq. yd.

Surface Treatment. Costs per sq. yd.:—

Northamptonshire.—1944–45: 'Refreshment of road surfaces with tar and chippings.' Classified roads, 5·10*d*.; unclassified, 4·11*d*.

Holland.—1944–45: 'Tar-sprayed,' 7·22*d*. 1945–46, surface treatment: classified roads, 4·39*d*.; unclassified, 4·12*d*.

Somerset.—1947–48: surface dressing, 5·94*d*.

ANALYSES OF TARRING COSTS.

Dorking.—Tar-spraying: manual labour, 17·3 per cent.; team labour, 16·7 per cent.; tar, at 7½*d*. per gal., 33 per cent.; grit (at about 1 ton per 30 gals. of tar, costs 14*s*. 1*d*. per ton), 25 per cent.; tradesmen's bills, 8 per cent.

Consett, Durham, 1929 (*T. Hutton*).—Hot work: tar at 5½*d*. per gal., 5 sq. yds. per gal.; limestone chippings at 14*s*. 1*d*. per ton, 127 sq. yds. per ton; cost: chippings, 1·34; tar and bitumen, 1·12; labour, 0·72; transport, 0·21; plant, etc., 0·05; total, 3·44*d*. per sq. yd.; Cold work: emulsion at 9*d*. per gal., 4½ sq. yds. per gal.; chippings, 115 sq. yds. per ton; cost: chippings, 1·44; emulsion, 2·31; labour, 0·52; transport, 0·21; plant, 0·05; total, 4·53*d*. per sq. yd.

FOOTPATH CONSTRUCTION.

Ayr Burgh (J. Young).—All footpaths are now laid as *in situ* concrete pavements in liftable panels conforming to the sizes of standard concrete flags; to the following:

Specification.—On 4 ins. thick of broken stone bottoming, blinded and rolled, covered with 1 in. of sand, is laid cement concrete composed of ¾-in. chippings and cement (3 to 1) 2 ins. thick; well beat down, on top of this is laid, while the under-bed is moist, 1 in. thick of ⅛-in. to dust granite chippings and cement, and the upper surface straightened and smoothed off, margin lines formed and the panel surface indented with a brass roller. Before the concrete forming the slabs is put in, strips of iron 3½ ins. by ⅜ in. are placed round the panels transversely and longitudinally so as to divide the panel into slabs as near as possible 3 ft. long by 2 ft. wide, the joints breaking bond in the courses. These dividing strips of iron are slightly tapered in section and have handles fixed to the upper edges, and can be readily lifted, leaving clean joints the full depth of the slab. After the cement slabs are thoroughly set the joints are filled in flush with lime and mortar, well packed and brushed in.

Midlothian.—New footpaths provided on main roads are coated with tarred gravel, at a cost of 1*s*. 6*d*. to 2*s*., usually at the end of the year, after the road work is completed. Subsequent treatment, for maintenance, spraying with cold emulsion and strewing with clean sand or ¼-in. gravel.

Tarmacadam Footpaths.—Market Harborough: reconstruction; new materials, tar, ashes for bedding, and slag dust for top-dressing. A portable 'cracker' used for breaking up the old material. Ten gals. of tar mixed with 7 barrows of the broken material, by hand; rolled with

an 8-cwt. roller. Cost per sq. yd.: labour 6*d*.; haulage, 1½*d*.; tar, at 7½*d*. per gal., 5½*d*.; slag dust, ½*d*.; petrol and oil, ½*d*.; surface tarring, 3½*d*. Total, 1*s*. 4½*d*.

***Ayrshire*.—More especially a method adopted on roads near towns and villages. Excavation as deep as is necessary, usually 4 to 6 ins. Refilling with ashes which, after being rolled until firm, are left for from 4 to 6 weeks before the path is trued up. It is then sprayed with bitumen at 3 sq. yds. per gal., covered with ⅜-in. chippings, and rolled. Cost, 1*s*. to 1*s*. 2*d*. per sq. yd.**

ROAD MACHINERY AND PLANT.

Machines for Concrete Roads.

Based on the paper, 'Equipment for Concrete Carriageway and Cycle Path Pavement Construction,' by C. M. Willcock and N. E. Kerridge (Public Health and Municipal Engineering Congress, November 1948.

Spreading Machines.—All these machines, also finishers, are arranged to run on the tops of the forms containing the concrete or on rails which are mostly attached to the forms, the latter practice being preferred. To deal with concrete slabs of different widths, these machines are made telescopic or are made adjustable by other means. On both spreaders and finishers provision is made for cross flow of the concrete. Both are most often driven by diesel engines, sometimes by petrol engines.

The Blade Type.—For this type of spreading machine the concrete is dumped on to the subbase between the forms and is distributed across the full width by means of a blade which moves to and fro. The incident face of the blade is inclined at about 45° in plan, so that it moves the excess concrete forward as well as across. Immediately behind the blade there is a striking-off plate which spans the full width. The bottom edges of both blade and striking-off plate are adjustable to about the same height, and that is the level to which the concrete is spread. The machine travels forward into the heap of concrete as the blade moves from side to side.

Finishing Machines.—This machine comes into operation after the concrete has been evenly spread—preferably by a machine, a sufficiently slight excess of concrete being left by the spreader in front of the screed of the finishing machine to allow it to perform its first function of striking off the concrete immediately in advance of the compacting beam. Machines for use on dry-mix have two beams; the screeding beam which oscillates across the width between the forms to produce a level surface for the compacting beam, which, by means of an intense vibration compacts the concrete and leaves a suitable finish. The beam has a either a rotary vibration or a purely vertical vibration. The acceleration varies between 4½ g. and 10 g., at frequencies of between 3,000 and 4,000 cycles per minute.

The amplitude is up to $\frac{1}{64}$ in. The different mixes of concrete used, and variable weather when the concrete is laid make it desirable that machines should have a variable travelling speed.

Pressure Grouting.—A recent application of the method is described in an article by C. K. Tallach, County Surveyor, Lindsey (*The Surveyor*, November 12, 1948). The concrete slabs are 10 ins. thick, 22 ft. wide and 16 ft. 6 ins. long. Some 40 slabs had settled, 19 of the worst being raised. The grout consisted of 1 part of rapid hardening Portland cement and 3 parts of building sand. Two-inch injection holes were drilled through the slab and about 6 ins. into the sub-base and injection made through steel tubes inserted in the drilled holes, sealed with paper packing and held by wooden wedges. Other drilled holes through which the grout began to escape were sealed with wooden plugs. They gave indication of the travel of the grout under the slab. Injection was made at one hole at a time. Holes were not drilled nearer to edges or joints than 2 ft. The pressure was usually 50 lbs. per sq. in., but that had, in some cases, to be exceeded. When the grout very soon began to escape at edges or through other drilled holes, injection was stopped until the grout had hardened. The next injection, though usually more difficult, and requiring greater pressure initially, moved the slab. At three sites, the maximum settlements and the areas of slab settlement were: 1·8 ins., 315 sq. yds.; 4 ins., 135 sq. yds.; 6½ ins., 405 sq. yds.: and the grout used, tons: 27·95, 22·55 and 72·65. Costs per sq. yd. were: 23·3*d*., 38·2*d*. and 28·5*d*. Tons of grout used per cu. yd. of lift: 4·04, 3·05, 2·16.

Note.—In essentials, the operation is the same as that of Poulter (see p. 569), and when the 'sub-grade' is not a laid course of material, but is a roadbed of firm earth, mud with a small proportion of cement may be a fully satisfactory grout.

Excavating and Earth-Moving Machines.

Victoria, Australia (*L. F. Loder.*)

***Trailbuilders.*—The machine consists essentially of a blade, say, 10 ft. wide by 3 ft. 3 ins. deep, with cutting edges along the bottom and both sides. Using 60 drawbar b.p. machines the cost of getting and placing earth on an average lead of about 120 ft. was 7*d*. per cu. yd. On some sections of the Kiewa Valley Road where the work was almost 'pure side cut,' the cost of making an 18-ft. formation was about 3*d*. per cu. yd.**

Self-loading Scrapers.—Costs vary greatly, but as a guide the following figures are given: 7 to 9 cu. yd. four-wheeled scoop, 1,000 ft. lead, 9*d*., 250 ft. lead, 5*d*. per cu. yd.; 5 to 6 cu. yd. two-wheeled scoop, 1,000 ft. lead, 1*s*.; 250 ft. lead, 7*d*., per cu. yd.; for solid measure and exclusive of overhead charges, but including all plant charges, both capital and operating.

EARTH-MOVING MACHINES.

The Drag-Shovel.—This single-bucket excavator combines certain advantages of the shovel and the dragline.

Loaders.—Have been replaced to a large extent by small power-shovels, some of which are built around the power unit of agricultural tractors.

Bulldozers and Bullgraders.—These machines have practically replaced the older type of grader. They are becoming indispensable on many excavating jobs and can move earth lengthwise for distances of 100 ft. to 200 ft. more cheaply than other equipment.

Scrapers.—The most popular type in Great Britain is the machine which dumps its load between its four wheels, and is efficient for hauls of 300 to 1,500 ft. or, subject to careful calculations, for hauls up to 3,000 ft.

Crawler Excavator.—Weight, 75 tons; tools, shovel, dragline, grab, back-acting trencher, skimmer-scoop, crane. Petrol-paraffin engine, 4-cyl., 32 b.h.p. (or steam, or electric), 1 to 1¼ gals. of paraffin per hour. Three digs per minute when working shovel.

Roadside Excavator.—For road-widening work: cut, 18 ins. wide up to 2 ft. high; speed, 30 lin. yds. per minute (presumably average soil). May be adjusted to work below road level, or above (up to 18 ins.) and below at the same time. Can be fitted to any tractor or roller.

Elevating Grader.—Equipped with a 65-h.p. engine and with power controls, and drawn by a 95 h.p. tractor; output, when loading into large wagons, 400 cu. yd. per hour.

Carrying Scraper.—One 13-cu. yd. scraper drawn by a 95 h.p. tractor can move earth at the following rates: length of loaded haul, 400 ft., cu. yd. per hour compacted load, 115; as 400–115, also, 600–92; 800–76; 1,000–65; 1,200–57; 1,400–50; 1,600–45; 1,800–41; 2,000–38.

OTHER PLANT.

De-aerated Concrete Mixers.—Have been employed on road works in Surrey. In place of the standard drum, a cast-iron drum is provided, having a machined face to ensure airtight contact between the drum and the mixer door, which consists of a light steel pressing mounted on ball-bearings. A 26-in. vacuum can be obtained in half a minute.

Road Surface Heaters.—The heating of bituminous road surfaces facilitates the removal of worn materials, renders surfaces from which material has not to be removed more amenable to treatment for the re-dressing of inequalities, and may be employed, in some processes, for laying carpets of bituminous materials. One type of machine heats air in a furnace, the hot air being delivered under a hood, beneath the edges of which, close to the road surface, it escapes. Progress is by stages, the machine being stationary during the heating. In another machine the heat is radiated from a hot plate, and the machine crawls forward at the pace necessary to allow of the road surface being adequately softened.

A recently introduced heater, in which the low-pressure, air-forced draught system is employed, is rated at 80 to 100 sq. ft. per hr. It is designed for either direct flame heating or indirect heating. The hood is 4 ft. 6 in. long by 4 ft. wide. The burner is of the pressure-jet type.

Road heaters are (1948) available with single, twin, or multiple burners, the fuel usually being light diesel oil or gas oil. Some very small units burn paraffin. In one type, the heater is suspended from the overhanging frame of a self-propelled vehicle resembling a small motor lorry.

Poulter's Mud Pump.—Successfully employed in the United States in raising sunken road-slabs to their proper levels. The process is interpreted (*The Surveyor*) as one by means of which force is exerted to move a part of a structure, compressing, in the reaction, plastic materials enclosing the operating space, and filling hollows with cheap material. A small proportion of cement mixed with the mud greatly expedites the setting of this material.

Toronto.—A portion of reinforced concrete pavement, 1,000 ft. long by 46 ft. wide and 8 ins. in thickness, having settled transversely, was brought up 1¾ in. without cracking, by this method, but using cement grout and air pressure. Poulter's initiative is, perhaps, as significant as regards the material as in respect of the method.

Power Cost Data (E. Dalton).—Quarrying: oil-engine compressor, drills, etc., 100 h.p.; fuel oil nearly 0·4*d*. per brake horse-power-hour: average load factor, 50 per cent.; fuel oil and lubricating oil, 0·75*d*. per ton of rock quarried, the calorific value of the fuel oil being 19,400 B.Th.U., and lubricating oil 3*s*. 6*d*. per gallon.

PLANT AND MACHINES FOR BITUMINOUS-BOUND ROADS.

Mixers for Bituminous Materials.—Modern units are either trailers, four-wheeled if large, two-wheeled if small or are self propelled. The usual fuel is paraffin, light diesel oil, or gas oil. Mechanical stirrers, usually power operated through a safety drive worm reduction gear, provide for thorough mixing.

Storage Tanks.—The best storage tanks for binder are electrically heated, the heating being thermostatically controlled.

Spreader-Finishers.—The use of spreader-finishers effects important saving of time and cost. In the year 1944–45 the Barker-Greene machine was introduced in Leicestershire. Employed on two road works, its outputs were 169 and 183 tons of bituminous material laid per day, saving 2¼*d.* per sq. yd. in the cost of laying.

In the years 1945–46, nearly all the tarmacadam surfacing laid in Northamptonshire was spread and consolidated by machines.

Spreader-finishers handle successfully all types of mixes. The advantages of tamping finishers over machines that spread and level only are described (E. L. Harber) as that they will lay material with uniform density at varying thicknesses, eliminating depressions due to uneven compaction and reducing the amount of rolling required; also that they effect a well bonded joint even against cold material of the temperature of the material laid produces a workable mixture. The tamper has a ⅛-in. vertical stroke. The screed directly behind the tamper slides on to the mix as it is struck off and compacted by the tamper, smoothing and ironing the surface for final compaction by rollers; an operation reduced and simplified by the uniform density obtained with the tamper and the level surface obtained by the operation of the screed and levelling arms.

Developed, in the United States, over a period of seven years, the Barber-Greene machine was put on the market in 1937. To-day its merits are generally recognised by road engineers.

Road Planers.—It is usual practice to heat and plane a considerable stretch of road at one time. The material removed may be used for making up foot paths and other purposes. By planing, the corrugations are *eliminated*, so that a new carpet laid on a planed road retains a true running surface for a long time; which is not likely to be the case when the corrugations are for a time *obliterated* by a confirming carpet of accommodating thicknesses.

Planers usually have power-operated blades with an oscillating action, and are adjustable as to height and camber by power-operated rams. More than one cut over a heated surface may be needed to obtain the desired result.

At present most heating and planing work is done by independent heaters and planers, but manufacturers and others are giving consideration to machines which combine both operations, the machine being a single unit mounted on a chassis, self propelling or trailed. (J. E. Hobbs—see 'Literature,' No. 11, p. 572.)

ROADSIDE TREES.

TREES IN THE HIGHWAY ENGINEERING ASPECT.

(*R. A. Ryves.*)

***Shade in Summer.*—The value of shade, apart from comfort, is in a reduction of evaporation from macadam and flint roads, reducing the tendency to disintegration in dry spells. On tarred roads it prevents licking-up of the tar and the formation of ruts, and prolongs the life of the film. In the case of concrete roads the reduction of maximum crust temperatures reduces the tendency to the forming of cracks. On asphaltic and bituminous roads it either reduces the rutting and wearing effects of traffic, or, when its effect is taken into consideration in designing the crust or carpet, it is possible economically to construct a more efficient carpet, less hard and less slippery than one intended to withstand the effects of a whole day of hot sunshine.**

***Screening Effects.*—While some trees, such as the elm, have a greater frost-screening effect than others, such as the ash, all have an important effect in decreasing the range of temperature to which the road crust is subjected, in one day or in one year. Generally, they reduce very much indeed the number of occasions on which a road is slippery because frozen. By lessening the severity of freezing and by afterwards screening the frozen surface from the full effects of sunshine, trees save macadam and flint roads from most of the licking-up after frosts. They also, by mitigating the effects of severe frosts, save such roads from disintegration due to the formation of ice within the crust. Slipperiness due to hoar forst, or to the freezing of condensed mist or drizzle on a cold road, is much worse on the new surfaces than it is on water-bound macadam, so that, in this respect, the value of the tree is increased by the new conditions in winter and early spring.**

***Costs per Mile, 60 ft. apart.*—Basis for independent calculation: per penny per tree, 7*s.* 4*d.* per mile; per shilling per tree, 4*l.* 8*s.* (vi) Single line of trees, stakes at 9*d.* each: at 2*s.* 6*d.* per tree,**

14*l*. 6*s*.; at 3*s*. per tree, 16*l*. 10*s*.; at 3*s*. 6*d*. per tree, 18*l*. 14*s*.; at 4*s*. per tree, 20*l*. 18*s*.; at 4*s*. 6*d*. per tree, 23*l*. 2*s*.; at 5*s*. per tree, 25*l*. 6*s*.; at 5*s*. 6*d*. per tree, 27*l*. 10*s*.; at 6*s*. per tree, 29*l*. 14*s*.; at 6*s*. 6*d*. per tree, 31*l*. 18*s*.; at 7*s*. per tree, 34*l*. 2*s*.; at 7*s*. 6*d*. per tree, 36*l*. 6*s*.

(vii) *Hedges*, such as (v) (above), taken as 2 ft. apart; as stakes are not required, other spacings are in proportion: for *single hedge*, at 2*s*. 6*d*. per tree, 330*l*.; at 3*s*., 396*l*.; at 3*s*. 6*d*., 462*l*.; at 4*s*., 528*l*.; at 4*s*. 6*d*., 594*l*.; at 5*s*., 660*l*.; at 5*s*. 6*d*., 726*l*.; at 6*s*., 792*l*.; at 6*s*. 6*d*., 858*l*.; at 7*s*., 924*l*.; at 7*s*. 6*d*., 990*l*.

(viii) *Hedges*, such as (iii) (above), taken as 1 ft. apart, other spacings in proportion; double the values in (vii).

Quickset Hedge (thorn).—Taken as 7 to the yard: at 5*l*. per 1,000, 61*l*. 12*s*.; at 7*l*. 10*s*. per 1,000 92*l*. 8*s*.; or, with *a* per yard and *b* shillings per 1,000, the cost of *one* hedge is £ $\frac{a \times b \times 88}{1{,}000}$.

BRIDGE DESIGN.

Allowable Stresses on Concrete.

Increased stresses on concrete in bridge members are allowed by the Ministry of Transport, provided that: (1) The cement complies with current B.S.I. specifications for Portland cement; (2) Crushing tests on 6-in. cubes made with concrete taken from the mixer during progress of the work show consistently the results as in the table; (3) The cement is weighed and not measured by volume.

Allowable Stresses on Concrete in Bridge Members and Corresponding Requirements.

Concrete Mixture Cement : Fine Aggregate : Coarse Aggregate.	Working Stress.	Modular Ratio.	Crushing Strength of Concrete in 6-in. Cubes. At 28 days with ordinary Portland cement. At 7 days with rapid hardening Portland cement.	Crushing Strength of Concrete in 6-in. Cubes. An additional test (if required) as an indication at 7 days with ordinary P.C. at 3 days with R.H.P.C. should give results :
lb. cu. ft. cu. ft.	lb./sq. in.	n.	lb./sq. in.	lb./sq. in.
A : 2 : 4	5A + 300	—	15A + 900	10A + 600
90 : 2 : 4	750	15	2250	1500
120 : 2 : 4	900	15	2700	1800
150 : 2 : 4	1050	12	3150	2100
180 : 2 : 4	1200	10	3600	2400

Heavy Vehicles on Bridges.

Section 25 of the Road Traffic Act, 1930, provides that: The bridge authority of any bridge may, by means of conspicuous notices, prohibit the use of the bridge: (*a*) by a vehicle exceeding a defined maximum laden weight, not less than five tons; (*b*) in respect of the laden weight of a vehicle when travelling at more than a defined speed. Provision is made for appeal to the Minister, who may order a prohibition or restriction to be removed, or may refer the matter to an arbitrator. The method of calculating the weight is laid down in Section 26, and procedure in respect of the weighing of vehicles in Section 27.

ROAD TUNNELS.

Dartford-Purfleet Tunnel: River Thames.

As designed in 1936. Length, not including open approaches, nearly 1 mile. Estimated cost, £3,200,000. Section, a semi-circle of 30 ft. diameter on a segmental invert. Shell, cast-iron segments, concrete-filled and concrete-lined. Provides a 20-ft. carriageway and two 2-ft. patrol footways. Ventilation: fresh air enters from a longitudinal duct under the soffit, the spent air being drawn to a suction duct, through orifices along the kerb line. Construction, by means of a 12-ft. diameter pilot tunnel and the two 18-ft. diameter ventilating shafts, about 100 ft. deep and terminating in shield chambers, 35 ft. in diameter, from which the full-section tunnel will be driven.

THE MERSEY TUNNEL.

(Described and illustrated in *The Engineer*, July 13, 20, and 27, 1934.)

Total length, including open approaches and branches, 15,191 ft.; main tunnel, portal to portal, 10,584 ft., branches, 3,216 ft.; length between waterside ventilating shafts, 4,500 ft., outside and inside diameters of the under-water tunnel, 46 ft. 3 in. and 44 ft.; width of main carriageway, 36 ft.; two footways, for patrol only; two-way (four lanes) traffic; headroom, 13 ft. 6 ins.; maximum depth to invert below H.W.O.S.T., 103 ft.; maximum gradient, 1 in 30; estimated traffic capacity, 4,150 vehicles per hour; carriageway paved with cast-iron blocks; lining, segmental rings of cast iron, metal thickness, 1¼ ins., the segments filled with concrete. Cost of the works, £7,077,800. Opened July 18, 1934.

THE ANTWERP, RIVER SCHELDT, TUNNELS.

(Described in *The Surveyor*, September 2, 1932.)

Total length, including open approaches, 6,923 ft.; portal to portal, 5,803 ft. (cast-iron-lined, 4,053 ft., reinforced-concrete-lined, both ends, 1,750 ft.); length between waterside ventilating shafts, 2,836 ft.; outside and inside diameters of under-water tunnel, 30 ft. 9 ins., and 28 ft. 6 ins., of the reinforced-concrete-lined portions, upper half ring and invert, 34 ft. 6 ins., and 28 ft. 6 ins. nearly, sides continued downward as walls, externally; width of carriageway, 22 ft. 2 ins.; one footway, for patrol only; two-way (2 lanes, one each way) traffic; headway 14 ft. 9 ins.; maximum gradient, 1 in 28·6; maximum depth to invert below H.W.O.S.T., 99 ft.; below the bed of the river, about 54 ft.; estimated traffic capacity, 2,000 vehicles per hour. The segments of the cast-iron-lined portion are filled with concrete. Cost of the works, inclusive of land and property, buildings and engineering costs, £1,425,000.

A separate tunnel, at some distance, has been constructed for foot-passengers; length between centres of lift shafts (the design following Barlow's precedent), 1,877 ft.; tunnel diameters, 15 ft. 7 ins. and 14 ft. 1½ in.; footway width, 12 ft. 5½ ins. at floor level; space for pipes and cables under the footway; estimated capacity, 16,000 persons per hour. Cost, including electrical and mechanical equipment, about one-fifth that of the vehicular tunnel.

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'Vertical Curves for Roads.' By F. G. Royal-Dawson. Spon, 1946. Price 15*s*.

'Limestone Roads.' By P. A. Valton. Chapman and Hall, 1947. Price 10*s*. 6*d*.

'Design and Construction of Concrete Roads.' By R. A. B. Smith and J. R. Grigson. Concrete Publications, Ltd., 1946. Price 8*s*. 6*d*.

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'Computation of Traffic Stresses in a Simple Road Structure.' By L. Fox. Road Res. Tech. Paper, No. 9.

'Concerning Certain Practices in Road Engineering.' By R. A. Ryves. Article, *The Surveyor*, August 20, 1948.

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'Soil Mechanics as Related to Road Construction.' A lecture (M.I.Mech.E.) delivered March 9 1938, by Dr. L. Casagrande. Abstract, *The Surveyor*, March 18, 1938.

'Road Engineering and its Development in Australia.' A paper (epic and technical) by H. H. Newell (Eng. Conf., Sidney, March 1938). *Trans. Inst. E. Australia.* Abstract, *The Surveyor*, May 6, 1938.

'American Highways and Highway Practice.' By E. E. R. Tratman. Article, *The Engineer*, October 8, 15, 22 and 29, and November 12, 19 and 26, 1937.

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'Architectural Principles of Bridge Design.' By W. J. Watson. An address delivered in January 1938. Abstract, *Civil Engineering (United States)*, March 1938. Extracts, *The Surveyor*, May 27, 1938. Mr. Watson cites Vitruvius' treatise, written nearly 2,000 years ago, in which the Roman architect enunciated six principles or elements of architectural design: Order, Symmetry, Arrangement, Eurythmy, Propriety, Economy—and finds, in modern conceptions in regard to bridge design, recognition of the same six principles.

Report of the Select Committee of the House of Lords on the Prevention of Road Accidents. Issued April 1939. Comment: An editorial note in *The Surveyor*, April 7, ends: 'Because the committee have ruled out considerations of natural law in regard to matter and motion as involved in the running of vehicles, their report has the aspect of an elaborate piece of special pleading on behalf of those who find pleasure in swift travel on the roads. Nevertheless, it is of very great value as providing, in all other aspects, the basis for an immediate and for a long-term policy in regard to road safety.'

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'The Dragline Excavator.' By W. Barnes, *Journal Inst. C.E.*, March 1940.

'Earth-Moving Plant.' By L. F. Loder. Article, *The Commonwealth Engineer* (see also *The Surveyor*, October 11, 1940).

'Observations of Settlements on Embankments.' By A. H. D. Markwick and G. C. Wilson, of the Road Research Laboratory. A note (Crown copyright reserved), *The Surveyor*, July 5, 1940.

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BRIDGE DESIGN.

'Rigid Frames without Diagonals: the Vierendeel Truss.' By L. Baes. Paper, *Proc. Am. Soc. C.E.*, January 1941. Abstract, *The Surveyor*, March 21, 1941.

'Simplified Theory of the Self-anchored Suspension Bridge.' By C. H. Gronquist. Paper, *Proc. Am. Soc. C.E.*, February 1941. Abstract, *The Surveyor*, April 18, 1941.

'Bridge Design and Construction in the United States.' By W. G. Bowman. Article, *Engineering News-Record*, February 13, 1941.

'Trends of Bridge Design in the United States.' Editorial note, *The Surveyor*, April 4, 1941. The writer discusses the failure of the Tacoma Narrows suspension bridge, in the light of Mr. Bowman's observations thereon (*v.s.*), and with regard to Mr. L. S. Moisseiff's recent explanations of the reasons for the adoption of new or changed features of suspension bridges, including very much reduced truss depths. See 'Growth in Suspension Bridge Knowledge.' By L. Moisseiff, *Engineering News-Record*, August 17, 1939. Abstract, *The Surveyor*, September 29, 1939.

'The Suspension Bridge Tower Cantilever Problem.' By B. Birdsall. Paper, *Proc. Am. Soc. C.E.*, April 1941.

'The Graphical Solution of Portal Frames.' By J. W. H. King. Paper, *Journal Inst. C.E.*, December 1940.

'Aesthetics of Engineering Structures.' By O. Faber. Paper and Discussion, *Journal Inst.*, *C.E.*, April 1941.

ADDENDA 1945.

'Relative Merits of Wire and Bar Reinforcement in Pre-stressed Concrete Beams.' By R. H. Evans. Paper, *Journal Inst. C.E.*, February 1942. Presents experimental results in regard to deflection, hysteresis loops, permanent set and cracking loads, for increasing values of initial pre-stressing force. The test beam was 10 in. deep, 4 in. wide, and 11 ft. long, the load applied on a span of 10ft. The permanent set in all the pre-stressed concrete beams was exceedingly small, no more than that in plain concrete beams. The cracking load of the pre-stressed beams with bar reinforcement was considerably lower than that of pre-stressed beams with wire reinforcement, about half for an initial pre-stressing force of 25 tons. That, the author maintains, is important, because the working load should not exceed that necessary to produce tensile stresses in the concrete. The experiments showed that piano wire having a tensile strength of 150 to 175 tons per sq. in. is a very efficient type of reinforcement in pre-stressed concrete.

Birchenough Bridge.—A two-hinged arch with a span of 1,080 ft. in 27 panels of 40 ft.; lower chord parabolic with a rise of 216 ft. Depth, at crown 37 ft. 6 in.; at end posts, 46 ft. Arch trusses 45 ft. apart. The deck of cross girders, stringers and transverse joists supports a light reinforced concrete road slab, 68 ft. wide and 5 in. thick, and two 3-ft. timber footways. The roadway is about 25 ft. above springing level and, except at end panels, the cross girders are suspended from the arch by hangers of wire cable. An expansion joint is provided at panel point from the west end and an articulation joint at panel point 4 from the east end.

The Ristna Bridge.—Described by Mr. C. G. Sexton, in the May 1946 number of the *Journal of the Institution of Civil Engineers*, this outstanding work is notable with respect to its design, the method of construction and the manner in which were overcome difficulties of the site and of conveyance of materials to the site. The rainfall at the site, which is in a gorge in jungle country, averages 200 ins. a year, and there are very severe floods. The middle arch has a span of 276 ft. and a rise of 132 ft. On one side there are two approach spans of 52 ft., on the other, three spans of 56 ft. 6 in., curved to a radius of 120 ft. The method of erection was to build out as cantilevers, finally 78 ft. each, two part-arches, and to close the gap by lowering the steel frame, or skeleton, of a Melan arch, concreted after being attached to the part-arches. In the erection of the cantilever part-arches, each increment of concreting left protruding stout steel rods, stiff enough and strong enough to sustain the next increment of concreting. Study of Mr. Sexton's paper makes it clear that the erection-design was adopted for good reasons in preference to the method of building as cantilevers two half-arches of steel skeleton and concreting afterwards.

Brathay Church Footbridge.—Erected in 1946, this bridge, which crosses a swift mountain stream, links Lancashire and Westmorland. It is an arch of 50 ft. span, 10 ft. rise, 7 ft. wide over parapets and 4 ft. wide inside parapets. The arch, weighing 50 tons, was erected by means of tubular steel scaffolding. The stone of the arch was obtained from the spoil banks of two slate quarries, one stone being of a lighter and the other of a darker green. The masonry of the arch has been described as 'rubble,' but a photograph shows large stones of the appearance of voussoirs. Between the ring faces the remainder of the arch is reinforced concrete. The whole of the exposed concrete face in the arch soffit was painted with a half-and-half mixture of cowdung and butter-milk, to encourage the growth of moss, facilitated by the use of sawn segments for the centering timbers.

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SECTION XVI

NAVIGABLE WATERWAYS—VELOCITY OF RIVER FLOW—TREATMENT FOR EROSION—CHANNEL TRAINING WORKS—RIVER DREDGING—COST OF RIVER REGULATION—RIVER CANALISATION—COST OF CANAL CONSTRUCTION—CONTINENTAL CANALS—LOCKS AND MECHANICAL LIFTS—CANAL CURVES—LOSS OF WATER ON CANALS—TRACTION AND HAULAGE—SHIP CANALS

(pp. 577–602)

(Contributed by Brysson Cunningham, D.Sc., B.E., F.R.S.E., M.I.C.E.)

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NAVIGABLE WATERWAYS.

CLASSIFICATION.

(1) Natural Channels, i.e., those rivers, or portions of rivers, including estuaries, the normal flow of which is not interfered with by artificial works;

(2) Canalised Rivers, frequently called 'Navigations,' the flow of which is more or less under artificial control, and

(3) Canals, proper, entirely artificial in conception and construction.

The distinction between the three classes is fundamental and important. In the case of the natural channel, improvement works are limited to a rectification of the bed, and no attempt is made to control or augment the natural flow of water, which, being derived from the drainage of the basin, is normally sufficient for purposes of navigation, within the limits of the channel capacity. The canalised portions of rivers consist of sections in which the natural flow is conserved and controlled, as in the manner of a canal, but generally without extraneous sources of water supply. Canals, being works of an entirely artificial character, have to be fed with supplies obtained by the interception of streams, or from other external sources, frequently necessitating pumping and reservoir storage.

IMPEDIMENTS TO NAVIGATION OF RIVERS.

In the seaward section the most pronounced disability, where it exists, is the bar. For its removal a very efficient means, producing immediate amelioration, is dredging; but the effect in the majority of cases is merely temporary, and operations must be continuous if the improvement is to be maintained. Where bars are not in evidence, or not very pronounced, a river mouth is often encumbered by shoals, due either to detritus brought down by the river itself or to coastal erosion and drift. In this case also, dredging may be resorted to with immediate advantage. For permanent benefit it is requisite to supplement this temporary expedient by training works of a scope and magnitude dependent on circumstances. Such works comprise longitudinal training walls, or embankments, and groynes, dykes, or jetties, to be alluded to later.

In the upper sections of rivers, the principal difficulties encountered are irregularities in flow, due to floods in rainy seasons and depletion of water in periods of drought, excessive current velocity, instability of channels and the formation of shoals. In low-lying lands, it is often necessary as a protection against inundation to raise the river banks with impervious material, well supported from behind. Insufficiency of water may be met by the introduction of weirs, or the formation of storage reservoirs. The remedy for excessive and irregular currents lies in training works of a suitable character.

Velocity of Flow in Rivers.

The velocity of river flow varies very considerably, as will be seen from the following table. As regards navigation, currents exceeding three to four miles an hour or four and a half to six feet per second make a serious deduction from the speed of propulsion of vessels proceeding upstream, and render control difficult in the case of small craft travelling downstream. See also par. on Horse Traction, page 597.

River Seine at Meulan	2·45 ft. per second.	River Rhine at Basle	6·36 ft. per second.
" " at Paris	4·00 " " "	" Aar at Berne	7·50 " " "
" Thames at London Bridge	3·00 " " "	" Durance, below Sisteron	8·65 " " "
" Tiber at Rome	3·40 " " "	" Maranon, S. America	13·00 " " "

Velocity Observations are taken by means of (*a*) Floats, (*b*) Current Meters, and (*c*) Pressure Tubes.

Floats.—For surface velocities an orange is excellent, floating almost wholly immersed. In the winter season, fragments of ice give similarly good results. The topmost layers of water, however, are frequently affected by the wind. To obviate this the float should extend some distance below the surface. A corked bottle, weighted at the bottom, has proved satisfactory for the purpose. Tube floats are also employed, consisting of tin cylinders about 1 in. diameter, weighted so as to maintain perpendicularity, and leave two or three inches projecting above the surface. Double floats are sometimes used, consisting of a small surface float connected by means of a cord to a float of larger bulk below. In Mississippi experiments, the lower floats were kegs 9 ins. high and 6 ins. diameter, or 12 ins. high and 8 ins. diameter, according as depth was less or greater than 5 ft. They were without top or bottom, and ballasted with lead. The connecting cords were $\frac{1}{16}$ in. to $\frac{1}{8}$ in. diameter. The upper float was of light pine, $5\frac{1}{2}$ ins. square by $\frac{1}{2}$ in. thick, or of hollow tin, $5\frac{1}{2}$ ins. by $1\frac{1}{2}$ in.

Current Meters.—These record velocity by means of the revolution of a screw, being calibrated by drawing through still water at various speeds. The Price current meter, used by United States Geological Survey, consists of six cups attached to a vertical shaft, which revolves on a conical hardened steel point when immersed in moving water. Meter measurements can be made from a bridge, from a cable and car specially installed for the purpose, from boats or by wading. Current meters are affected by friction, and are not very suitable for low velocities; in fact, in feeble currents they cease to act.

Pressure Tubes.—Pitot tubes, which are L shaped with the lower horizontal limb pointing upstream, record the velocity by the height to which the water in the tube is forced above surface level by the pressure of the current. Darcy's adaptation consists of two tubes in communication, enabling the differential reading to be taken more accurately and eliminating the effect of ripples and oscillations. The apparatus gives good results in small channels with currents of fair velocity, but it is not suitable for large rivers or for very low rates of flow.

Position of Maximum Velocity may be taken generally at or about surface level, but is not infrequently below. The view is generally held that the velocity varies as the abscissæ of a parabola, whose axis is parallel to the surface. The equation for the parabola in the case of the Mississippi was found to be

$$v = -0{\cdot}79222\, d^2 + 3{\cdot}2611$$

where,
v = velocity in feet per second and d is distance from axis in fractional parts of whole depth considered as a unit.

Mean Velocity.—The position of mean velocity has been found by repeated experiments in U.S. Geological Survey to be located at about 0·6 of the total depth. The mean of velocities obtained at 0·2 and 0·8 of the depth gave the mean velocity of the stream very accurately.

In ordinary cases the least, mean and maximum velocities may be taken in the ratio 3 : 4 : 5. In very slow currents they are nearly as 2 : 3 : 4.

Prony's Formula:

$$\frac{\text{Mean velocity}}{\text{Maximum velocity}} = \frac{\text{Maximum velocity} + 7{\cdot}71 \text{ ft. per sec.}}{\text{Maximum velocity} + 10{\cdot}28 \text{ ft. per sec.}}$$

Computed Velocity.—A great number of formulæ have been published connecting the velocity with the hydraulic mean radius (R) $= \dfrac{\text{area of stream}}{\text{wetted perimeter}}$, and the sine of slope (S) $= \dfrac{\text{fall in level}}{\text{length of channel}}$. Most of these expressions can be reduced to the form

$$v = c\sqrt{\text{RS}},$$

where c is a coefficient varying with the class and nature of the channel, as well as with the personality of the observer. Some of the coefficients are very complicated, and the values are variable to a considerable degree. Mr. A. A. Barnes ('Hydraulic Flow Reviewed,' 1916) proposed to modify the powers of R and S so as to admit of a coefficient of constant value. His expression is

$$v = 58{\cdot}4 \ \text{R}^{\cdot 694}\text{S}^{\cdot 496}$$

or, approximately,

$$v = 60 \ \text{R}^{\cdot 7} \ \text{S}^{\cdot 5}$$

In nineteen observations with ranges in width of stream from ·26 ft. to 2,598 ft., in depth from ·19 ft. to 69 ft., in hydraulic mean radius from ·0786 ft to 53·6 ft., and in slope from 1¼ to 241·2 ft. per mile, velocities of from 1 to 7½ ft. per second were computed and found to agree, within 1 per cent., with actual observation.

In the course of investigations made during three years in connection with the regulation of the Whangpoo and Yangtse Rivers, Dr. Herbert Chatley found that in fairly straight and uniformly sectioned reaches when the flow is steady the above exponential form $v = c\text{R}^{\cdot 7}\ \text{S}^{\cdot 5}$ gave consistent results. But when the flow was unsteady he found it more satisfactory to take the expression

$$v = c\text{R}^{\cdot 7}\left(\text{S} - \frac{a}{g}\right)^{\cdot 5}$$

where a is the acceleration. In other words, the effective slope was equal to the algebraic difference between the actual slope and an 'acceleration' slope determined by the ratio of the actual mean acceleration of the water to the gravitational acceleration.

Critical Velocities.—The following is a statement founded on the researches of Dubuat and Beardmore of velocities at which various materials in a river bed commence to be eroded:

Soft clay	3 ins. per second.	Rounded pebbles, 1 in. diam. .	2 ft. per second.
Fine sand	6 „ „ „	Rough gravel „ „ .	2½ „ „ „
Coarse sand and pea gravel	9 „ „ „	Pebbles 1½ in. „ .	3½ „ „ „
Gravel as large as French beans	12 „ „ „	Heavy shingle	4 „ „ „

TREATMENT FOR EROSION.

Protection of River Banks.—Water plants form a very efficient natural protection and they assist in retarding the current—sometimes not an unmixed advantage.

Earthen banks of a soft and erodible character may be artificially protected, where stone is procurable, by rubble pitching, laid at a slope of 1½ or 2 to 1, from 8 to 12 ins. thick at the top, and increasing in thickness downwards at the rate of 3 ins. per yard. The foot of the slope should be adequately secured to some firm abutment, such as piling, otherwise the pitching may be undermined and, in that case, it will subside.

Where stone is not procurable, fascine aprons or mattresses, formed of brushwood, may be utilised. These are bundles of branches and twigs (usually willow) from 9 to 12 ins. in diameter, about 12 ft. long and bound with tarred rope at intervals of 4 ft. or so, laid side by side and tied together so as to produce a surface covering of the desired extent. The 'fascines,' as they are termed, should be secured by stakes or short piles to the bank upon which they are deposited.

Reinforced concrete slabs in various forms have been employed, notably the De Muralt systems, in Holland and elsewhere.

A plantation of small willows above water level is useful in counteracting the more deleterious effects of floods.

River Bank Protection of the Missouri.—Regulation work on the Lower Missouri, comprising the formation of a 6-ft. navigable channel as far as Kansas City, a distance of about 390 miles from the mouth, was put in hand on an intensive scale in 1927.

'The cheapest form of bank protection on the Missouri has been found to be a mat below low water and a paved bank above. The mat is woven continuously, with an average width of about 85 ft. (26 metres), laid from the low-water line out into the river, and, after weaving on the surface of the water, sunk on the bottom by ballast. Willows, with "basket weave," were used for the mat, where available; otherwise it was built of 1-in. plank, with suitable bracing. Some 716,000 ft (230.000 metres) of such mat have been completed, and are in place (1930). Some individual mats in long bends were 5 miles or more long, woven continuously and sunk as they were woven. Above the water line the bank was graded by hydraulic jet to a slope of 1 on 3, and paved with broken stone, the fragments weighing from 40 to 80 lbs. This type of protection was first built, on a small scale, 30 or 40 years ago, and with adequate maintenance will apparently stand indefinitely. It costs about \$20,00 per ft., the maintenance averaging about 3·5 per cent., or \$0,70 per ft. per year' (*Extract from Report by Major G. R. Young to International Navigation Congress at Venice*, 1931).

Mississippi River Bank Protection.—The banks of the Lower Mississippi are protected where necessary from erosion below water surface level by means of brushwood mattresses loaded with stone. The portion of the bank above surface level is protected by a covering of stone only, 10 ins. thick, laid to a suitable slope.

The mattresses were deposited from a single barge, or from a couple of barges, placed end to end, so that the combined length of the barges exceeded the width of the mattress. The ordinary size for mattresses was 800 ft. by 250 ft., but some mattresses were 1,000 ft. long by 300 ft. wide. 'The mattresses are constructed on the mattress-barge by laying parallel to the shore, and, 8 or 10 ft. apart, a series of poles, whose aggregate width equals nearly that of the proposed mattress. These poles are small trees from 3 ins. to 6 ins. diameter at the butt, and 25 ft. to 30 ft. long. Smaller poles are then placed at right angles to the larger series, so as to pass alternately beneath and above the successive larger poles, as in basket-work. These smaller poles, when forced into close contact with each other and with the other series, form a tolerably compact web or mattress. When so many of the small poles have been added that they approach the ends of the larger series, whose poles are parallel to the shore these larger poles are each continued by having another large pole spliced to it, and so on, for the length of the mattress. The brush in the mattress is fastened together frequently with iron wire and cross poles. The whole mattress is strengthened by wire cables about 20 ft. apart, and by ⅜ in. to ⅝ in. wire ropes running longitudinally the whole length of the mattress. Transverse cables of wire, about 16 ft. apart, also strengthen the mattress; and transverse wire ropes in addition are sometimes used.'

As each mattress was woven it was allowed to be floated down stream off the barge by the current, and when a sufficient length had been accomplished it was loaded with stone and sunk the upper end being sunk first. The customary under-water slope did not exceed 1 in 3, and the depth of the outer edge often ranged from 60 ft. to 100 ft. (*From Report by General C. B. Comstock, on Mississippi River Improvement, to Chicago International Congress*, 1893.)

Flexible mattresses of reinforced concrete are being experimented with on the Mississippi.

CHANNEL TRAINING WORKS.

The instability and eccentricity of channels is remedied by training works, comprising (*a*) projecting jetties and groynes, (*b*) longitudinal dikes and embankments, and (*c*) ground sills.

Groynes or Jetties project from the banks to the stream at right angles, or thereabouts (generally within 70° to 90°) to its course, and have the effect of reducing the channel and concentrating the scour of the current within certain bounds. The beneficial effect, however, is largely local, and, in the absence of restraint between the jetties, the stream tends to diffuse itself over a wider area, so that while the bed may be deepened and improved opposite the ends of the groynes it often becomes actually shallower in the intermediate spaces. To be effective, groynes should be spaced apart not more than five times their own length, and this often renders them less economical than a system of longitudinal diking, which, on the whole, will be found far more satisfactory and is often ultimately adopted. Groynes should, in fact, only be looked upon as of the nature of a temporary expedient. They may be formed of piling, fascine work, or rubble.

Groynes on River Missouri.—For contracting the river in a reach or at a crossing, or from the convex side of a bend, lateral groynes or longitudinal dikes were found equally efficient, the former being cheaper since they could nearly always be spaced more than their own length apart. They were, therefore, commonly used.

'The most satisfactory device is an openwork fence built nearly normal to the channel, consisting of several rows of stout piles, or clusters of piles, suitably braced, driven an average of 20 ft. (6·1 metres) into the bottom, the top of the structure being a foot or two below the natural bank from which they are built. (Two alternative designs, about equally efficient, are in use.) A striking feature is that it is not necessary to screen these dikes, or to provide any device other than the dikes themselves for arresting the flow of the water. While the dikes are entirely permeable when first built, being merely a skeleton framework, they rapidly accumulate driftwood at high stages of the river, which banks up behind them and forms a mass, sometimes many acres in area. This drift retards the water sufficiently to cause it to precipitate the sediment that it carries, and to produce an almost immediate fill between the dikes almost to their outer ends' (*Extract from Report by Major G. R. Young to International Navigation Congress at Venice*, 1931).

Groynes on the River Waal.—These are so arranged as to point upstream with the axis of the groyne on the concave bank forming an angle of 80° with the projected channel, and that on the convex bank an angle of 70°. This system has been found favourable to the deposit of alluvium between the groynes.

The groynes themselves are formed of a core of sand extending to 11 ft. 6 ins. below ordinary low water. The core is covered with mattresses of fascine work, having a lateral slope of 6 to 4, surmounted by a layer of broken brick and rubble stone. The slope towards the river is 1 in 4. The width of the crown of the groynes is 11 ft. 6 ins., and their height along the boundary of the normal river bed 1 ft. 6 ins. above ordinary low water level, corresponding to about 1 ft. 6 ins. below mean water level.

It is stated that sand forms a satisfactory core if banked in large quantities at a time, and that no harmful results ensue from the slipping away of a certain amount during submergence. A few years after formation the groynes receive a protective covering of basalt.

The River Waal has a velocity varying between 2½ and 3½ ft. per second. The discharge is about 21,000 cubic ft. per second at lowest water level, and 220,000 cubic ft. during highest floods. (*From Report by M. Baucke to Philadelphia Navigation Congress*, 1912.)

Longitudinal Dikes are walls or embankments of timber, stone, concrete, fascines, or other material built up so that the channel is rigidly confined within definite limits throughout the length treated. At the outlet of the Fen Rivers in the Wash, banks of fascine faggots and clay were constructed 22 ft. wide at the base, 17 ft. high, and 13 ft. wide at the top, at a cost of about 1*s*. 8*d*. per cubic yard. On the River Weser, stone and gravel embankments have been found preferable to fascines on the ground that the surface slopes may be rendered less steep than 45°, which is stated to be the limit for fascine work.

River Training Works at the Mouth of the Danube.—Operations for the improvement of the navigable channel at the Sulina mouth of the Danube have been in progress for more than half a century, under the direction of the European Commission of the Danube, an international body, constituted in 1856 by the Treaty of Paris.

There are three main arms, or branches, of the Danube delta, at its influx into the Black Sea. The Sulina arm, with the port of Sulina at its mouth, was chosen for treatment by reason of its greater depth and general use, despite the fact that its discharge was scarcely 10 per cent. of the total.

Rudimentary timber piers, consolidated by concrete mass-work and stone rubble, were constructed in the first instance, thus concentrating the outward current, and by the scour so occasioned, the depth across the bar of the river was increased from 9 ft. to 17 ft. rapidly; and subsequently more gradually to 20½ ft., at which figure it remained, naturally, for a period of 22 years until 1894, when dredging was commenced and a depth of 24 ft. obtained, necessitating, however, continuous maintenance work. In 1915 the depth of 24 ft. was lost and three dredgers proved incapable of re-establishing it. On account of the progressive advance of the delta, the action of natural forces, aided by artificial means, had become insufficient to cope with the deposit of alluvium, which, indeed, threatened to destroy serviceable access to the river for shipping.

In 1922 a prolongation of the piers or jetties was begun, and in 1925 was completed to a point about 3,300 yds. from the Sulina Lighthouse, which is located at the port entrance. On July 25, 1925, the new channel, 200 ft. wide, was opened to traffic with a minimum depth of 20 ft., which within a month was increased to 23 ft. Deposit, however, continued to take place at the outer extremities of the piers, and a further prolongation was necessary in order to reach deeper water for the pierheads.

Levees constitute a type of longitudinal dike. They contract the river channel in its higher stages and protect the adjacent country from inundation. As used in American practice, they are generally banks of earth or spoil, well consolidated, with a central trench forming an impervious connection with the natural ground.

Channel Curves.—The natural tendency of river channels is to form a series of deep water pools on the concave sides of bends, with shoals at the intervening points where the current passes over from one side of the river to the other. A judicious system of training will maintain the natural sequence of these deep pools, at the same time suppressing any unduly sharp bends which might be awkward for navigation. Generally speaking, for vessels of 400 to 600 tons moving at a fair rate of speed a radius of 550 yards should be regarded as a minimum—at low speeds, 220 yards will suffice. The minimum for vessels of very small tonnage (say, 200 tons and under) may be as low as 66 to 88 yards.

Ground Sills are serviceable where there is excessive scour of the bed or undue restriction in width of channel. They have the effect of rendering the fall of the stream more uniform. In German practice they are set from 40 to 50 feet apart, and found to be better than continuous pitching or filling of the bed. They are laid across the river so that their highest levels at the bank are about 12 inches below the normal surface, and they slope downwards to the centre of the stream at an angle of about 1 in 40. On the Mississippi the ground sills consist of either rubble mounds or cribs filled with rubble, or, again, of brush mattresses and stone in alternate layers. Whatever form be adopted, the top should be finished off smoothly to prevent injury to passing vessels. Ground sills may be used where it is desired to raise the low water level or lengthen out a slope.

Storage Reservoirs.—For the augmentation of the flow in dry seasons storage reservoirs have been advocated, and in two instances—the Mississippi and the Volga—have been adopted. Circumstances, however, have to be favourable if they are to prove a satisfactory and economical arrangement. The Volga has its source in a lake and swamp region where land is cheap and unfit for agriculture. The dams required were only 17½ feet high, and a storage capacity of 35.000 million cubic feet was obtained. On the Upper Mississippi above St. Paul, five reservoirs were formed out of an extensive lake area of 480 square miles, with a storage capacity of 93,400 million cubic feet. It is stated that the increase in height obtained at St. Paul during the low water period—about 90 days—averages only 14 inches. The system is not regarded as sufficiently satisfactory to justify its extension elsewhere in the United States. Where there is no original headwater accumulation its adoption is out of the question.

RIVER DREDGING.

The improvement of river channels by means of mechanical dredging has received considerable impetus of late years by reason of the development in power and capacity of dredging plant. The various appliances are described in detail in connection with harbour dredging (*vide* p. 634 *et seq.*). In some cases, as for instance the Volga, with its tributaries the Oka and the Kama, the application of dredging alone, without the assistance of artificial training works, has proved sufficient to ensure the maintenance of a serviceable channel, while in other cases, notably on the Mississippi and the Mersey, engineers employ both systems conjointly.

COST OF RIVER REGULATION.

Country.	River.	Depth.	Length.	Total Cost.	Average Cost per Mile.	Average Fall per Mile.
		Ft.	Miles.	£	£	Ft.
United States	Upper Mississippi (St. Paul to Missouri River)	4½	632	2,050,000	3,241	0·44
France . .	Rhone (Lyons to Sea) .	4·1	205	2,689,000	13,000	2·5
Germany .	Rhine	4–10 3–4·1	214	2,636,400	12,200	1·14
Ditto .	Weser (near mouth) .	5–10 3·6–4·6	210	463,300	2,210	1·65
Ditto .	Elbe (near mouth) .	6–10 2·6–4·25	252	2,047,900	8,130	0·98
Ditto . .	Oder (near mouth) .	5–8·7	337	1,229,600	3,650	1·29

(*Extracted from Report by Major Harts to Philadelphia Navigation Congress*, 1912.)

RIVER CANALISATION.

The canalisation of rivers is effected by dividing their beds into a series of reaches separated by dams, or weirs, which are provided with a communicating lock, or locks, arranged generally at one side. Dams are either fixed or movable, and they may also be permeable or impermeable. Fixed, impermeable structures are generally resorted to as being more economical in construction and more convenient in action. Movable dams may be said to be required only in cases of excessive flood flow, or where the conditions admit of free navigation during appreciable periods. Fixed weirs are usually constructed of solid stonework, rubble, piling or concrete provided with the requisite sluices and overflows. Movable weirs are more often of timber or iron framework with mechanism for lowering and raising them, either wholly or in part by means of leaves or panels. They are of various types with distinguishing names, such as Needle, Chanoine, Boulé and Bear-trap.

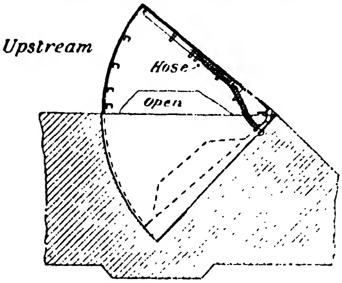


Fig. 1.—Section of Weir near Bremen.

Sector Weir near Bremen.—A notably large sector weir which constitutes a development of the bear-trap type, across the River Weser, near Bremen, is illustrated in cross sectional outline in fig. 1. The weir is in two lengths of 177 ft. each, and holds up the water on an average to a height of about 12 ft.; in winter, the height rises to 18 ft. The structure forms the hollow horizontal sector of a cylinder of 21½ ft. radius turning on a horizontal shaft, the angle of the sector being about 45 degrees. The weight of each sector is 129 tons. A description of the weir with the method of raising and lowering is contained in *Engineering* of July 7, 1916.

Cost of River Canalisation.

River.	Length Canalised. Miles.	No. of Locks.	Depth. Feet.	Fall in Feet per Mile.	Total Cost.	Cost per Mile.	Operation and Maintenance.	
							Total.	Per Mile.
France:					£	£	£	£
Saone	232	30	8·2	·85	1,755,000	7,550	12,100	52
Seine (Montereau to Paris)	61	12	—	·85	991,000	16,300	10,870	178
Seine (Paris to Rouen) .	145	9	10·5	·85	3.500,000	24,200	18,450	127
Seine (New Works) . .	140	9	—	- -	2,535.000	18,100	—	—
Yonne	67	26	-	2·43	1,118,000	16,700	6,730	100
Marne	113·5	19	7·25	1·16	1,047,000	9,250	7,980	70
Aisne	35·5	7	—	—	195,400	5,470	2,130	60
Germany:								
Saar	19·5	6	4·25– 7·9	2·15	354,400	18,170	6,410	330
Main	23·5	5	5·9– 8·9	1·40	448,200	19,070	8,010	340
Fulda	17·0	7	6·6-11·9	3·30	157,000	9,240	4,385	260
Salle	89·5	15	4·4– 8·8	1·27	383 400	4,280	9,080	100
Unstrut	40·5	12	2·5– 8·5	1·72	105,600	2,600	2,840	70
Oder	53·2	14	—	1·90	1,211,800	22,780	43,420	815
United States:								
Black Warrior . .	91	7	6	2·76	508,080	5,583	25,207	275
Coosa	25	3	3	2·58	209,687	8,386	1,587	64
Allegheny . . .	25	3	—	2·06	267,574	10,703	9,328	373
Monongahela . . .	131	15	7–8	2·4	1,369,171	10,452	34,673	265
Muskingum	84	10	6	1·55	—	—	10,026	120
Great Kanawha . . .	90	10	6	0·84	844,765	9,394	16,361	187
Kentucky	226	11	6	0·87	580,662	2,578	31,989	141
Green and Barren . .	190	7	5	0·49	327,682	1,725	15,177	80
Cumberland	131·4	8	6	0·67	525,076	3,996	6,611	50
Illinois	194	2	7	0·05	303,145	1,563	2,568	153

(Extracted from Report by Major Harts to Philadelphia Navigation Congress, 1912.)

TYPES OF UNITED STATES RIVER REGULATION WORKS.

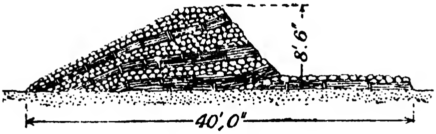


FIG. 2.—Brushwood and Stone Dam, Upper Mississippi.

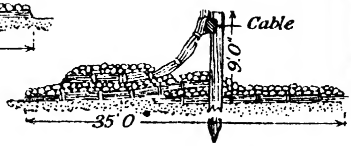


FIG. 3.—Pile and Brushwood Dam, Upper Mississippi.

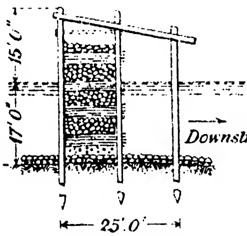


FIG. 4.—Pile, Brushwood, and Stone Dyke, Arkansas River.

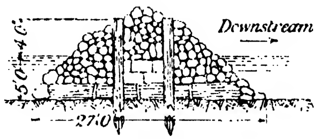


FIG. 5.—Pile and Stone Dam, Alleghany River.

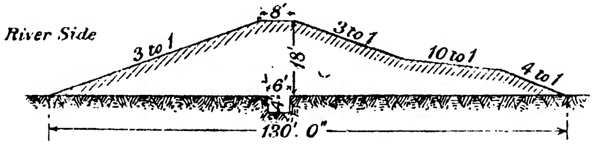


FIG. 6.—Levee on the Lower Mississippi.

Lateral Canals.

As an alternative to river canalisation and where the channel exhibits features which do not conveniently lend themselves to treatment, an independent lateral canal may be constructed, generally following the lie of the river, but also making use of any facilities which may offer themselves for straightening and rectifying the course.

The River Rhone, in the chequered history of its amelioration, has been furnished with three lateral canals to overcome the difficulties of navigation at its natural mouth. Each in turn of the earlier two became inadequate for dealing with the traffic which accrued from its construction, and now there is a recently constructed artificial waterway to the port of Marseilles which is described in detail on p. 587.

COST OF LATERAL CANALS.

Locality.	Length in Miles.	Locks.	Total Cost.	Cost per Mile.	Maintenance and Operation.	
					Total.	Per Mile.
			£	£	£	£
FRANCE:						
Loire, Digion to Briare	121·7	37	2,071,000	17,000	6,330	52
Garonne	120	53	2,484,000	20,700	7,500	62½
UNITED STATES:						
St. Mary's Falls	1·1	2	1,611,450	1,464.955	20,620	18,740
Des Moines Rapids	8	3	316,600	39,576	7,754	969
Muscle Shoals	18	11	638,345	35,463	10,280	571
Colbert Shoals	8	1	441,588	55,200	—	—
Cascades Canal	½	2	764,065	1,528,130	2,877	5,751

(*Extracted from Report by Major Harts to Philadelphia Navigation Congress*, 1912.)

BRITISH CANALS.

The following summary of the conditions prevailing in Great Britain is extracted from the Report of the Royal Canal Commission (1909):

'On the whole, the waterways may be roughly divided into two classes, those having locks of more, and those having locks of less, than 14 ft. in width. Corresponding with these two classes of locks are two classes of vessels usually known respectively as barges and narrow boats, the latter being often called "monkey boats," and chiefly in use on the Southern and Midland canals of England. The barge usually measures from 60 to about 72 ft. in length by 14 ft. in width of beam; the narrow boat is about 70 to 72 ft. in length, but only 7 ft. in width. The narrow boat when loaded, and drawing about 3 ft. 3 ins. of water, will carry about 30 tons over many of the canals. The barge will carry about twice that weight, with the same draught. Both the narrow boat and the barge could carry a heavier cargo if the depth of water were increased sufficiently.

'The usual length of lock, both on the wider and the narrower waterways, in the South and the Midlands, is, accordingly, about 75 ft. But on the wider, or barge canals, the locks are usually between 14 and 15 ft. or more in width, and on the narrow canals between 7 ft. and 8 ft. In both classes of lock the depth of water on the sill varies, speaking very generally and with many exceptions, from 4 ft. to 5 ft.'

	Miles.	Chains.
Canals with locks under 14 ft. (narrow)	**1,165**	**5**
" " " over 14 ft. (barge canals)	**762**	**29***
Navigations under 14 ft.	**478**	**57**
" over 14 ft.	**834**	**20**

*** Including the Manchester, Gloucester and Berkeley, and Exeter Ship Canals.**

DEVELOPMENTS IN CANAL CAPACITY.

In France and South East Belgium the standard boat carries about 300 tons, while in Germany the newest canals are adapted to boats of 600 tons and upwards. In this country opinion as to future policy is divided between the adoption of the 300-ton type and the retention of the smaller class of boat in general use, not exceeding 100 tons. The following are the dimensions

recommended for the two classes respectively by Sir J. Wolfe Barry and Partners, who were called in to advise the Royal Commission on the matter.

Dimensions.	100-ton Boat. Canal.	300-ton Boat. Canal.
Length of barge over all	80 ft.	115 ft.
Beam " "	14 "	21½ "
Draught	5 "	6½ "
Area of immersed midship section of barge	69 sq. ft.	138 sq. ft.
Cross-sectional area of canal	350 " "	690 " "
Depth of canal	7 ft.	8½ ft.
Top width of waterway in open country	59½ "	92½ "
" " " " in towns	51¾ "	83 "
Width of waterway at bridges	32 "	48 "
Minimum headway of new bridges	10 "	13 "
" " " existing bridges	9 "	13 "
Ruling radius for bends of canal	12 chains	17 chains
Locks: Length, mitre to mitre of gates	253 ft.	266 ft.
" Width between side walls	15 "	22½ "
" Depth on sills	6½ "	8 "

GOVERNING DIMENSIONS OF LARGE BOATS AND CANALS AS DETERMINED BY FRENCH COMMISSION OF PUBLIC WORKS.

Tonnage of Boat.	Dimensions of Boats.				Features of Navigable Waterway.					
						Curves.		Locks.		
	Length.	Beam.	Draught.	Width at 6½ Ft. below Surface.	Depth on Axial Line.	Increased Width.	Radius R.	Serviceable Length.	Width.	Depth on Sill.
	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.
300 .	126·3	16·4	5·9	32·8	7·9	1,246/R	984	132·8	19·7	8·2
600 .	196·7	23·3	5·9	51·3	7·9	3,018/R	1,919	213·2	29·5	8·2
900 .	246·0	28·2	5·9	65·6	8·2	4,724/R	2,690	262·4	34·4	8·2
1,200 .	295·2	32·8	5·9	77·7	8·4	6,888/R	3,509	311·6	39·4	8·2

A boat, 24 to 25 ft. wide by about 200 ft. long, with 6½ ft. draught, has been recommended by experts as a suitable type for a 600-ton barge on the Canal du Nord.

On the Rhone, barges of the same width and 220 ft. long have been put into service, carrying 650 tons on a draught of 6 ft.

The type of boat on the Rhine is the lighter of 246 ft. by 36 ft., which with the same draught carries about 900 tons, but when immersed to 6½ ft. is capable of taking 1,200 tons, with a displacement coefficient of 0·75.

(*'Annales des Ponts et Chaussées,' March–April* 1920.)

CONTINENTAL CANALS.

Canal du Nord, France, duplicating the St. Quentin Canal, which forms part of the navigable waterway connecting Paris with the Northern parts of France.—The ordinary cross-

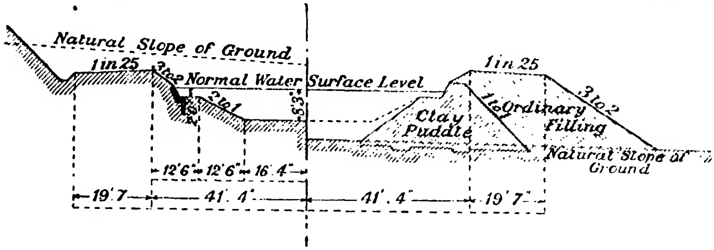


FIG. 7.—CANAL DU NORD

Section in Excavation. Section in Filling on Impermeable Ground.

section has a minimum width of 33 ft. at the bottom and 39 ft. at a depth of 6½ ft., with a navigable depth of 8 ft. Towpaths, 13 ft. to 20 ft. wide on each bank, one for upstream and one for down-stream traffic. Locks provided to accommodate two 300-ton barges in tandem; dimensions, 279 ft. long by 19½ ft. wide by 8 ft. deep. Lifts, 13 ft. to 22 ft., with minimum reach of 1,100 yards between locks. Tunnels of two types: those for single train of boats are 33 ft. wide at springing of arch, and those for double trains of boats abreast are 52½ ft. wide at springing. Each type has two side platforms, 6½ ft. wide carried on piles. Largest tunnel at Ruyaulcourt, which is 2¾ miles long. Canal designed for mean speed of two miles per hour, with barges mechanically hauled. Fig. 7 gives typical sections.

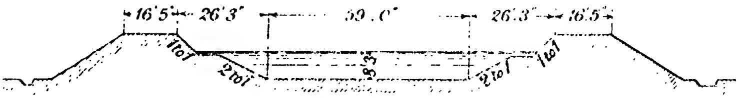


FIG. 8.—Brussels-Charleroi Canal.

The Rhone-Marseilles Canal.—Opened in 1926, the cross-section in the straight lengths gives a width of 82 ft. at a depth of 6½ ft., while the normal depth is 8 ft. 2 ins., and that in the outer portion approaching Marseilles 10 ft. Sills and inverts throughout are laid to this lower level, in order to admit of future deepening. The width of the canal is narrowed to 59 ft. where it passes through the Nerthe Mountain (Rove Tunnel). The canal section on the straight portion has been fixed at five times the immersed section of the present Rhone barges, when loaded to their maximum capacity of 600 tons on a 5 ft. 9 in. draught. These barges have a beam of 26 ft. 3 ins. The dimensions of the locks are 525 ft. internal length, by 52½ ft. in width. The lock at Arles has a lift of 23 ft. at low water. The total cost of the canal is stated to have exceeded 3,560,000*l*. The section of the Rove Tunnel shown in figs. 9 and 10 cost 52*l*. per ft. run, of which the masonry work accounted for 16*l*. per ft. run.

Modern German Canals.—Herr Germelmann, in a Report to the Philadelphia Navigation Congress 1912, remarks that in Germany high economic value is attached only to large efficient canals, suitable for vessels of 213 ft. in length by 26¼ ft. in breadth (600 ton barges), or, at least, for vessels of 180½ ft. in length and 26¼ ft. in breadth (400-ton barges) with draughts of 5 ft. 9 ins. The cross-sectional area of canals recently constructed is as follows:—Spree-Oder Canal from 705 to 760 sq. ft.; Dortmund-Ems Canal, in cuttings, 640 sq. ft.; in embankment, 810 sq. ft.; Teltow Canal, 662 sq. ft.; Ems-Weser Hanover Canal, in cuttings, 662 sq. ft.; in embankment, 780 sq. ft.; Rhine-Ems Canal, 986 sq. ft., Berlin-Stettin Canal, about 732 sq. ft.

Trollhatta Canal, Sweden.—This canal, first opened in 1607 and recently enlarged and extended at a cost of about 1,300,000*l*., has a length of about fifty-two miles, and connects Lake Vanern with the Kattegat, between which there is, at ordinary water level, a vertical difference

RHONE-MARSEILLES CANAL. SECTIONS THROUGH ROVE TUNNEL.

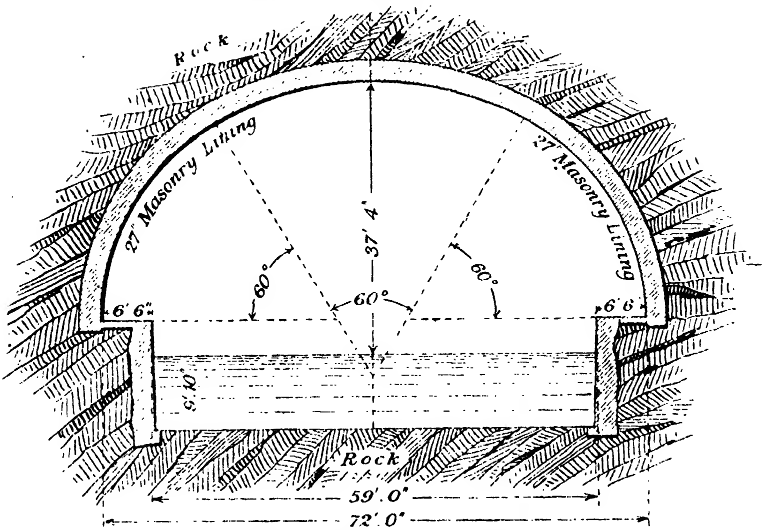


FIG. 9.—In Sections of Low Thrust.

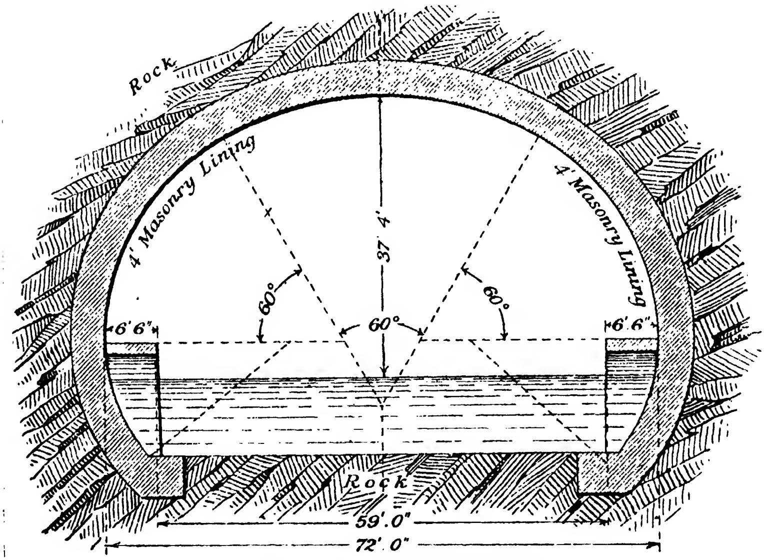


FIG. 10.—In Sections of High Thrust.

EXAMPLES OF MODERN GERMAN CANALS.

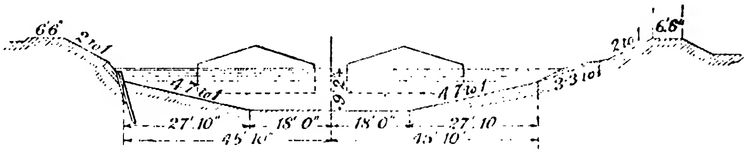


FIG. 11.—Spree-Oder Canal.

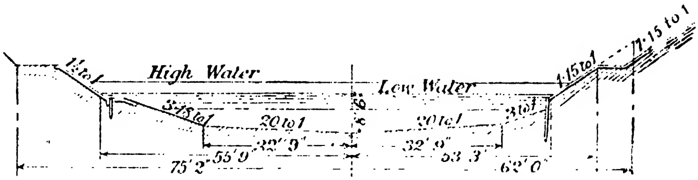


FIG. 12.—Teltow Canal.

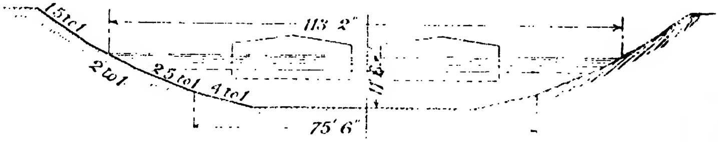


FIG. 13.—Rhine-Ems (Herne) Canal

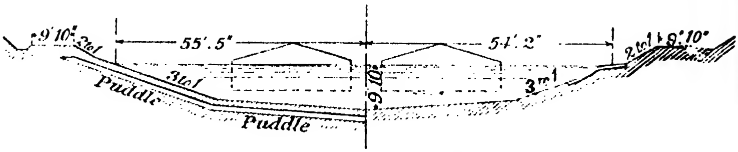


FIG. 14.—Berlin-Stettin Canal.

of 146 ft. The canal is built to accommodate vessels of 13 ft. 1½ in. draught, but with a view to future developments the locks are constructed to pass vessels of 16 ft. 5 ins. draught. The normal bottom width of the canal is 79 ft., but this, at passing places, is increased to 95 ft. and 115 ft. The maximum depth in rock cutting for vessels drawing 13 ft. 1½ in. is 14 ft. 5 ins., and in earth excavation, 18 ft. 3 ins. For the 16 ft. 5 in. vessel these dimensions are increased to 18 ft. and 19 ft. 7 ins. respectfully. The lock chambers have a length of 295 ft. and a width of 45 ft., with 18 ft. of water over sills. The greatest change in level at any one lock is 26 ft.

New York State Barge Canal.—This canal, the inception of which dates back to the beginning of last century, has recently been enlarged so as to afford accommodation not only for the largest class of barges at present constructed, but also for those up to 3,000 tons which may be built in the future. A minimum depth of 12 ft has been provided, with a minimum width of 94 ft. in rock cuttings and of 125 ft. in earth excavation. The locks, 57 in number, have chambers 310 ft. long and 45 ft. wide and are capable of passing vessels with maximum dimensions of 300 ft. by 42 ft. The lock at Little Falls has a range of 40½ ft., exceeding any single lift on the Panama Canal. The syphon lock at Oswego is the first of its type in the United States, and with a range of 25 ft. is probably the largest of its kind in existence. A description of it is as follows (*vide Engineering*, August 2, 1918). The general design of the culverts is not unlike that of an ordinary lock, except that at the upper and lower ends the culverts are curved upwards so as to form necks. These rise a little above the highest water level and are shut off from all communication with the outer air, except through the operating pipes. The flow of water in the syphon is started by means of tanks, one of which is built in each wall near the upper end, and communicates through pipes with the upper and lower levels and with both syphons in the same wall; they are shut off from all other communication with the outer air. The cycle of operations is as follows: The tank is first filled with water, then the intake valve is closed and the outlet valve opened. In consequence there is a body of water suspended by its weight, but tending to escape into the lower pool, and thus producing the necessary vacuum. Therefore, on opening the air valve, the air in the syphon rushes towards the vacuum, and water begins to flow up and over the crest in the neck. When using both syphons, the lock chamber can be filled in from 4½ to 5 mins., and it can be emptied in from 5½ to 6 mins. It has been found that the draught of the vacuum is such that, soon after the flow has started, the direction of the air is reversed and the vacuum is restored in the flask. The operating power is accordingly self-renewing except for a small amount of air leakage.

Cost of Canal Construction and Improvement in Great Britain.

In 1905 Mr. J. A. Saner estimated the constructional cost of new canal, suitable for two lighters of not less than 250 tons carrying capacity, or 200 tons when self-propelled, with locks 230 ft. by 22 ft. having 6½ ft. draught of water over sills, the canal itself being 40 to 45 ft. wide at bottom, 80 to 85 ft. wide at top, and 8 ft. deep in centre, at 30,000*l.* per mile, and the cost of improving existing routes to reach the same standard at 15,000*l.* 'These figures are based on the actual cost of the Weaver Navigation and the Manchester Ship Canal; the former has cost about 42,000*l.* per mile, and the latter about 440,000*l.* per mile, inclusive of docks in each case. The Worcester and Birmingham Canal originally cost 20,000*l.* per mile; and between 1879 and 1891 the French improved 2,500 miles of river, 2,250 miles of canal, and built 875 miles of new canal, for a total of 21,000,000*l.*, or an average of 3,734*l.* per mile; so that the foregoing figures should be ample.' (*Saner on Waterways in Great Britain, Min. Proc. Inst. C.E.*, vol. clxiii.)

RELATIVE ADVANTAGES OF LOCKS AND LIFTS ON CANALS.

Changes of level in canals and canalised rivers are effected by (1) locks, and (2) mechanical lifts, the latter being either (*a*) vertical or (*b*) on an inclined plane.

Locks are the simplest and most stable means of attaining the desired object. There is no question of their superiority for all differences of level up to 12 or 15 ft., and circumstances are generally favourable for their adoption for differences up to 25 or 30 ft., and even more. Among

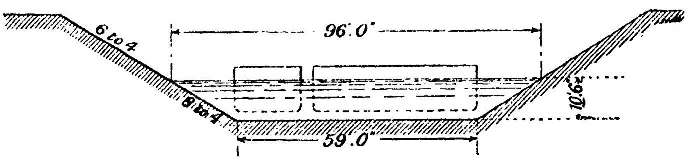


FIG. 15.—Bruges-Ghent Canal (General Section).

the older canals may be instanced a lock on the River Moldau, which has a lift of 29 ft., and another on the St. Denis Canal, which has a lift of 32½ ft. Such heights used to be considered as marking about the limit of the advantageous employment of locks over mechanical appliances. For greater lifts, the strain on the lower gates became very considerable, and the consumption of water in the majority of cases was more than could be afforded.

It is true that by the aid of side ponds and syphons a large proportion of the water used in lockage could be saved—as much as 50 or 60 per cent. in many cases—but even allowing for this the loss was heavy and a mechanical lift was almost universally substituted.

In modern German practice, however, a very striking advance has been made owing to the development in size of vessels, which have now reached to a tonnage of 600 to 1,000 tons. A lock at Henrichenburg has a lift of 46 ft. It is provided with five side ponds for the economisation of water, the saving amounting to about 76 per cent. The chamber (312 ft. long by 33 ft. wide with 9 ft. 10 ins. depth of water over sill) is built in concrete on a rock foundation requiring an invert only 19½ ins. thick. Other locks have been constructed with lifts up to 60 ft. and over. The former limit, therefore, can no longer be said to apply, except in so far that, if it be exceeded, due provision will have to be made for the increase in strength of the gates and side walls, and for the economisation of water.

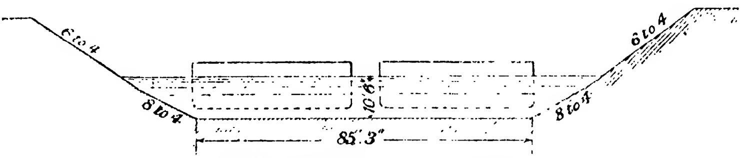


FIG. 16.—Bruges-Ghent Canal (Side Bay).

An appropriate test to apply to the serviceability of locks is the gradient or slope of the canal route. Where this is fairly regular and does not exceed 1 in 100 the rise lends itself to convenient subdivision into reaches of a quarter of a mile each, or more, separated by single locks of about 13 ft. lift. Under ordinary circumstances this is found to be a suitable standard for barge traffic, with barges of moderate tonnage, such as are customary in this country.

Where, on the other hand, the level of the ground changes abruptly, locks of higher lift or a series of locks may be employed, but only if an adequate supply of water is forthcoming to make good the loss from lockage. Where water is scarce it will be necessary to fall back upon some system of mechanical transfer.

Mechanical Lifts are generally more expeditious and time saving in action than locks, but they are also as a rule more expensive, and from their nature they are less durable and more liable to derangement.

From a number of examples of direct acting vertical lifts may be mentioned those at Anderton (Cheshire), Les Fontinettes (France), and Peterborough (Canada), *vide* figs. 17–19.

Lift at Anderton.—In this case the difference in level is 50 ft. The lift consists of two wrought-iron troughs each 75 ft. long by 15½ ft. wide, originally worked hydraulically and counterbalancing one another so that it was only necessary to reduce the depth of water in the lower trough by 6 ins. in order to set the apparatus in action. Owing to excessive wear and tear which developed in the hydraulic rams and cylinders, these were replaced in 1907–8 by a system of suspension gear passing over pulleys on an overhead gantry with counter weights at the free ends of the ropes. The total weight so suspended is 1,000 tons, each of the caissons weighing 250 tons, of which 80 tons is iron and the remainder water. The reconstructed lift is worked electrically. The main 30 B.H.P. motor runs at about 750 revs. per min., and the large 6 ft. pulleys make about 1 revolution in 2 minutes.

Lift at Les Fontinettes, near St. Omer, France.—The lift consists of two iron troughs containing water and each capable of taking a boat of 300 tons. Each trough is bolted at its centre to the head of a ram or piston, working in an hydraulic press. The two presses are in intercommunication by means of a pipe and constitute an hydraulic balance, vertical movement being obtained by admitting a surcharge of water to the upper tank. The surcharge is 12 ins. in depth, equal to about 64½ tons weight. The difference in level is 43 ft. The troughs are 129 ft. 7 ins. long by 18 ft. 4½ ins. wide, with a minimum depth of water of about 7 ft. The ends are closed by lifting gates. The rams or pistons are 57 ft. long, 6 ft. 6¾ ins. diameter, and 2 ft. 8 ins thick. They are formed in sections 9 ft. 2 ins. long, flanged inside, bolted together, and made water-tight by a ring of sheet copper inserted between the flanges. The hydraulic presses are 71 ft. high and 6 ft. 10 ins. diameter, resting upon concrete foundations at the bottom of excavated pits.

When one trough is raised to summit level there is a margin of 1⅞ ins. between its upstream extremity and the downstream end of the canal aqueduct. At the moment of lifting the gates to allow of the passage of a boat in or out, the joint is made good by indiarubber hose running round the end of the aqueduct and protected by springs. This hose is inflated by air at a pressure of 1½ atmospheres. The same connection is made at the lower level.

The machinery consists of two turbines, one of 50 H.P. and the other of 15 H.P.

The actual time of descent and ascent of the troughs is five minutes. An entire operation, including entry and exit of barges, takes twenty minutes.

The total cost of the lift was about 74,800*l*.

Lifts on the Trent Canal, Canada.—**There are two large lifts, one at Peterborough and the other at Kirkfield. The former consists of a pair of tanks, or chambers, 137 ft. long by**

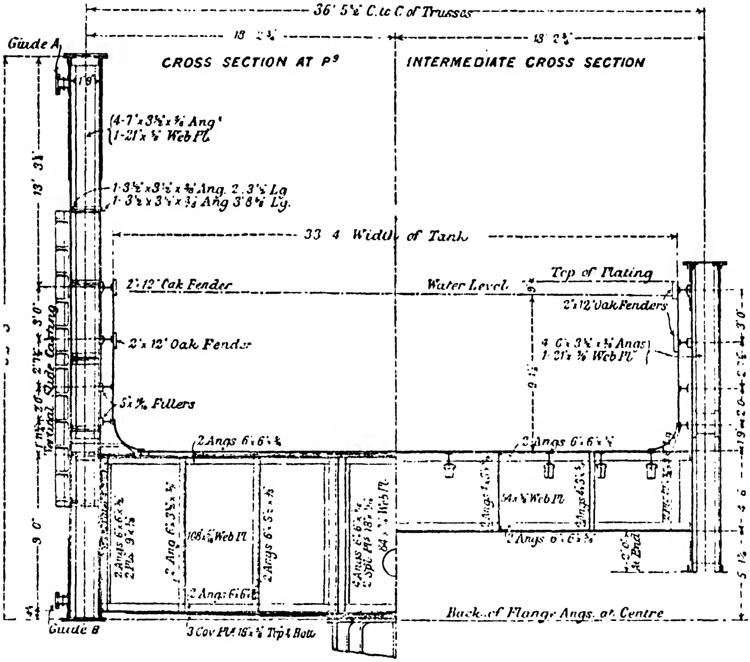


FIG. 17.—Cross-Sections of Canal Lift Lock at Peterborough, Canada.

33 ft. 4 ins. wide, and with 9 ft. depth of water. Each tank is supported by a ram 7 ft. 6 ins. diameter worked hydraulically, so that the descent of one chamber automatically forces up the other. The range of lift is 65 ft. Under ordinary conditions the upper chamber is held in position at a level about 6 or 8 ins. below the limit of stroke, and this surcharge of water is sufficient, when the by-pass is open, to overcome friction and give the necessary movement. The tanks are supported by openwork lattice trusses, and the total load of water and steel is carried to the top of the ram by four plate girders 9 ft. deep. The following is a description of the gates, which are in pairs, one closing the chamber end and the other the canal reach. 'Each gate is made up of three main 15-in. channels, with a small plate and angle girder at the top, and is stiffened vertically by 10-in. I beams. In every space between these vertical I beams there are placed two galvanised iron air chambers about 11 ft. long. These have the effect of making the gates practically buoyant, and this decreases markedly the amount of work which has to be done in raising the gates from a horizontal to a vertical position. The water-tight plating is, in

each case, on the outside of the gate, and the hinges are at the bottom. In the case of the lower reach gates and chamber gates, the hinge for the reach gate is 2 ft. 4 ins. below the hinge of the chamber gate, and 1 ft. 10 ins. separates them in a longitudinal direction. Consequently, the chamber gate is approximately 10 ft. deep, while the reach gate is about 12 ft. 6 ins. deep. As it is never necessary to operate any gate singly, the gates are arranged to work in pairs. At each end of the reach gate there is a pinion meshing into a circular rack set into the dock wall. The centre of the pitch circle of this rack is, of course, coincident with the centre of the hinge of the reach gate. The pinions above referred to are connected by a horizontal shaft running right across the gate, and power is applied to this shaft by means of a sprocket with chain, driven from another shaft, the centre of which is the same as that of the hinge. This shaft is actuated by one of the hydraulic gate engines provided.

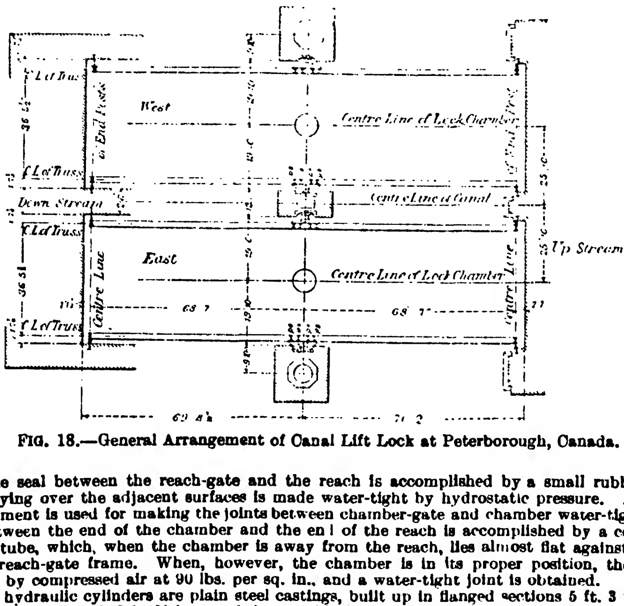


FIG. 18.—General Arrangement of Canal Lift Lock at Peterborough, Canada.

'The seal between the reach-gate and the reach is accomplished by a small rubber strip, which lying over the adjacent surfaces is made water-tight by hydrostatic pressure. A similar arrangement is used for making the joints between chamber-gate and chamber water-tight. The seal between the end of the chamber and the end of the reach is accomplished by a collapsible rubber tube, which, when the chamber is away from the reach, lies almost flat against the end of the reach-gate frame. When, however, the chamber is in its proper position, the tube is inflated by compressed air at 90 lbs. per sq. in., and a water-tight joint is obtained.

The hydraulic cylinders are plain steel castings, built up in flanged sections 5 ft. 3 ins. long. The inside diameter is 7 ft. 8½ ins., and the metal is 3¼ ins. thick. The joint packing is of copper, ¾ in. wide and $\frac{1}{16}$ in. thick, with $\frac{7}{16}$ in. diameter lead wire in press sections. The rams are of cast iron in flanged sections, also 5 ft. 3 ins. long, 90 ins. external diameter, thus giving a water space of 1¼ in. completely round the rams. The jointing is the same as that for the cylinders, except that the lead ring is omitted.

A very full description of this lift appeared in *Engineering* of June 9 and 16, 1916, from which the above particulars were extracted.

Inclined Planes represent a class of mechanical appliances suitable for intermediate conditions, viz., a route gradient too steep for single locks with intermediate reaches, and not sufficiently abrupt at any one point for a vertical lift. Generally speaking, gradients of between 1 in 8 and 1 in 10 are most favourable but they may run up to 1 in 75, or 1 in 80, and, on the other hand, inclined planes have sometimes been adopted where the rise is much steeper as, for instance, at Foxton on the Grand Junction Canal, where the gradient was 1 in 4. The Foxton incline now dismantled was similar in character to the balanced vertical lift, consisting of two caissons travelling transversely, each maintaining a barge of 70 tons afloat. It is claimed for inclined planes that, in comparison with locks, they save water, and, in comparison with vertical lifts, they save time, since a boat is travelling forward simultaneously with its alteration in level. Moreover they do not require such deep or expensive foundations as the lift proper. In spite of these advantages inclined planes have not attained a very marked degree of acceptance among canal engineers.

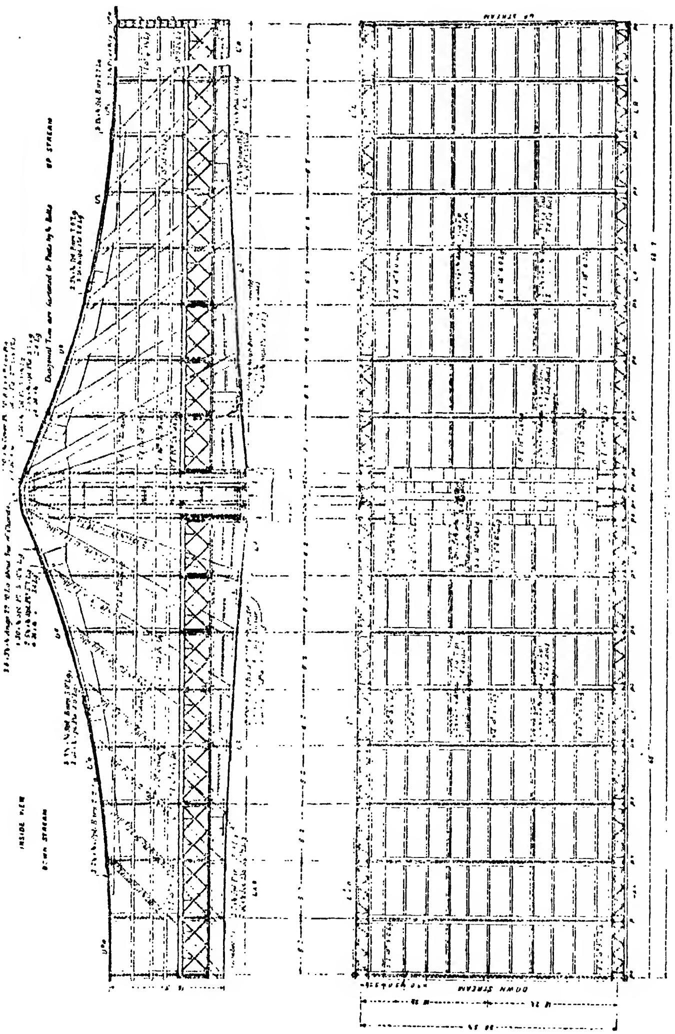


Fig. 19.—Internal View and Plan of Canal Lift Lock at Peterborough, Canada.

Canal Curves.—The formula adopted in Belgium for the increased width at curves is

$$S = \frac{A^2 + a^2}{R}.$$

Where S is the increase in bottom width to be added to the normal semi-width on the convex side of the curve, R is the radius of the curve, and A and a are the half lengths of the boats which have to pass one another. The units are all in metres, and the value of S must therefore be multiplied by 3·28 to bring it to feet. The increased width is splayed into the straight length at a distance ot 2 (A + a) from the commencemant of the curve.

Recent Tendencies in Lock Design.—The following are extracts from the conclusions passed by the International Association of Navigation Congresses at the Congress held at Venice in 1931 :—

'Concerning locks on canals and canalised rivers, a tendency has been noticed to do away with culverts for filling and emptying locks, and to assign the functions of admitting and evacuating the water to the lock-gates themselves, provided these are fitted with sluices of sufficiently large dimensions.

'In any case, when one has to deal with large volumes of water, it will be necessary to devise some special arrangement to reduce the violence of the discharge.

'As regards methods for closing locks and their working appliances, one notes still a tendency in recent years to adopt lifting gates, which present certain advantages, notably that all the moving parts are clear of water when the gate is raised, that the gate can be constructed in such a manner that it may be closed in case of a flow of water in the lock, which gives an added security ; or the lock may be utillseed to discharge water to the lower reach.

'With regard to further progress in the construction of navigation works, the adoption of lifts appears advisable only when we meet with a certain number of special circumstances combined, such as good foundations, a considerable water-height, and the impossibility of pumping the water back at a reasonable expense.

'In appropriate cases sheet piling, especially of steel, has given satisfactory and economical results in the construction of lock chamber walls, cut-off walls, and guiding walls.

'In the case of new canals intended either for navigation or for agricultural purposes (irrigation or drainage canals), as well as in that of already existing canals, one should always take into consideration the technical possibility and the economical advantages of uniting both purposes in the same canal, in the whole of its extension or in part of it.'

Twin Locks on German Canals.—The Furstenburg twin locks on the Oder-Spree Canal have a fall of 14·5 metres (47 ft. 6 ins.). When filling or emptying is to be done the water-level is equalised in both locks ; then one of the chambers is emptied into the downstream reach while the other is filled by water from the upstream reach. The four communicating culverts between the two chambers are closed, either by means of horizontal cylindrical sluice-gates with the Ardfelt system of sliding rings, in which a bell-shaped member with a cylindrical cover constitutes the means of closure, or by means of sliding cylinders on the Freund system, in which two cylinders with segmental partitions are arranged in such a way as to form side walls to the culvert when open. By rotating the cylinders through 90 degs. the partitions meet at an angle and close the passage. The sectional area of culvert is 3 sq. metres in each case.

At the Anderten twin locks on the Mittelland Canal (Hindenburg Lock), where the fall is 15 metres (49 ft.), each chamber is independent of the other, and is provided with 10 storage chambers arranged in two sets of five above one another as shown in fig. 20. The sluices are cylindrical and work in a vertical cast-iron tube passing through the successive floors of the chambers, sliding up or down to open or close.

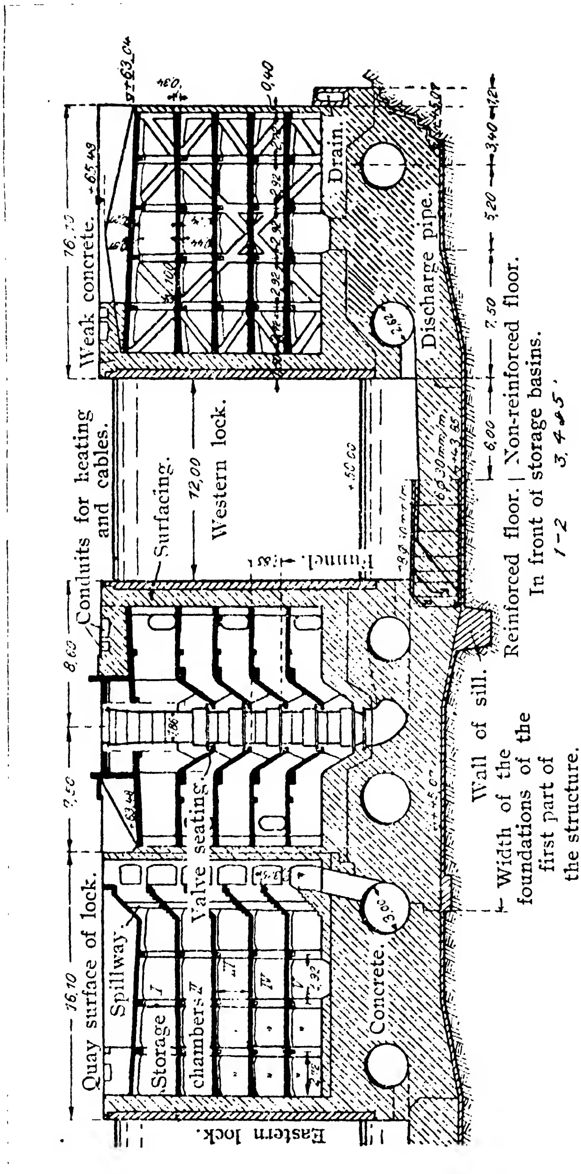
LOSS OF WATER ON CANALS.

Rankine summarises the sources of loss of water on canals as follows :

Waste of Water by leakage of the channel, repairs and evaporation *per day* = area of surface of canal × $\frac{1}{6}$ of a foot, nearly.

Current from the higher towards the lower reaches, produced by leakage at the lock gates, *per day*, from 10,000 to 20,000 cubic ft. in ordinary cases.

Lockage.—Single locks are more favourable to economy of water than flights of locks. At a single lock, single boats ascending and descending, alternately, cause less expenditure of water than equal numbers of boats in trains. On the other hand, at a flight of locks, boats in trains cause less expenditure of water than equal numbers of boats ascending and descending alternately. For this reason where a long flight of locks is unavoidable, it is usual to make it double, using one flight exclusively for ascending boats and the other exclusively for descending boats.



Section taken at 24,40 and 145,30 m. distance from the overflow wall of the upstream head.

FIG. 20.—Cross Section of the Storage Chambers of the Anderten Twin Locks on the Mittelland Canals.

Loss by Percolation from Erie Canal.
In Cubic Feet per Second per Mile of Canal.

Location.	Material in Bank.	Cub. ft. per Sec. per Mile of Canal.
Near Dunsbach Ferry	Gravel	1·68
Near East of Schenectady	Sandy loam	1·50
Near West of Schenectady	Gravel	2·24
East of Amsterdam	Gravel	3·03
West of Amsterdam	Coarse gravel	12·89
Canajoharie	Gravel	2·00
Fort Plain	Gravel	6·85
East of Utica	Sandy loam	1·45
East of Rome	Sand and gravel	8·48
West of Rome	Clay loam and gravel	1·75

NOTE.—The Erie Canal has bottom widths of 52½ ft. and 42 ft. and depths of 7 ft. and 9 ft. in various portions.

Canal Bank Protection in Italy.—The sandy banks along the Candiano Canal in the province of Ravenna are strengthened with a row of reinforced concrete piling. The king piles, 10 to 16 ft. long and 8 ins. square, are driven at intervals of 10 ft. They are secured together at the top by a longitudinal runner 8 ins. by 8 ins., behind which is driven a row of sheet piling from 5½ to 6½ ft. long and 2 ins. thick. The length of the piles depends upon the nature of the ground and the depth of water, and, where necessary, further security is obtained by anchorage tie rods and bars to short piles in the rear. At points where it was possible to remove the water from the canal, or where the bottom was occasionally exposed, it was considered desirable and economical to revet the banks with a surfacing of reinforced concrete 2½ ins. thick in which the reinforcement consisted of expanded metal ·4 to 1·6-in. mesh and ·2 to ·32 lbs. per sq. ft. Where the canal bottom is not subject to erosion during floods, this revetment is continued down as far as 18 ins. above the level of the bottom, and the toe is protected by rubble. In cases of erosive action there is a concrete foundation or sill, sometimes supported by piling; and where the water level cannot be lowered sufficiently for the formation of a sill, the toe of the revetment rests against sheet piling.

TRACTION AND HAULAGE.

Means of Traction.—On ordinary small barge canals in this country, horse haulage has maintained its predominant position, but tug traction has been substituted on some of the larger systems, generally with successful results, provided the banks are sufficiently protected to withstand the wash of the tug propellers. Locomotive traction is common along the banks of continental waterways, but has not been able to establish itself on British canals where conditions are less favourable. Some craft are self-propelled, especially on larger waterways, and ship canals are, of course, navigated by vessels under their own steam. For passage through the locks of the Panama Canal electric locomotives are provided. There are 56 locomotives in service, each of which is 32 ft. long overall, has a wheelbase of 12 ft., and weighs 86,000 lbs. The gears are arranged to provide a maximum speed of 2 miles per hour on the inclines, or while towing, and 5 miles per hour when returning light. For vessels up to 350 ft. in length, four locomotives, two on each side at bow and stern, are employed. For larger vessels, six, eight and ten locomotives are assigned.

Horse Traction on a straight reach is most effective at speeds between 2½ and 3 miles per hour. Within these limits a steady pull of between 100 and 125 lbs. can be maintained, which is sufficient for loads up to 60 tons. There is the further consideration that at higher speeds the wash against the banks becomes excessive for unprotected surfaces. Where tugs or locomotives are employed some form of bank protection becomes essential.

As regards river navigation Rankine gives the following formula as a rough estimate of the effect of the current of a stream on the load which one horse is able to draw against at a walk.

Load drawn against current = load drawn in still water $\times \left(\dfrac{3\cdot6}{3\cdot6+v}\right)^2$; v being the velocity of the current in feet per second.

Cable Towage.—Lévy system, as adopted on St. Maur Canal between Paris and Joinville.—Cable weighs 2·45 lbs. per lin. ft. Set with permanent tension of five tons. Breaking strength 50 tons. Deflection due to haulage does not exceed 4 ins. Vertical supporting pulleys are 2 ft. 7½ ins. diameter, and have depth of groove of 7⅞ ins. Circuit may cover distance of 10 to 11 miles, so that two driving machines may be set 20 to 22 miles apart. For a speed of 1½ miles per hour 2 h.p. is required to draw a barge loaded with 350 tons. For a speed of 2½ miles an hour 4½ h.p. is required. Tractive force $= \dfrac{Wv^2}{2gl}$, where W = weight of boat in lbs., v = velocity in ft. per sec., g = acceleration due to gravity = 32 ft. per sec. per sec., and l = length of tow-line in feet. For a boat of 300 tons a velocity of 3 ft. per sec. and length of line of 300 ft., the tractive force = 315 lbs. Estimated cost of plant, about 12*s.* 6*d.* per lin. yd. Cost of working ranges from ½*d.* to ¼*d.* per ton per mile.

(*Report of Dr. Watson to Water Commerce Congress at Chicago*, 1893.)

Electric Traction on an elaborate scale has been installed on the Teltow Canal in Germany since 1905. The tractors run on rails on both banks. The current is continuous at 550 volts, from overhead mains, and each locomotive is driven by two 8 h.p. motors, which are sufficiently powerful to haul two 600-ton barges at a speed of 2½ miles per hour. A similar system is in use on the Douai section of the Canal d'Aire, where the tractors are of 10 h.p. In 1918, some petrol locomotives of 20–25 h.p. were introduced on the Rhine-Rhone Canal in place of horses, but though the experiment was successful the system did not come into general use on the Canal because it would have entailed considerable alteration in the permanent works.

The following advantages among others have been claimed for the use of electric tractors in place of steam tugs :—

1. Assuming that the electric lines are laid on each bank, there will be greater regularity and uniformity of speed in the service than where tugs are used, and there will be less delay in despatching cargoes at short notice and at regular intervals.

2. The banks and bottom of a canal are not so much injured by electric-towing as by tug-towing. In the latter case the peculiar disturbance to the bottom, with consequential injury to the side banks, caused by the motion of the screw, is added to the wave from the moving boats common to each mode of transport.

3. In the case of electric towing there is considerable economy of space and water in locks and lifts, because tugs and steam boats, which by reason of their machinery, have a smaller cargo-carrying capacity than barges of like dimensions, need not be taken through.

4. Electric power, when once installed along a line of waterway, can be utilised not only for traction, but also for operating locks and other works, and lighting the waterway at night, and can even, perhaps, be utilised for factories establishing themselves along the waterway.

(*Report of Royal Commission on Waterways*, 1909.)

In 1924 there was installed, on a short section of the canal from the Marne to the Rhine, a new system of electric traction known as the Chenean system. The underlying principle is the utilisation of an electric tractor, with proportional contact, travelling on an aerial cable. Owing to the light weight of the tractor (about 12 cwts.) it is possible to space the supports of the aerial cable, in general, at about 20 yards distance. The cable is of steel, 9 ins. diameter, weighing 1¾ lbs. per lineal ft. The speed of movement varies from 1 mile per hour in narrow cuttings, tunnels, etc., to 2 miles per hour and over in the open.

Tractor Boats.—In Germany small motor boats with Diesel engines of 17 to 34 h.p. were introduced a few years ago for towage between ports and loading-stations on canals. They are 8 metres in length, with a beam of 2 to 2½ metres, and are operated by one man. They can also be attached to the after-part of a vessel so as to play the part of a screw propeller.

'They are very mobile, easy to steer and have considerable tractive power. They have proved to give good service and to be economical. We are even going as far as to plan their construction on a somewhat larger scale (60 h.p.) for regular towing service on small canals. They can be advantageously used for a standard service, as auxiliary tugs, at the entrance and exit of locks' (*MM. Hoebel and Paxman, International Navigation Congress*, 1931).

Caterpillar Tractors.—Another form of towage used in Germany is that of caterpillar tractors of about 50 h.p. A winch drum for the haulage cable is arranged in front of the tractor, and as soon as the pull becomes excessive, a friction clutch automatically slackens the cable, which can also be lengthened or shortened on the drum, as required, during haulage. The cable can be lifted so as to clear craft lying alongside the canal banks by means of a mast 3 metres in height.

MECHANICAL TRACTION ON FRENCH WATERWAYS

Electric Traction.—The system consists of a rail track of metre gauge, running along one bank. The engines using the track weigh from 10 to 15 tons ; they have two driving axles and are generally equipped with a single motor, from 10 to 15 kw., supplied with power from a 600 volt D.C. trolley line. The rails are laid on wood or metal sleepers, according to the sections, and at certain points have to cross the waterways, in order, for example, to reach a bank free from other installations. The traction current is supplied by sub-stations situated along the waterway, which receive a three-phase current the tension of which can reach 31,000 volts and which in most cases transform the alternating current by means of mercury vapour rectifiers. To ensure safety of distribution every sub-station possesses an emergency equipment.

In short sections, where it is impracticable to lay a rail track, but where the importance of traffic demands electric traction, the difficulty has been surmounted by the use of electric engines on tyres and a double trolley line, the distribution being ensured as before.

Diesel Motor Traction.—Electrification is used only for heavy and medium traffic. For light traffic, engines on tyres are employed. These run along the towpath without any rails, on four wheels, each of which is used for propulsion and driving, with 5 h.p. diesel motors, having one or two cylinders to run on gas oil or fuel oil. This type of engine is used almost exclusively on non-electrified canals.

Special Forms of Traction.—Special systems of traction are used on a limited number of sections, mostly underground, which include electric tow-boats with the chain under water (St. Quentin Canal), electric tractors suspended from a mono-rail (Souterrain de Braye), and funicular cables (Mont Billy).

L. P. Alvin, 'Mechanical Traction on the French Waterways,' *Journal of Institute of Transport*, vol. 17, no. 8.

SHIP CANALS.

General Considerations.—It is desirable that the ratio of cross-sectional area of the canal to the immersed section of ships, where the traffic is commercial, should not be less than 4 to 1, and it may with advantage be much higher. In the Suez Canal it is about 7 to 1. For vessels of war, the ratio may be slightly less, as, for instance, in the Kiel Canal, where the ratio is about 3½ to 1. The limiting value is, however, dependent on the speed at which vessels are permitted to travel. In the artificial portion of the Suez Canal the speed is limited to between 6 and 7 knots, and about 8 knots represents an approximate upper limit for narrow shallow waterways. In passing through such waters vessels sink to a lower level than in the open sea on account of the displacement. Thus pilots have a rough rule that when a ship is near the bottom her 'squat' or 'sinkage' in feet may be about one-fifth of her speed in statute miles per hour.

For curves, M. Quellenec, late Consulting Engineer to the Suez Canal Commission, recommended widening the bottom, so that the longest vessel may preserve in its own direction a free space equal to 1½ times its own length between the stem and the slope of the concave side. The formula used on the canal was:

$$W = \sqrt{R^2 + (2a)^2} - (R + b)$$

where,
- W = amount of widening in feet;
- R = radius of axis in feet;
- a = length of ship in feet;
- b = ½ bottom width (on tangent) of canal in feet.

Kiel, or North Sea and Baltic Canal.—This canal, which connects the North Sea and the Baltic, was originally constructed between 1887 and 1895 at a cost of about 7,800,000*l*., including land purchase. It has a length of 61½ miles. The former dimensions have since been very considerably increased—viz. the bottom width from 72 ft. to 144½ ft., the surface width from 226 ft. to 334½ ft., the depth from about 30 ft. to 36 ft., and the cross-sectional area from about 4,500 sq. ft. to 8,868 sq. ft.

At the entrances at each end of the canal there are now twin locks of 1,083 ft. available length, by 147½ ft. clear width and 45 ft. 2 ins. sill depth at mean water level. These locks take the place of the earlier locks, which were 492 ft. long by 82 ft. wide with 32 ft. of water over sill. The new locks are constructed in concrete, and each has three pairs of caissons dividing the chambers into lengths of 725 ft. and 330 ft.

Manchester Ship Canal.—The Manchester Ship Canal was constructed between 1887 and 1893. It has a total length of 36 miles and includes four sets of locks, with an average

MANCHESTER SHIP CANAL: TYPE SECTIONS.

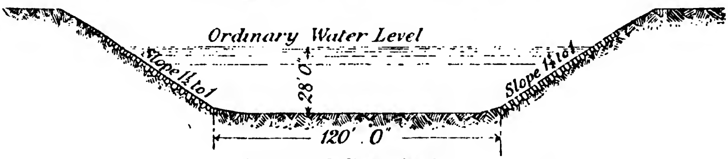


FIG. 21.—Ordinary Section.

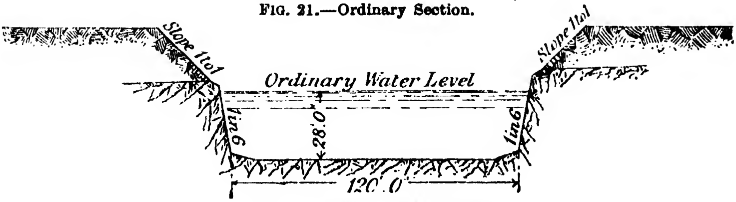


FIG. 22.—Section through Rock.

lift of 15 ft. each, about 160 acres of docks with warehouses and railway sidings, six high level bridges. seven swing bridges, and a swing aqueduct. The cost was 13,500,000*l.*, including land purchase and promotion, the actual constructional cost being about 9,000,000*l.* The capital of the company has since been increased on account of extensions and new works to 20,000,000*l.* The bottom width is 120 ft. The original depth was 26 ft., but this has now been increased to 28 ft. Deepening operations are being continued. A depth of 30 ft. has been extended from Eastham Locks as far as Stanlow Oil Dock, a distance of 4½ miles. The average annual cost of maintenance, which, before the war, was stated to be about 2,000*l.* per mile of canal, at the present time is about 5,400*l.* per mile.

Suez Canal.—The length of the Suez Canal is about 100 miles. It was opened in 1869, the cost of construction, apart from promotion, etc., having then amounted to 10.680,000*l.* It has been progressively deepened, so that the authorised maximum draught for ships in the canal has been as follows:—

From 1870 to 1890	7 m. 50 = 24 ft. 7 ins.
,, 1890 to 1902	7 m. 80 = 25 ft. 7 ins.
,, 1902 to 1906	8 m. 00 = 26 ft. 3 ins.
,, 1906 to 1908	8 m. 23 = 27 ft. 0 ins.
,, 1908 to 1914	8 m. 53 = 28 ft. 0 ins.
,, 1914 to 1915	8 m. 84 = 29 ft. 0 ins.
,, 1915 to 1920	9 m. 15 = 30 ft. 0 ins.
,, 1921 to 1924	9 m. 45 = 31 ft. 0 ins.
,, 1925 to 1929	9 m. 75 = 32 ft. 0 ins.
,, 1930 to 1935	10 m. 00 = 33 ft. 0 ins.
,, 1936	10 m. 35 = 34 ft. 0 ins.

The canal is available throughout for vessels drawing 34 ft. of water. The present depth of the canal below the mean level of L.W.S.T. is equal at least to 40 ft. or more, over the whole length; maintenance dredging is being carried to a depth of 44 ft. The width of the canal at a depth of 33 ft. is at least 196 ft. The surface width depends on the inclination of the banks, which changes with the nature of the ground through which the canal passes. This width varies normally between 400 ft. and 500 ft.

Of the total length of about 100 miles, 12 miles only are in curves, the minimum radius being 2,734 yards, and the maximum radius 8,083 yards, most curves having a radius of about 3,300 yards.

SUEZ CANAL.

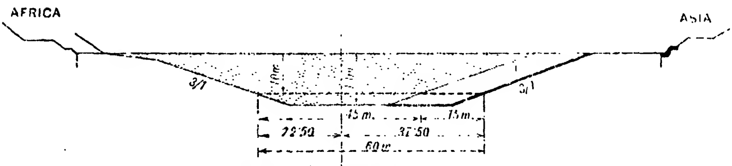


FIG. 23.—Section of Suez Canal, showing widening.

There are no locks. the canal being level throughout. At the South entrance there is a tidal range, amounting to about 5 ft. in the average at springs. The current flow does not exceed about 3½ ft. per second.

Panama Canal.—The total length of the Panama Canal, measured along its axis, is 50½ miles, but the portion actually within land area between coast lines is only 41½ miles. Of the total length, 14½ miles are at sea level, over 23½ miles in Gatun Lake, nearly 3 miles in locks of alongside approach walls, 1½ miles in Miraflores Lake, and 8 miles in the Gaillard Cut. The summit level is 85 ft. above sea level. The depth lies between 40 and 45 ft. The bottom width varies as follows: For 16 miles the canal is 1,000 ft. wide; for 4 miles, 800 ft.; for 22 miles, 500 ft.; and for 8 miles, 300 ft. wide.

There are six pairs of locks with dimensions of 1,000 ft. in serviceable length and 110 ft. in width; three of these are at Gatun, one at Pedro Miguel, and two at Miraflores. The sills are laid so as to afford a depth of 40 ft. in salt water, and 41 ft. 4 ins. in fresh water. With the exception of the lower lock in the Miraflores flight, the chambers are provided with intermediate gates, dividing them into two sections of 600 ft. and 400 ft. The range of lift of the Pedro Miguel Lock is 30½ ft. All the locks are double, i.e. with parallel chambers, so that vessels may pass in opposite directions. At the Gatun flight the lift is 85 ft. divided among the three locks; the actual operating pressure between consecutive locks is therefore nearly 60 ft.

There are 46 pairs of lock gates, constructed of mild carbon steel with yokes and pintles of vanadium steel. The height of the gates varies from 46 ft. to 82 ft. The largest gate weighs about 700 tons.

DIAGRAMMATIC SECTIONS OF PANAMA CANAL. (Not to Scale.)

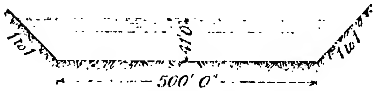


FIG. 24.—From Mindi River to Gatun Locks, and from Miraflores to Panama Bay.

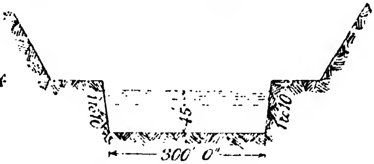


FIG. 25.—Gaillard (formerly Culebra) Cut.

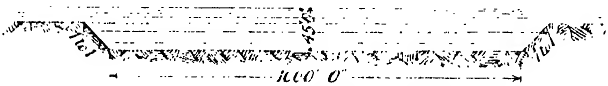


FIG. 26.—From Gatun Locks to Angle North of San Pablo.

The approximate cost of the canal, including purchase money, is 75,000,000*l*.

The following is a statement of the unit cost of excavating the Culebra (now Gaillard) Cut for the year 1912:

	Cost per cu. yd.
Transportation	\$0·1331
Drilling and blasting	·1157
Tracks	·0885
Loading by steam shovels	·0681
General expenses and supervision	·0503
Dumps	·0479
Plant, arbitrary	·0395
Drainage, structures and clearing	·0045
Total unit cost	\$0·5476

Bruges Ship Canal.—The canal connecting Bruges (Belgium) with the sea was constructed between 1896 and 1907. The works comprised an outer harbour with breakwater at Zeebrugge, and a sea-lock with a chamber 518 ft. long, 65½ ft. wide, the sill being laid at 19 ft. 8 ins. below L.W.O.S.T., giving a depth of 31 ft. 8 ins. at H.W.O.N.T. The canal proper is 6¼ miles long, 72 ft. wide at the bottom, and 229 ft. wide at water surface level. There is a depth of 26 ft. throughout, and the side slopes are about 3 to 1. Two railway and three road bridges cross the canal. At Bruges there are two basins and a lock connecting with the internal barge canal system of Belgium.

Amsterdam, or North Sea Canal.—This canal, constructed 1865–76, connected Amsterdam with the North Sea, along a route due west of 15½ miles, and supersedes in utility the Amsterdam North Sea Ship Canal, or Great North Holland Canal, made between 1819–25 which has a northerly direction, is 52 miles long, and is only suitable for vessels drawing not more than 16 ft. As originally constructed, the new canal had a depth of 23 ft. and a bottom width of 88½ ft. The entrance locks at the port of Ymuiden were two in number, the larger having an available length of 394 ft., a width of 60 ft., and a depth over sill of 26 ft. 6 ins. at ordinary low water. In 1896 a new and larger lock was provided, 737 ft. long, 81 ft. wide, and 33 ft. deep over sills. Yet again, in 1930, a third large lock was opened, at present the largest (though not the longest) in the world; it is 1,312 ft. long, 164 ft. wide, and it has a depth of 49¼ ft. of water over sills. The canal itself is in process of being deepened to 49¼ ft., and will have a bottom width of 328 ft.

The cost of the original works, including the harbour at Ymuiden, which has a sheltered area of about 250 acres enclosed by two breakwaters springing from the shore, each 1,670 yards in length, 3,940 ft. apart at the shore line and 853 ft. apart at their pierheads, was about £3,000,000. Improvement and maintenance works (including supervision) brought the total up to £5,667,000 by the end of 1910. The cost of the latest lock was about £1,600,000.

New Welland Ship Canal, Canada.—The new ship canal between Lake Erie and Lake Ontario, completed and opened in 1932, is designed to serve large-size bulk freighters of the present day which are about 600 ft. in length and draw up to 21 ft. of water. The work was begun in 1913, but was interrupted between 1917 and 1919 on account of the war. The cost has somewhat exceeded $120,000,000.

For all practical purposes of navigation the new canal is a straight line throughout. It has a length of 27·7 miles between its outermost ends at Port Weller and Port Colborne. Its present depth is 25 ft.; later this will be increased to 30 ft. Sufficient clearance is provided so that any ship passing through the lock at Sault Ste. Marie (the outlet from Lake Superior) can continue its voyage through to Lake Ontario. Later the Canadian Government expects to provide a waterway of equal dimensions from Lake Ontario down the St. Lawrence River to Montreal, whence there is ample waterway to the Gulf of St. Lawrence. There is a difference of 221 ft. in level between Lake Ontario and the river at Montreal.

The difference in level between Lake Erie and Lake Ontario is 325·5 ft. This is overcome in the new canal by 7 locks, each of which, rising 46·5 ft., has a usable length of 820 ft., a clear width of 80 ft., and a depth over sills of 30 ft. Two additional guard locks are provided, one at each end of the canal to meet high and low water differences in the level of the lakes. The guard lock at Humberstone (No. 8) is notable as being the longest lock in the world, having a length of 1,380 ft. between inner gates. Its nearest rivals are two locks on the American side at Sault Ste. Marie, which are 1,350 ft. long.

The superseded Welland Canal overcomes the difference in level between the lakes by means of 26 lifts. The locks have a usable length of 270 ft., a clear width of 45 ft., with depths over sills of 12 to 14 ft. The constructional cost of the canal was $31,031,226, and the maintenance cost to the year 1920 amounted to $9,237,961.

Moscow-Volga Canal.—The canal connecting Moscow with the river Volga and thus with the sea, which was opened in May 1937, has a length of 80 miles, the actual length of canal being nearly 68 miles and the remainder consisting of lakes and reservoirs. The canal is 150 ft. wide at the bottom and 280 ft. wide at the surface and has a minimum depth of 18 ft. The most difficult constructional feature was surmounting the elevated ridge which separated the Volga from the Moscow River—this has been achieved by a series of five locks, each providing a lift of 26½ ft. with two corresponding double chamber locks for the descent.

The **Mittel-land Canal**, in Germany, completed in 1938, is the culmination of a great scheme of internal navigation in that country, connecting the waterways of the Oder, the Elbe, the Weser and the Rhine, and placing Berlin in direct communication with the industrial region of Westphalia. The canal accommodates craft of 1,000 tons carrying capacity (29½ ft. broad and 6½ ft. draught). Among the more notable engineering features of the undertaking are the twin locks at Anderten (near Hanover) with a vertical range of 49 ft. (15 metres), see fig. 20, p. 596, and the mechanical lift at Rothensee (Magdeburg) with a vertical range of about 60 ft. (18 metres).

The **Albert Canal** in Belgium, which was put into service at the close of 1940 after ten years of constructional operations, has a length of close on 100 miles and serves to connect the industrial city of Liège with the port of Antwerp. The total fall between the extremities is 184 ft. and changes of level are negotiated by six sets of locks, five of which have a fall of 33 ft. each and the sixth one of 19 ft. The canal has a capacity for craft of 2,000 tons deadweight.

SECTION XVII

HARBOURS AND BREAKWATERS — DOCKS — QUAY-SIDE APPLIANCES—GRAVING AND FLOATING DOCKS — DREDGING—SUBAQUEOUS ROCK REMOVAL—DIVING—FLOATING LANDING STAGES

(pp. 605–644)

(Contributed by Brysson Cunningham, D.Sc., B.E., F.R.S.E., M.I.C.E.)

CHANNEL DEMARCATION—BUOYS—LIGHTSHIPS—LIGHT-HOUSES—SUBMARINE SIGNALS

(pp. 645–656)

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SECTION XVII

HARBOURS AND BREAKWATERS — DOCKS — QUAY-SIDE APPLIANCES — GRAVING AND FLOATING DOCKS — DREDGING — SUBAQUEOUS ROCK REMOVAL — DIVING — FLOATING LANDING STAGES.

Contributed by Brysson Cunningham, D.Sc., B.E., F.R.S.E., M.I.C.E.
(Author of 'Dock Engineering,' 'Harbour Engineering,' etc.)

CHANNEL DEMARCATION—BUOYS—LIGHTSHIPS LIGHTHOUSES SUBMARINE SIGNALS

(Contributed by J. P. Bowen, C.B.E., B.Sc., M.I.C.E.)

HARBOURS AND BREAKWATERS

Harbours are sheltered areas of water affording safe anchorage for ships, generally also, when discharging the functions of a port, provided with quays and facilities for landing passengers and cargo. There are many harbours where the requisite shelter is forthcoming naturally from the physical configuration of the coast line (Queenstown, Rio de Janeiro, Sydney, etc.). Where protection cannot be obtained in this way it has to be provided by means of breakwaters, as at Portland, Dover and elsewhere.

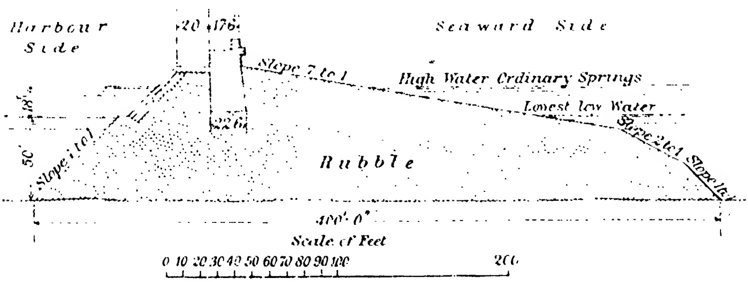


FIG. 1.—Holyhead Breakwater.

Breakwaters are of two types: (*a*) the mound of rubble stone, and (*b*) the upright wall of masonry or concrete. Frequently the types are combined. Sometimes the mound predominates and is simply capped by a slight superstructure of regular masonry, as at Algiers and Oran; in other cases it is reduced to a minimum, becoming a mere foundation layer for a wall of massive proportions, as at Ymuiden and Zeebrugge.

The relative advantages of the mound and the wall are as follows :—

1. As regards *efficiency*, the mound is not so serviceable as the wall. The undulations of the sea are transmitted through the interstices of a mass of rubble, and the harbourage area is kept more or less in a state of agitation.

2. As regards *initial cost*, the mound can be formed by unskilled labour, and where that is plentiful and stone abundant, this type of breakwater will commend itself on account of the facility with which it can be constructed. In the absence of these essential conditions, and provided the depth of water be moderate and the foundation sufficiently firm, the wall, involving a much less quantity of material, will be found preferable. Especially will this be the case where skilled labour is readily procurable at reasonable rates. Even the difficulty of a defective foundation may be overcome by one or other of several expedients without perceptibly altering the relative positions. But where the sea bottom lies at a considerable depth (40 feet or more) the superior economy of the pure wall cannot be maintained.

An interesting comparison of the estimated (pre-war) cost of four different types of breakwater, based on the conditions at Zeebrugge, Belgium, is made by MM. Bonnet and Braeckman in a paper to the Fourteenth International Congress of Navigation. The depth of the sea bed at Zeebrugge is 26 ft. 3 ins. below zero datum, and the tide rises 15 ft. above zero, giving a total depth of 41 ft. 3 ins. of water. The unit costs per lineal metre (3·28 ft.) are given as follows :—

	Francs.
(*a*) Solid wall of the Zeebrugge type. Total height 66 ft., mean thickness 23 ft.	**6,760**
(*b*) Composite breakwater with mound of unassorted rubble, the slopes being lightly covered by large natural blocks of 10 cwts. to 6 tons ; wall superstructure 33 ft. high	**7,600**
(*c*) Composite breakwater, mound having core of unassorted rubble, larger material on slopes protected on outer side by artificial blocks of 33 tons deposited haphazard	**8,700**
(*d*) Composite breakwater with base as before, but with slopes protected by artificial blocks of 31 tons laid symmetrically to inclination	**9,000**

Constructional Cost of Breakwaters.*

Harbour.	Type.	Mean Depth of Water.	Cost per lin. ft.	Remarks.
		Ft.	£	
Alderney .	Composite	90	259	Mound terminates 12 ft. to 15 ft. below L.W.
Algiers. .	Mound	50	122	Large random concrete blocks surrounding rubble.
Dover . .	Wall	40	415	Concrete blocks weighing 26 to 40 tons.
Holyhead .	Composite	54	162	Mound very predominant.
Marseilles .	Mound	35	87	
Bizerta .	Composite	58	60	Wall built on caisson system.
Portland .	Mound	44	127	
Naples . .	Composite	93	186	
Valparaiso .	Composite	60 to 150	560	Concrete monoliths on rubble mound.

3. As regards the cost of *maintenance*, this is generally greater in the case of mounds than in the case of walls. A wall, provided it be properly constructed in the first instance, calls for no further attention, save for such rare and occasional damage as results from some storm of exceptional severity. The rubble mound, on the other hand, is peculiarly susceptible to wave action, and needs constant replenishment. The cost of this varies very considerably according to

* The impossibility of accurately adjusting costs and prices, where they are quoted, to the **altered and variable conditions arising has rendered it advisable to retain them on their old pre-war basis, leaving the necessary adjustments to be made as occasion may require out of war-time conditions and influences.**

situation and exposure. At Holyhead it was for many years as low as 1*s*. 3*d*. per lin. ft. per annum, but latterly there has been some extensive replenishment at considerable outlay; at Genoa, the

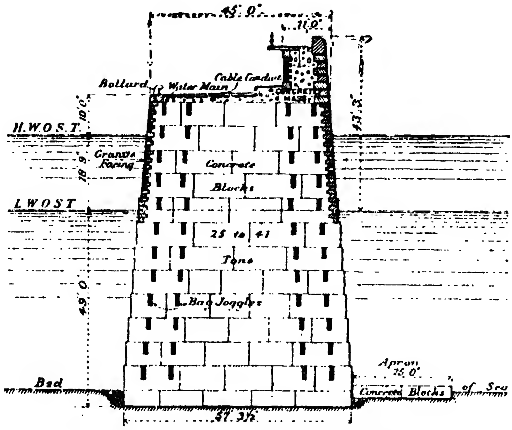


FIG. 2.—Admiralty Pier Extension, Dover.

cost is stated to be 7*s*.; at Naples, 13*s*.; while at Alderney, during a certain period, it rose as high as 45*s*. This last, however, is quite an abnormal figure, and under ordinary conditions the pre-war annual expenditure on mounds of suitable design hardly exceeded a pound a foot.

Wave Action.

Wave Action.—Wave action on breakwaters is most keenly marked between the levels of high and low water, but it extends to a depth of 30 ft., and there are instances on record of serious disturbances at even greater depths. Thus, at Peterhead, blocks weighing upwards of 41 tons have been displaced at 36½ ft. below L.W.O.S.T. But cases of this kind are exceptional.

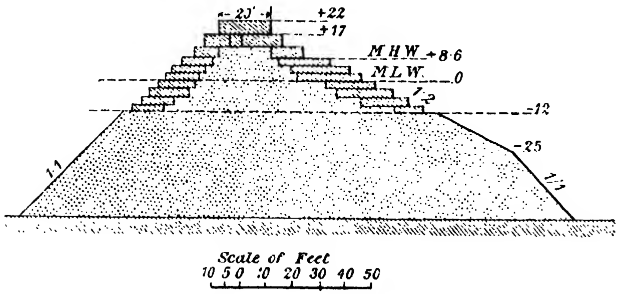


FIG. 3.—Sandy Bay (U.S.A.) Breakwater. 1902 Project.

Generally speaking, at a depth of 5 fathoms the sea is fairly quiescent and a rubble mound may be counted upon to stand therein with side slopes of from 45° to 50°.

Above this level much flatter slopes become necessary: 2, 3, 4, 5, and even 7 to 1. Unfortunately, while the equilibrium of the mound demands such gradual inclinations, they serve also to foster the disruptive action of waves, which, passing up a long seaward incline, have their oscillatory character changed into one of translation. At the summit the impact delivered is so powerful as to produce a pressure which Stevenson records as being six times as great as that against a steep wall. This inimical effect can only be counteracted by the use of huge blocks and monoliths, covering the exposed surface of the mound. The blocks, which in practice rarely weigh less than 25 or 30 tons apiece, and often considerably more, may be deposited either in courses or at random. In the former case they are generally stepped so as to conform to the general inclination. In either arrangement they should be laid as headers, pointing seaward, so as to expose the least extent of surface to wave-action and at the same time gain the utmost advantage from the moment of resistance to overturning. Hence prisms or parallelopipeds are decidedly preferable to cubes.

The *maximum pressure* exerted by waves does not appear to exceed 3½ tons per sq. ft. At Skerryvore, on the Atlantic, a pressure of from 2½ to 2¾ tons per sq. ft. has been observed; at Bell Rock, in the North Sea, 1½ ton; at Dunbar (East Lothian) 3½ tons, and at Buckie (Banffshire) 3 tons.

Composite breakwaters, in which the wall plays any important part at all, should be so designed that the *back draught* or *after-tow* of the waves cannot undermine the superstructure. To be really effective, this means that the wall should be carried down to a depth considerably below low water, and though in the past many such walls have been founded at low water level, or very slightly below it, yet the experience gained has proved that this depth is quite inadequate (the low water zone is particularly dangerous), so that of late years there has been a continual tendency towards increasing the depth of the foundation level. At Madras the wall commences at 27½ ft.; at Naples, 33 ft.; at Peterhead, 43 ft.; and at Algiers, 49 ft.*

Methods of Construction.

Methods of Construction.—Breakwaters are constructed in several ways, according to situation and type.

For Mound Breakwaters, rubble may be deposited—(*a*) from barges or scows; (*b*) by means of cranes running on temporary staging; and (*c*) from wagons passing over roads laid at or about the level of the top of the mound and extended as the work proceeds.

The first method is best adapted to sheltered situations, and affords an opportunity of depositing rubble uniformly and simultaneously over the whole site, but it is not economically practicable to carry on operations in this way at or about surface water level nor above it. Moreover, the progress of the work is very dependent on weather conditions, and there may be long periods of enforced inaction. It is to be noted in passing that irregular deposits of rubble lead to current action and scour. The second method (staging) admits of operations being carried on uniformly over the site, though not with quite the same degree of freedom as in the case of barges. But the advantages of a stable working base with infrequent interruptions and higher degree of safety and protection for workers, are very pronounced. The damage sustained by timber staging in storms and from the attacks of marine borers are drawbacks which detract from the merits of the system, though, generally speaking, the former is proportionately not a serious matter, and any evil results of the latter can be to some extent guarded against by careful inspection. The third, or low-level system of construction, by means of tipping from wagons running on tracks laid

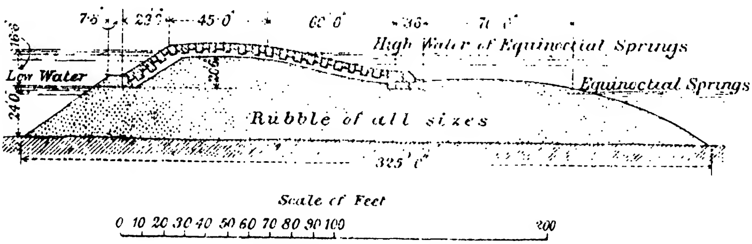


Fig. 4.—Plymouth Breakwater.

* **Even the last-named depth did not prevent the collapse of a length of 400 metres of the Mustapha Mole during an unprecedented tempest in February 1934.**

on the breakwater itself, assists in the consolidation of the work as it proceeds, and saves the cost of staging, but the area of operation is limited and there is liability to interruption during the

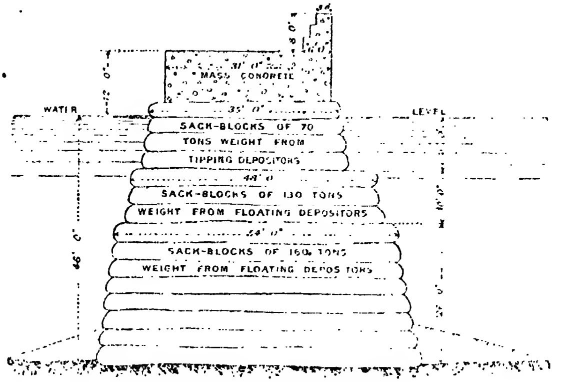


FIG. 5. - La Guayra Breakwater.

prevalence of heavy seas. It is a method more adapted to small embankments, where time also is perhaps not a matter of prime importance.

Wall-Breakwaters.

Walls are usually constructed either by means of staging, or on the low-level, end-on system; the former being generally more suitable when concrete is deposited in mass, and the latter for blockwork. The blocks, or monoliths, are deposited in place by huge revolving cranes, known as Titans, and overhead travellers, known as Goliaths. A modern Titan is a powerful cantilever crane capable of dealing with blocks of from 30 to 60 tons weight, at a working radius of as much as 100 ft. A 41-ton steam Titan used on the breakwater at South Shields had a radius of 75 ft., and was carried on 16 central-flanged wheels running on rails set to a 22 ft. gauge. The speeds of the various motions were as follows: Lifting full load, 6 ft. per min.; lifting loads up to 10 tons, 25 ft. per min.; racking full load, 20 ft. per min.; racking loads up to 10 tons, 50 ft. per min.; slewing, one revolution in 4 min. Goliaths are constructed to the same lifting capacity as Titans, and some command a lateral range of travel of 100 ft. or more, but the span is generally limited to about 50 or 60 ft. At Dover, a 42-ton steam Goliath had a range of 130 ft., a clear headway of 25 ft., and a total lift of 120 ft. The speeds of movements were: Lifting, 10 ft. per min.: crab travel, 50 ft. per min.; and main travel, 60 ft. per min.

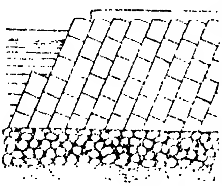
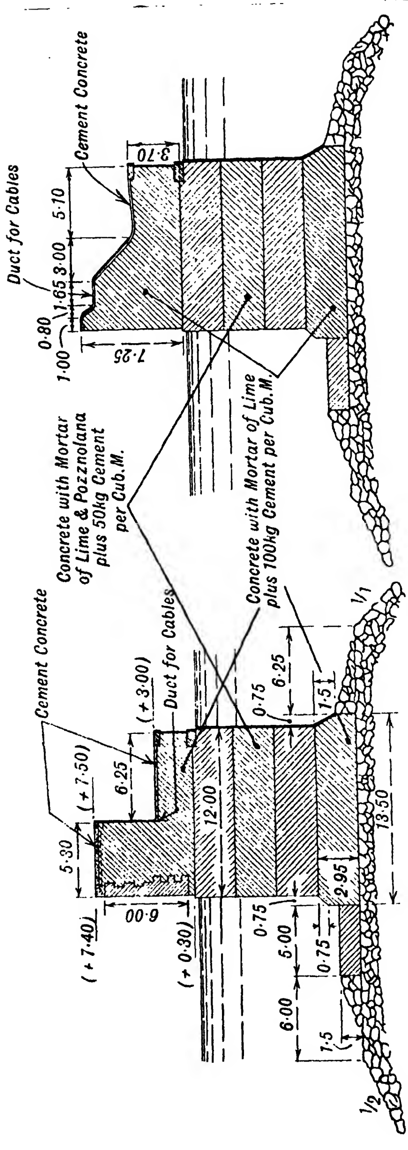


FIG. 6.—Sloping Bond.

Breakwater Construction at Genoa.

For the construction of the extension of the Molo Principe Umberto at Genoa, the type adopted was a vertical wall of concrete blocks superimposed on a rubble mound foundation. A length of 800 metres was built to the section shown in fig. 7. It consists of four courses of monolithic concrete blocks, three of which are 12 metres in length, extending right across the wall, by 2·95 metres high by 4·50 metres wide. The lowermost block is similar, but has a length of 13·50 metres to form an enlarged base for the structure, and its upper edges are bevelled. The weight of the blocks is 350 tons in the bottom course and 300 tons in those above. Lifting and setting were done by a floating pontoon with overhanging sheers. Above the blocks, which bring the structure just above water level, is a crown or capping of mass concrete with masonry facing. In consequence of the revelation of structural weakness in the design at the foot of the parapet wall, where the sharp rectangular off-set tended to fracture under wave impact producing a complex stress of flexure and torsion in the angle, the design for the last 1,050 metres length of the breakwater was changed to the section shown in fig. 8. At the pierhead, the dimensions of the blocks were increased to 13·50 metres by 9·00 metres by 2·95 metres in height. The foundation of wall has been laid at 11·50 metres below mean sea-level and of the mound sub-structure at from 15 to 16 metres below the same datum. The cost of the breakwater complete is given as 35,000 lire per lin. metre which, if the lira be taken at its exchange value at the time, in round figures 60 to the pound sterling, represents roughly, say £590 per lin. yd.


Duct for Cables
Cement Concrete
3·70
5·10
3·00
1·65
0·80
1·00
7·25
Concrete with Mortar of Lime & Pozznolana plus 50kg Cement per Cub.M.
Concrete with Mortar of Lime plus 100kg Cement per Cub.M.
Cement Concrete
(+7·50)
(+3·00)
Duct for Cables
6·25
5·30
6·00
(+7·40)
(+0·30)
12·00
0·75
6·25
1·5
1/1
13·50
2·95
0·75
5·00
0·75
6·00
1·5
1/2

Figs. 7 and 8.

Special Methods of Construction.

Special methods of breakwater construction include the use of bags, or sacks, of concrete, weighing from 50 to 200 tons, for deposition under water by divers, as at La Guayra: large caissons, or tanks, of reinforced concrete, with volumes of 60,000 to 70,000 cu. ft., floated out into position, sunk and filled in solid, as at Bizerta; 146 floating caissons were used in the formation of the invasion harbour at Arromanches, Normandy. The largest had a displacement of 6,544 tons and the smallest 1,672 tons; and blocks set on end, slightly tilted (15° to 20°) out of the vertical so as to form what is called 'sliced work' or *sloping bond*, the object being to allow for unequal settlement in the foundation. Small breakwaters in shallow waters are sometimes constructed of cribwork—box-shaped frames of timber in openwork with cross partitioning and ties, weighted when in position with rubble—but the system is not very durable and only suitable for comparatively sheltered situations.

Among notable breakwaters of recent date, mention should not be omitted of the very remarkable example at Valparaiso, completed in 1930, which is founded at a depth of more than 50 metres below sea level, on a bottom which consists of sand and mud. In order to reduce the cost of bringing up a mound of rubble from so great a depth, as well as to distribute the load over an adequate bearing area, the expedient was adopted of depositing a bank of sand with a base width of about 500 metres and side slopes of 5 to 1, reaching up to a level of 20 metres below the surface. Over the apex and upper part of this mound were deposited quarry waste and rubble to a level of 12 metres below the surface, at which level the wall of the breakwater was founded.

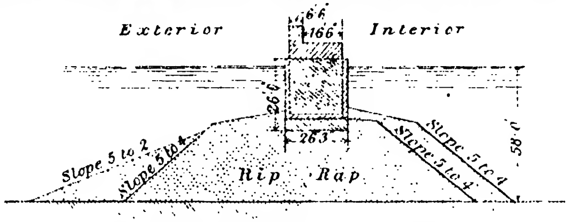


FIG. 9.—Bizerta Breakwater.

DOCKS.

Docks are enclosed repositories for shipping. Where the enclosure is not complete and there is free communication with the sea, the term *Basin*, though not universally employed, is more appropriate. Docks proper are fitted with gates or caissons, either in order to impound the water at a fairly constant level, in which case they are distinguished as *wet docks*, or to exclude the water, in which case they are known as *dry* or *graving docks*. *Floating Docks* are movable structures capable of lifting vessels out of the water and supporting them by their own displacement.

CONSTRUCTIONAL COST OF WET DOCKS.*

(Note: The variety and range of design and equipment render any very effective comparison out of the question.)

	Per acre of Water Surface.		Per acre of Water Surface.
Alexandra Dock, Hull . .	£28,900	Barry Docks	£12,950
Alexandra Dock, Liverpool .	23,300	Prince of Wales Dock, Swansea	18,600
Avonmouth Dock, Bristol .	22,600	Queen's Dock, Glasgow . .	24,450

KING GEORGE V. DOCK, LONDON.

One of the most notable docks constructed in recent years is the King George V. Dock at London, which adds 64 acres to the enclosed water area of the port. The dock has a depth of 38 ft., and the quayage aggregates approximately 10,000 ft. A section of the north, west, and east quay walls is given in fig. 13. An unusual feature on the south side of the dock, shown in fig. 14, is the provision of seven reinforced concrete jetties, each 520 ft. long and 22 ft. wide, which have been constructed at a distance of 32 ft. from the face of the main quay wall, the only shore connection being a timber foot-bridge. The arrangement is due to the fact that a preponderating portion of the goods arriving at the port is transferred to barges for conveyance to their ultimate destination up river. Ships are berthed on the outside of the jetties, and the jetty cranes can discharge goods therefrom either on to the quay for delivery to carts or railway trucks, or for sorting or temporary storage at the sheds behind, or, alternatively, into barges lying in the space between the jetties and the wall.

* See footnote, page 606.

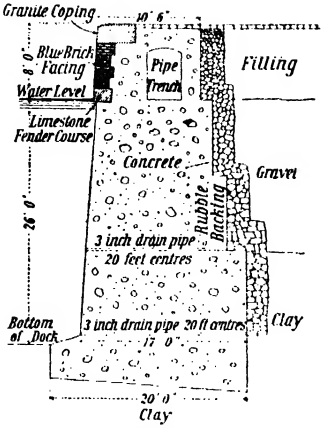


FIG. 10.—Manchester Dock Wall.

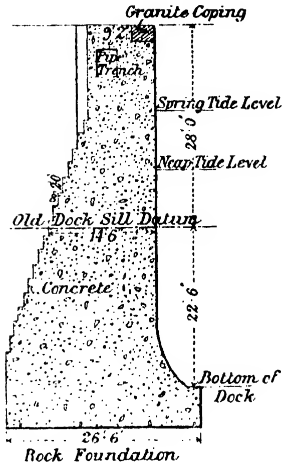


FIG. 11.—Liverpool Dock Wall.

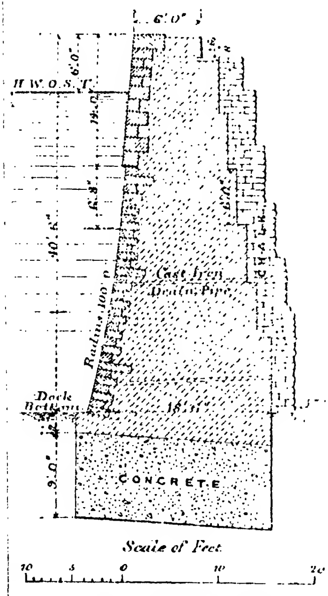


FIG. 12.—Hull Dock Wall.

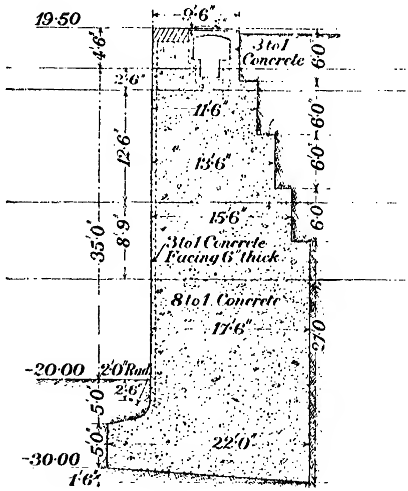


FIG. 13.—King George V. Dock, London: Section of N., E., and W. Quay Walls.

KING GEORGE V. DOCK, LONDON.

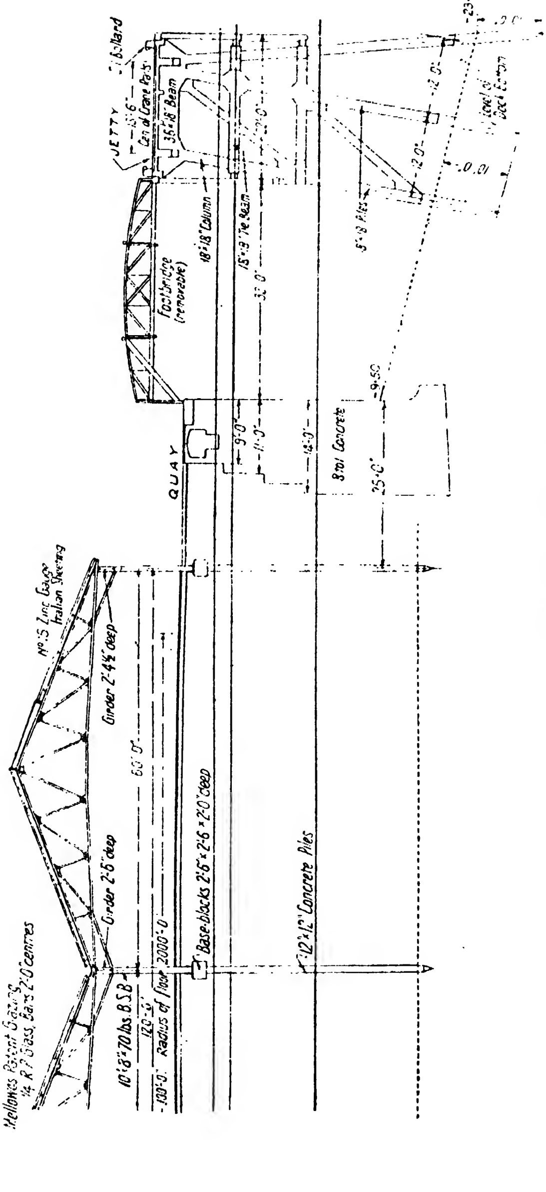


FIG. 14.—Cross-Section of South Quay.

The entrance lock from the river Thames is 800 ft. long by 100 ft. wide, with a depth of water on the sill of 45 ft. below Trinity high water. The chamber is divided into two compartments, of 550 ft. and 250 ft. respectively, by three pairs of steel lock gates, each leaf of which weighs 309 tons. By the use of a floating caisson, for which provision has been made at the inner end of the lock, the available length can be increased to 910 ft. The cost of the dock with its equipment was about 4,130,000*l*.

The Gladstone Docks, Liverpool.

Another notable example of recent dock construction is the Gladstone Dock System of the Mersey Docks and Harbour Board, opened to traffic in 1927. It consists of a vestibule, or turning, basin with two branch docks, the whole comprising a water area of 58½ acres. The vestibule dock is entered from the River Mersey by a lock of outstanding dimensions. It is 1,070 ft. in length and 130 ft. in width, divided into two compartments, with sills 20 ft. below Bay Datum, giving 48 ft. 4 ins. depth of water at H.W.O.S.T., 35 ft. at half tide level, and 22 ft. at L.W.O.S.T. Both the branch docks are 400 ft. wide; one is 1,420 ft., and the other 1,285 ft. long. They are equipped with treble storey sheds of reinforced concrete construction, providing floor areas (including the flat roofs which are designed for light loading) of 32 acres and 23 acres respectively. The system includes a graving dock, one of the largest in Europe. The length of the dock from the head to the inner caisson sill is 1,050 ft. 4 ins. The width of entrance is 120 ft. and there is a depth of water over sill of 43 ft. 6 ins. at H.W.O.S.T. This dock has the peculiarity that it can be used as a wet dock, and for that purpose it is equipped with a single storey shed on its north side.

Eastern Arm Extension of Royal Edward Dock, Avonmouth.

On May 23, 1928, H.R.H. the Prince of Wales opened a new arm or extension, 1,750 ft. long and 400 ft. wide, of the Royal Edward Dock at the port of Bristol which, with a water area of

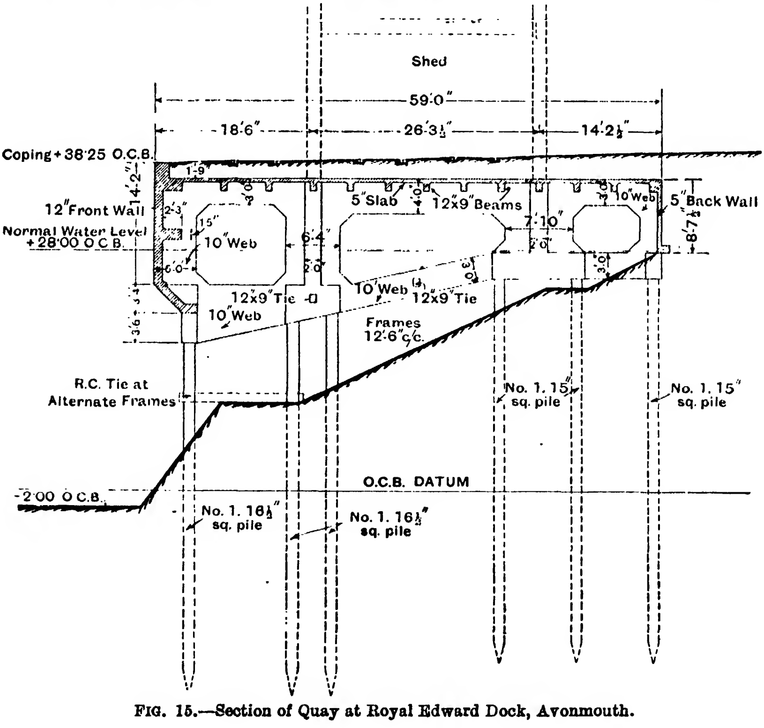


FIG. 15.—Section of Quay at Royal Edward Dock, Avonmouth.

about 17 acres, affords on each of its two main frontages three deep-water berths for large vessels, with an aggregate quayage length of 3,600 ft. The extension has been carried out on lines which, while generally applicable to wharfing, are of some novelty in dock quay construction. In place of a massive wall of the gravity type, there is a system of openwork framing and decking in reinforced concrete, supported on piles of the same material, as shown on the typical section in fig. 15. The number of piles driven amounts to over 4,500, and they are of lengths ranging from 55 to 60 ft. The main frames are set at 12 ft. 6 ins. centres, and the front spacing is made up of concrete slab and beam construction. The cost of the quay was between 70*l.* and 75*l.* per ft. run, covering the reinforced concrete work and the excavation down to the 2½ to 1 slope, within the limits of the front and back wall.

Dock Walls.

Dock Walls.—Under present-day conditions it is necessary for quay walls to possess a perpendicular, or almost perpendicular, face. Ships are now built to a square, box-like section amidships, with vertical sides inclining inwards towards the top. Unless, therefore, there is an absence of batter in the walls, vessels will be unable to lie as closely in contact with the quay coping as is desirable, and quay cranes for loading and discharging them will have to be provided with a greater outreach than is absolutely necessary, at an enhanced cost.

Dock and quay walls formerly, and still, constructed as 'gravity' retaining walls in mass concrete, occasionally faced with brickwork and generally coped with granite or other material resistive of abrasion, are, in many modern instances, designed on more slender lines with the aid of steel reinforcement, so as to dispose of the qualities of both materials to the best advantage in withstanding the thrust of the earth pressure behind. In Germany and Holland a form of quay wall in vogue is constructed from just below water level on a timber platform carried on vertical and raking piles, the wall itself being L-shaped in section, with a base much greater than the height. There are, however, many varieties of design and construction, such as the hollow caisson, cribwork, mattress work, etc., practised under different local conditions. In ground of uncertain nature, the system of well-sinking is adopted for reaching a sound foundation and the wall is built up as a series of 'monolithic' piers, with hollow compartments, which may be filled with sand, or other suitable material.

The *essential features of a 'gravity' quay wall* are weight and resistance to slip. The first is secured by using heavy material and providing ample sectional area; the second by giving a suitable bevel to the surface of the foundation so that it slopes downward from the front of the wall to the back, at the same time taking care that the maximum intensity of pressure does not exceed a certain limit, which, in the case of stiff clay, is 3 tons per sq. ft.; in the case of unconfined sand and gravel is 2 tons; and in the case of soft, uncertain ground, is 10 cwts. per sq. ft.

Stresses in Dock Walls.

Stresses in Dock Walls.—The calculation of the stresses in a dock wall is a very complex problem, involving a considerable number of elements, the exact determination of which is impracticable, so that they have largely to be based on conjecture. Briefly stated, the conditions are as follows: The wall is subjected to the overturning influence of a wedge-shaped mass of earth which represents the excess of material behind the wall lying above the plane of natural slope of the material. The pressure, of course, is intensified by any surcharge upon the surface area of the wedge. The restraining force is the weight of the wall itself, acting vertically downward, assisted to some extent by the earth backing which may rest upon offsets, if any, at the back of the wall.

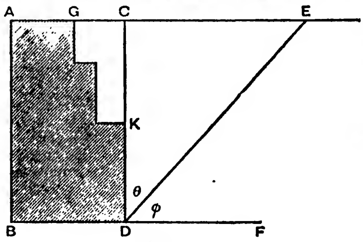


FIG. 16.

In fig. 16, let A B D K G be the outline section of a dock wall.

Let E D be the plane of natural slope of the ground behind the wall, i.e., the slope which the material would assume if left unsupported. The angle E D F will be designated ϕ and is known as the Angle of Repose.

Let the height of the wall A B be h, and w be the weight per unit volume; and let v be the weight per unit volume of the earth behind the wall.

Then the weight of the wedge of earth behind the wall, tending to overturn it, is

$$\frac{vh^2}{2} . \tan \theta .$$

In order to cover cases where the back of the wall is not vertical and where the surface of the ground is not horizontal, this expression may be written in general terms:

$$\frac{vh^2}{2} \times \text{constant.}$$

and the value of the constant, as determined by Professor Rankine in accordance with a theory which neglects any cohesion between the earth particles, is

$$\frac{1 - \sin \phi}{1 + \sin \phi}.$$

The line of action of the overturning force,

$$\frac{vh^2}{2} \cdot \frac{1 - \sin \phi}{1 + \sin \phi},$$

is taken as parallel to the surface of the ground and the point of application at one-third the height of the wall measured from the base.

The above formula, though strictly applicable only to walls with a backing of sand or granular material, in engineering practice has been applied very generally, with results that, as a rule, are 'on the safe side,' though they may in some cases be excessively so for strongly coherent earth backing, such as clay. Résal has shown (*Poussée des Terres*) that the cohesion of the material may increase the value of the angle of friction very considerably beyond the limits usually assigned to ϕ, though with pronounced variations, due to the degree of humidity of the soil. It is well known, of course, that banks of argillaceous earth will, under favourable circumstances, stand almost vertically without support for an appreciable height.

A recent paper by Prof. Frewen Jenkin on 'The Pressure on Retaining Walls' (*Min. Proc. Inst. C.E.*, vol. 234), descriptive of experiments made by him with models, gives what the author describes as a Revised Wedge Theory for sand, which is a departure from Rankine's theory. It cannot, however, be explained with sufficient brevity for notice here.

For simple practical calculations for dock walls where the quay surface is almost invariably

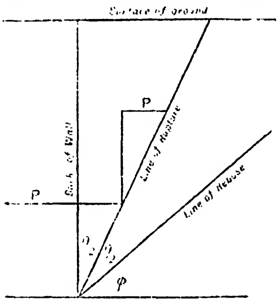


FIG. 17

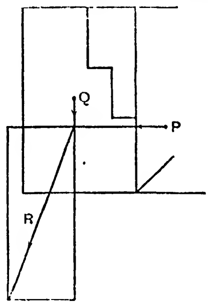


FIG. 18.

horizontal, the angle θ ($=90° - \phi$) is bisected and the value $\frac{\theta}{2}$, fig. 17, taken as representing the angle of the wedge producing maximum pressure, the other moiety being ineffective as regards overturning owing to the support it receives from the earth beneath it. The weight of the triangle $\frac{h^2}{2} \tan \frac{\theta}{2}$ is computed, and the horizontal component, P, taken of the force acting down the plane of rupture, of which $v\frac{h^2}{2} \tan \frac{\theta}{2}$ is the vertical component. This constitutes the overturning force, which may accordingly be written:

$$P = \frac{vh^2}{2} \tan^2 \frac{\theta}{2}$$

The point of application of P is again at one-third of the height of the wall.

The restraining force Q is measured by the sectional area of the wall A B D G, fig. 16, multiplied by w, together with the sectional area of the earth G K O resting on the wall, multiplied by v, the two acting vertically downwards through their common centre of gravity.

Calling the overturning force P, and the restraining force Q, fig. 18, and completing the parallelogram of forces, the resultant R will be obtained, representing in magnitude and direction the force acting at the base of the wall.

So long as the line of action of R falls within the base of the wall, there is theoretical stability, but, for practical constructional purposes, it is essential:

(1) That the line of action of R should not fall outside the middle third of the base line, and

(2) That the magnitude of R should not be sufficient to crush the base; in which connection it is to be noted that the intensity of pressure increases with the eccentricity of the resultant (i.e. its distance from the centre) and that when R passes through the limiting point of the middle third of the base, the intensity of pressure on the outer edge is just twice the mean intensity

As a rough practical rule in designing dock walls, the mean width or thickness may be made about one-third of the height. It should never be less than one-fourth. The base width may be about one-half the height, but need never be more than two-thirds.

EXAMPLES OF PROPORTIONS ADOPTED FOR DOCK WALLS.

Port.	Total Height. h	Base Width. b	Sectional Area. A	Mean Width $= \frac{A}{h}$	Ratios. Base. Height	Mean Width Height.
	Ft.	Ft.	Sq. Ft.	Ft.		
Manchester	38	20	476	12·5	·53	·33
Middlesbrough	48	21	648	13·5	·44	·28
Immingham	51	31	910	17·8	·60	·34
Hull	52	20	736	14·1	·38	·27
Liverpool	55	22	760	13·9	·40	·25
Southampton	71½	39	1,692	23·6	·54	·33

CONSTRUCTIONAL COST OF QUAY WALLS.*

	Height of Wall.	Cost per lin. yard.
	Ft.	£
Albert Dock, Hull	49	58
Albert Dock, London	37½	70
Alexandra Dock, Hull	52	84
Antwerp River Quays	56¾	250
Dublin River Quay	42	120
Princes Dock, Glasgow	57¾	120
White Star Dock, Southampton	71½	195

Entrance Locks.

Entrance Locks.—During periods when the water in a dock is impounded at a higher level than that of the external water (in some cases continuously), the dock can only be entered through the medium of a lock. The size of lock chambers is governed by the dimensions of the ships making use of them, and these vary at different ports. A large vessel was the 'Normandie,' with a length over all of 1,027 ft.,† a breadth of 118 ft., and a loaded draught of 36·6 ft. The length of 1,000 ft. is, perhaps, somewhat extreme, but there are a number of vessels from 900 ft. to 950 ft. long, by 95 ft. to 100 ft. broad, and they draw from 35 ft. to 40 ft. of water when loaded to full capacity. No port can claim to be of the first rank which does not receive vessels from 500 to 600 ft. long, with 60 to 70 ft. beam and drawing 30 ft. of water.

Some examples of modern dock entrance locks are as follows:—

Name.	Port.	Length of Chamber.	Width of Entrance.	Depth of Water over Sill at H.W.
		Ft.	Ft.	Ft.
Gladstone (1927)	Liverpool	1,070	130	48
Tilbury (1929)	London	1,000	110	45½
Kruisschans (1928)	Antwerp	886	115	47
North (1931)	Bremerhaven	1220	147½	44½

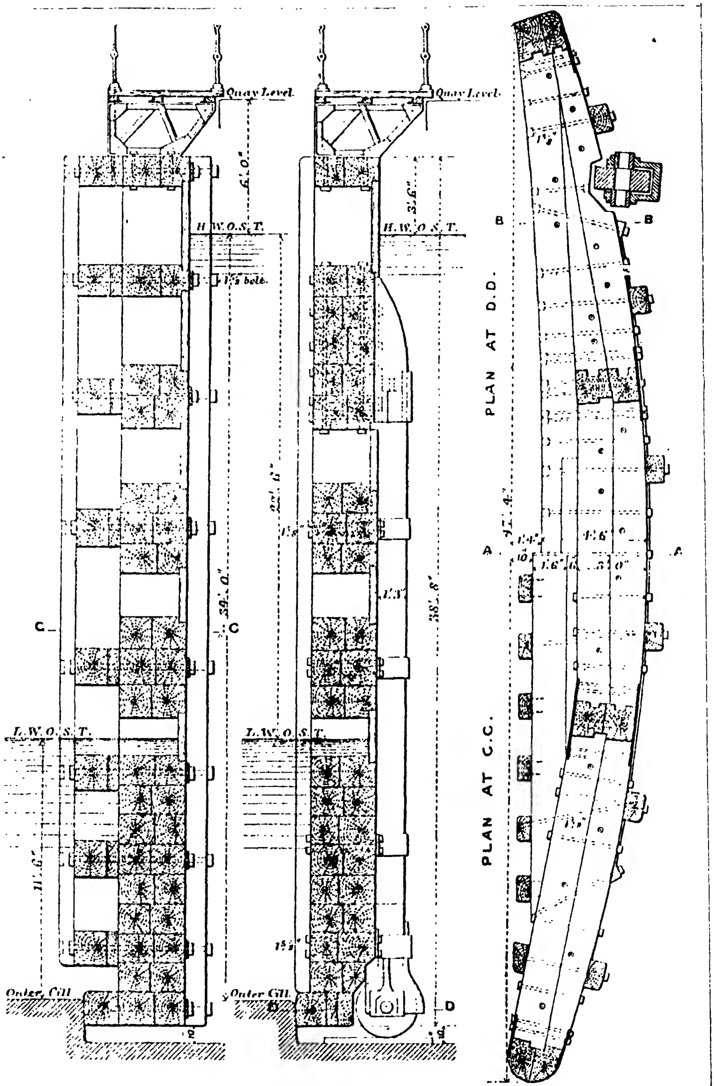
See also some prominent examples of Ship Canal Locks in Section XVI., page 599.

Dock Gates.

Dock Gates.—Where the rise and fall of the tide does not exceed about 12 ft. the inducement to provide completely enclosed areas for shipping is not great, but above that limit the necessity for impounding the dock water becomes evident on account of the inconvenience arising from rapid changes of level, frequent alteration in moorings, and chafing against the quayside.

Dock gates are either of steel or wood—in the latter case of greenheart, jarrah, or other hard, heavy, durable timber. Greenheart gates cost more than steel gates—in some cases nearly 50 per cent. more—but their natural durability is very great and the cost of maintenance practically nil. Unfortunately, they are subject to the attacks of sea organisms, and in certain ports considerable trouble has been experienced from this cause. Salt water, especially if contaminated in any way with sewage, is extremely deleterious to ironwork, and the life of a steel gate, under these circumstances, can scarcely be expected to exceed 30 years. Greenheart gates have been found in excellent condition at the end of 50 years.

* See footnote, page 606. † The length of the 'Queen Mary' is slightly less, 1,018 ft.



SECTION ON A.A. SECTION ON B.B. PLAN.

Scale of Feet

10 5 0 10 Feet

The generality of dock gates swing vertically and, consisting of two leaves which meet at an angle in the centre of the waterway, are recessed within the side walls when the passage is clear for shipping. A rather different type of gate is the 'Box' gate, which is specially adapted to conditions of site in which side recesses for ordinary double leaf gates would be inconvenient. It is a single leaf gate with a hinge or keel running horizontally across the sill, and it is lifted or lowered by means of wire ropes. When the passage is open, the gate lies in a prone position below sill level. A recent example at the entrance to a graving dock, 85 feet wide, on the river Clyde, is described in *The Engineer* of February 17, 1933. Another example at the entrance of the Dover Train Ferry Dock is described in *Min. Proc. Inst. C.E. Journal*, December 1937.

Stresses in Dock Gates.

Stresses in Dock Gates.—Dock (double-leaf) gates are maintained in equilibrium under the action of four external forces:—(1) the resultant pressure of the water against the back of the gate $= \frac{wh^2}{2}$ per lineal foot of gate, where w ($62\frac{1}{2}$ lbs. for fresh, 64 lbs. for salt water) is the weight of a cubic foot of water, and h its height behind the gate. This force acts horizontally at one-third of the height. When there is water on both faces $p = \frac{w}{2}(h^2 - h_1^2)$, and the line of action is determined by the centre of gravity of a quadrilateral representing the difference in pressure. (2) The reaction of the mitre or meeting posts of the gates acting perpendicularly to the meeting

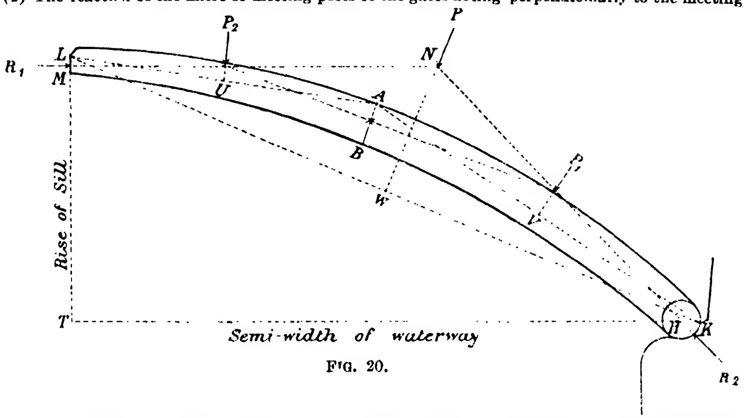


FIG. 20.

faces, and therefore also perpendicular to the centre line of the waterway. (3) The reaction of the hollow quoins passing through the heel-post of the gate and also through the point of intersection of the other two forces. (4) The reaction of the sill, which is generally neglected.

Fig. 20 represents the plan of one leaf of a pair of gates. P is the total water pressure upon the back of the leaf, assumed concentrated at its centre. R_1 is the mitre-post reaction of the adjoining leaf, taken as passing through the centre line of the abutting surfaces. R_2 is the reaction of the hollow quoin, assumed to act through the axis of the heelpost. These three forces being in equilibrium, the triangle of forces *ors* (fig. 21) can be drawn, having its sides parallel to the forces P, R_1, and R_2 respectively, and since the magnitude of P is known, the magnitudes of the other two are determined.

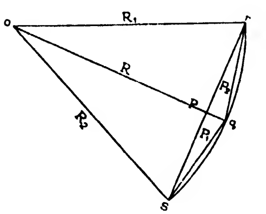


FIG. 21.

If now it be required to find the position and amount of the resultant stress across any section AB (fig. 20) of the gate, the procedure is as follows:—Join the point A to each of the extremities K, L of the water-bearing surface of the leaf; bisect these lines at U and V respectively, and draw perpendiculars to represent the total water pressure on each section. Each section is in equilibrium under the action of three forces: in one case, the water pressure, the heelpost reaction, and the stress across AB; in the second case, the water pressure, the mitre-post reaction, and also the stress acting across AB in the opposite direction. Join the points of intersection of the known forces to obtain the line of action of the resultant pressure at the section AB. Its magnitude may be determined by drawing a parallel line, qo, in the force diagram from the point o (fig. 21). By taking a series of sections in this way, it will be found that the locus of q is sensibly the arc of a circle, and therefore that, except in the case of very flat gates, the resultant pressure is so nearly constant as to be justifiably considered so without serious error.

Rise of Sill.—The 'rise of sill,' MT (fig. 20), varies in different cases from ¼ to ⅕ of the 'span of the gates' or width of waterway. The vicinity of the latter limit is the more common, especially for large cylindrical gates, where a large rise would entail a correspondingly long and deep recess in the side walls.

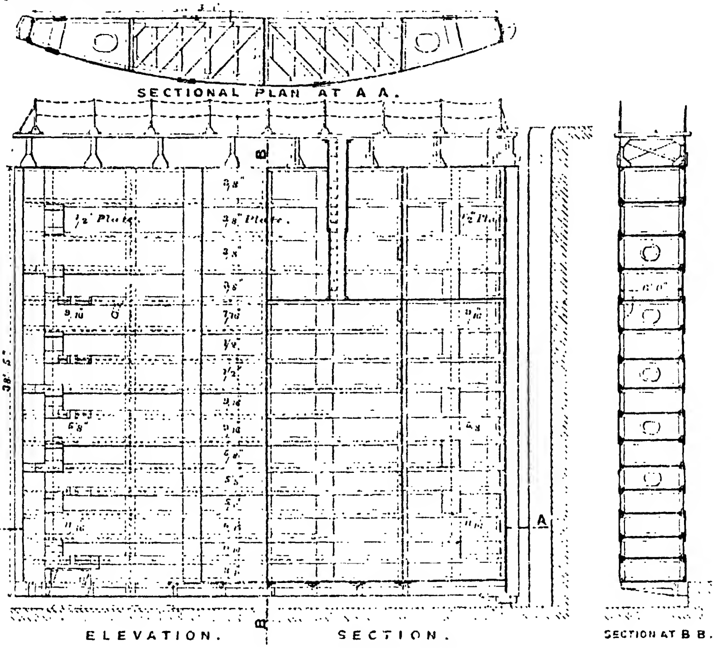


FIG. 22.—Dock Gates. (Steel.)

PARTICULARS OF DOCK GATES.

Port.	Material.	Width of Water-way.	Depth of Water at H.W.O.S.T.	Height of Gates.	Weight of one Leaf.	Cost per sq. ft. of Gate.*	Remarks.
		Ft.	Ft.	Ft.	Tons.	s. d.	
Liverpool .	Greenheart	90	39½	44½	165	45 3	
Do.	Steel	130	54	56	496	—	
London .	Steel	80	36¼	38½	185	—	
Do.	Steel	100	45	50	350	—	
Hull . .	Greenheart	85	34	38½	176	48 0	
Manchester	Steel	65	40	—	—	33 9	Similar Greenheart gates cost 49*s.* 9*d.* per square foot.
Limerick .	Steel	70	23½	26	45	25 3	
Avon-mouth	Greenheart, pitch pine and memel	70	38½	48	102	26 0	

* See footnote, page 606.

CAISSONS

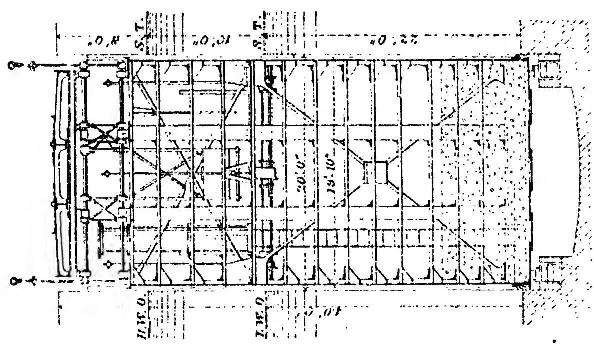


Fig. 23.—Section of Traversing Caisson.

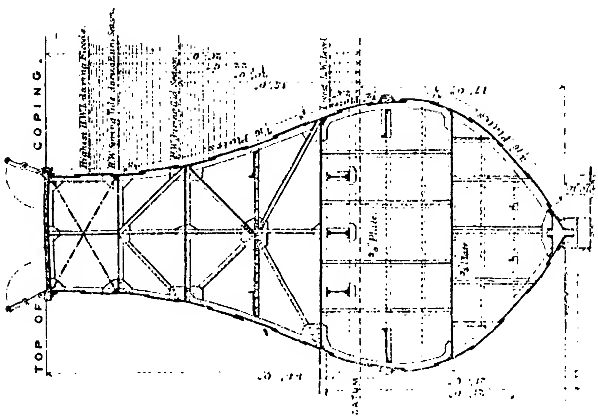


Fig. 24.—Section of Ship Caisson.

Caissons.

Caissons, which are hollow, framed structures of steel, with a platform at or about quay level, are sometimes preferred to gates for closing dock entrances, on the ground that they do not absorb the length of side walls which is required to house gates when a passage is open. Furthermore, although the first cost of a caisson is undoubtedly in most cases greater than that of a pair of gates, yet if the cost of a swing bridge for vehicular traffic, which is a necessary adjunct in the case of gates, be also taken into consideration, the advantage will be found to lie with the caisson.

Caissons are mainly of two types: traversing and floating. Traversing caissons are rectangular boxes with a rectilinear motion either sliding or rolling. When withdrawn from the passage way, they are housed in a special chamber constructed in the side wall. Floating caissons are usually ship-shaped, having more or less the form of a navigable vessel, though the curvature of the sides varies very much with the depth of water in which they have to float. Floating caissons are not easy to manage in boisterous weather and strong currents.

PARTICULARS OF CAISSONS.

Port.	Type.	Width of Waterway.	Depth of Water on sill at H.W.O.S.T.	Height of Caisson.	Cost per sq. ft. of Waterway.*
		Ft.	Ft.	Ft.	s.
Quebec	Rolling	62	25½	32	72
Malta	Sliding	94	36½	40½	85·7
Belfast	Ship	80	31	33	35
Barry	Floating	60¼	26¾	31½	47·2
Ymuiden	Rolling	164	49	65½	60·3

QUAYSIDE APPLIANCES.

Quay Cranes.

For general work, quay cranes range in lifting capacity from 1 to 3 tons. For hydraulic cranes the usual capacity is 30 to 40 cwts. It very often happens that the parcels from a ship's hold do not exceed 12 to 15 cwts., and, under these circumstances, a hydraulic crane of single and much higher power than this cannot be worked to advantage, as the consumption of pressure water is the same whether the load be infinitesimal or the maximum. Electric cranes are better adapted to variable loads, since the current consumed is directly proportional to the work done. On the other hand, the initial cost of electric cranes is some 30 to 40 per cent. more than that of hydraulic cranes, and the capital charges (interest and depreciation) are factors of considerable importance in estimating the relative economy. Electric cranes are now commonly installed at leading ports with 3 tons maximum capacity.

Crane Types.—Quay cranes are generally movable and include several types.

(*a*) The ordinary crane, with a wheel gauge of 10 to 15 ft. running at quay level.

(*b*) The portal crane, with an arched base spanning one or more lines of railway sidings and admitting of the passage of rolling stock beneath the crane.

(*c*) The semi-portal crane, like an inverted L, with one leg or support running on a rail at quay level, and the back support carried by a rail at a higher level on an adjacent shed structure, the object being the same as in the case of the portal crane.

(*d*) The roof crane, carried entirely on the roof of the shed or warehouse.

Types (*a*) and (*d*) are adapted to cases where there are no sidings, and type (*d*) where the shed is situated close to the edge of the quay.

The *cycle of operations* of a quay crane consists of lifting from a hold, slewing, depositing, and returning unloaded, and should, in ordinary cases, where the range of lift does not exceed 50 or 60 ft., be performed in about one minute. A good average under favourable conditions would be 50 cycles per hour; but spread over any lengthy period, this rate could hardly be maintained, for reasons entirely unconnected with the crane.

Speeds of Quay Cranes.—The ordinary speeds of 30 to 35 cwts. quay crane machinery are as follows:—

Lifting, full load	250 ft. to 300 ft. per min.
Lifting, light	350 ft. to 400 ft. per min.
Lowering, full load	400 ft. per min.
Lowering, light	150 ft. per min.
Slewing	one revolution in 25 or 30 seconds.

Luffing Cranes.—It is an important consideration in modern quayside work that cranes should have considerable outreach. Not only is it necessary for them to plumb the hatchways of vessels moored alongside, but in many cases it is desirable to command a barge or lighter beyond. This involves an outreach of 50 to 60 ft. from the face of the quay for even vessels of moderate beam, and entails a luffing or derricking movement of the jib. In view of the obstruction to radial movement caused by ships' yards and stays, the luffing of the crane jib has of recent years become

* See footnote, page 606.

an important feature of crane design. Formerly, the load was unavoidably raised during the inward travel of the jib with a correspondingly unnecessary expenditure of energy and time. This has now been obviated by the introduction of compensating luffing gear by means of which the load is carried at a uniform level throughout its range of travel.

Compound Cranes.—Quay cranes have generally a radial action, and though the luffing gear provides a rectilinear motion, the movement of goods is mainly along a more or less circular path. The joint effect is to afford considerable scope for the deposit of goods on the quay in contradistinction to cranes of the transporter type, which can only function along a straight line and often only deposit at a single point. Such cranes, however, have an advantage in being able in certain cases to transmit their loads within a transit shed and so under cover. At Hamburg, a combined type of quay crane has been introduced with the usual radial jib on a rotating platform above, while below, within the framework of the pedestal, there are one or two crabs travelling on horizontal tracks at right angles to the quay front. The aggregate power of these cranes is considerable, reaching to as much as 6, and even 9, tons.

Fixed Quay Cranes.—Fixed quay cranes are provided for dealing with special loads of considerable calibre. The powers provided are generally 5, 7, 10, 12, 15, and 25 tons, with appliances at first-class ports ranging up to 150 or 200 tons. The rates of lifting are correspondingly slower; about 150 ft. per min. for 5-ton cranes, 30 ft. per min. for 25-ton cranes, and 20 ft. per min. for 50-ton cranes.

Burtoning.

General cargo is mainly, and in fact almost universally, handled at North American ports by a system known as Burtoning, which is the same in principle as the Union Purchase used at English ports in conjunction with ships' gear. At American ports, the sheds are equipped with cargo masts, generally prolongations of the shed front columns, rising to a height of 80 ft. or so above the quay level and supporting a longitudinal girder, or rail, running the full length of the shed. To this rail, at any selected point, a pulley may be attached, worked with a line, one extremity of which is joined, with a similar line from a pulley at the extremity of a ship's boom, to the hook carrying the load. Two winches are required for operating: one for each line. The ship's winch and line serve to lift the load from the ship's hold to deck level; the shed winch then takes charge, and by means of the shed line draws the load overside to the quay, where it is lowered to the ground.

Floating Cranes.

Floating Cranes.—For the heaviest lifts, floating cranes are often preferred. It is a convenience to bring the crane to the ship rather than *vice versâ*. Against this must be set the greater initial and working cost of a floating crane. Floating cranes are provided at most first-class ports of lifting power ranging from 25 to 100 or 150 tons. A floating crane recently built for a Japanese port has a working load of 200 tons at 105 ft. radius, 100 tons at 150 ft. radius, and 30 tons at 160 ft. radius. The heaviest lifts are performed at the rate of 5 ft. per minute. A revolution takes 6 to 8 minutes.

A floating crane of 200 tons capacity is among the plant of the Mersey Docks and Harbour Board. It is an electrically-driven, derricking, and revolving crane, 154 ft. in length, 88 ft. in breadth, and 16 ft. in depth. The hoisting speeds vary from 6 ft. per min. for loads of 200 tons, to 20 ft. per min. for loads of 60 tons. The highest point of the jib at its highest position is 246 ft. above the water-line, with a maximum radius for loads of 60 tons amounting to 183 ft., and for loads of 200 tons, to 110 ft. The vessel is named the *Mammoth*; in conjunction with its outreach it is claimed to possess the largest lifting capacity of any existing floating crane. There are, however, two floating cranes of 250 tons capacity at the Panama Canal.

Even for relatively small power lifts (5 tons or less) floating cranes are employed with advantage at some French ports where the conditions are favourable for their use. The length of the crane pontoons are about 60 ft., a convenient dimension from the point of view of manœuvring; the width increases (to prevent transverse oscillation) from 17 to 31 ft. in proportion as the power of the cranes increases from 1½ to 5 tons and the outreach from 25 to 50 ft. The height of the jib pulley varies from 53 to 65 ft. above water level.

(See also CRANES, Section XXXVIII, Part I.)

Transporters.

Transporters.—Where loads are uniform, can be readily handled, and have to be transmitted some distance, a transporter offers considerable facilities for dealing with them at high speeds, ranging up to 1,200 or 1,500 ft. per minute. Quayside transporters have generally to be furnished with hinged cantilever ends capable of being raised when out of action. The travelling motion is not necessarily rectilinear, curved tracks being employed where necessary.

Coaling Appliances.

Shipment of Coal.—Coal is shipped in a variety of ways. The simplest method is that practised on the north-east coast, where the altitude of the banks of the rivers Tyne and Wear enables discharge to be made direct from wagon into steamer by means of 'staiths,' which are wharves or jetties provided with shoots inclined downwards towards the holds. Apart from this, the appliances used at low-level quays comprise the hoist and tip and the coaling crane, as also the belt conveyor travelling up an incline, which is perhaps most developed in the United States.

The belt conveyor requires a layout on the quay, which is not always practicable or convenient in congested areas such as prevail at English docks. The hoist raises the wagon to the requisite height and tips it at an angle of 45° or 50°, so that the contents pass into a shoot. The crane lifts the wagon bodily, slings it over the ship's hatchway and there empties it. The protection of coal from breakage is of such importance that in nearly all cases where coal is tipped from a considerable height, anti-breakage appliances are provided to deposit the first few loads at the bottom of the hold.

Coal wagons in this country are commonly built to contain 10, 12, 15 or 20 tons of coal, and the gross weight ranges from 15 to 30 tons. To this must be added from 6 to 9 tons for the cradle or platform. Consequently the lifting capacity of a coaling appliance will range between 20 and 40 tons according to the class of wagon dealt with. In certain special cases, as at Goole Docks, provision is made to lift a gross load of 50 or 60 tons, of which up to 40 tons is coal. The coal is brought along the Aire and Calder Canal in tanks, which are floated under the hoists.

In America the size of coaling wagons is very much greater and varies from 40 tons upwards to 100 or 120 tons. At British ports attempts are now being made to secure the general adoption of larger-sized wagons (20 tons capacity) than has hitherto been the practice.

The rate of coaling by an ordinary crane or hoist is from 300 to 450 tons and may even reach 600 tons per hour, but it is largely governed by the rate of trimming operations in the hold. The lifting and lowering of the wagons is done at speeds of between 100 and 180 ft. per minute and slewing at about one revolution per minute.

The belt conveyor consists either of a continuous rubber belt or of articulated steel plates. Of the two, the rubber belt has shown itself to be more convenient and more durable than the steel plating, the wear on which is considerable. Rubber belts are run at rates up to 600 or 700 ft. per minute, and they discharge the coal in a continuous stream. The belt is fed by means of a hopper recessed in the quay, into which the coal wagons are emptied. Installations in this country are of moderate proportions, the width of the belt running up to about 48 ins., but in the United States very much larger installations have been effected with belts of 60 ins. diameter. These belts travel up inclined ways to the summit of movable towers, whence the coal is passed on to a transverse belt discharging into the ship. Considerable expedition has been gained by the use of the mechanical trimmer, which is a short, rapidly rotating belt at the foot of the vertical shoot leading into the hold. The speed of the trimmer belt ranges up to 2,000 ft. per minute, and the acquired velocity projects the coal forward into all parts of the hold to distances up to 40 or 50 ft.

Ship's Bunkering.—The foregoing appliances are in vogue in connection with shipping coal cargoes. For the more moderate requirements of ships' bunkers, the loading appliance has generally to be brought to the ship's side, so as to proceed simultaneously with cargo discharging or loading operations. A good deal of bunkering is done by hand, with such primitive assistance as is afforded by baskets, ropes, pulley blocks, and winches, but the rate of coaling under such conditions is very slow and relatively costly—perhaps 2 tons per man per hour. There are a number of special floating machines provided at important ports for doing the work more expeditiously. A typical example, and one of the most modern designs, is the Sulsted Elevator, built for the company of that name by Spencers (Melksham) Limited, and at work in the port of London. It is capable of handling 300 tons of coal per hour, but such a rate is manifestly only practicable under favourable circumstances; it could not be maintained with trimming operations going on.

Coal Discharge.—For the importation of coal, as also of ore, the grab is perhaps the appliance most commonly in use in this country, as also on a much more gigantic scale at the Lake Ports of North America. As used at English ports, the capacity of the grab runs up to 6 tons, with a total combined load, including the grab itself, of about 12 tons. At the Lake Ports of North America the standard type is a grab capable of taking a load of 12 tons, and the largest can take 17 tons. There is a 'bucket' for clearing up the holds at the close of operations, which when outspread has a span of 24 ft. between the bucket lips.

Grain Elevators.

Grain Elevators.—Grain is handled either by the bucket elevator and conveyor, or by the pneumatic elevator or suction apparatus. The latter has the advantage that the pipes, being flexible, can easily be taken to the furthermost parts of a ship's interior, and that no restrictions are imposed by variations in freeboard and depth of hold. The bucket elevator, though cheaper to construct and requiring less power, is hampered in working by the necessity of ploughing and trimming a preponderant portion of the grain—often as much as 80 per cent.—to the elevator leg. A special gang of competent men is required for the purpose, and this, of course, increases the cost of handling.

It is impracticable to do trimming with a ship's winch at a higher average rate than about 80 tons per hour. Beyond this limit the pneumatic machine becomes more effective, and pneumatic elevators can be constructed so as to give outputs of between two and three hundred tons per hour. In practice something like 200 tons per hour is quite a normal rate.

Pneumatic plants are now commonly in use in warehouses and mills for handling granular material. One of the great advantages is freedom from dust which is a desideratum in handling foodstuffs and other material which must be kept clean.

An objection urged against the suction apparatus is that the surface friction of the pipes tends to injure some of the more delicate classes of grain, such as malting barley, but this drawback, for all practical purposes, has been eliminated of late years by the easing of bends and the removal of all internal projections in the pipes. In any case it should not be overlooked that the cutting and driving action of the buckets of a bucket elevator cannot be without some similar attritional effect.

Bucket elevators are either housed at the quayside or are mounted on barges, in both cases being fitted with derrick supports and suspension gear to enable them to be projected forward and set into the ship's hatchway. Elevator speeds generally range between 200 and 400 ft. per minute, but may be as high as 600 ft. per minute. Band or belt conveyors run at 300 to 600 ft. per minute, but for light grain, such as oats, the speed should not exceed 500 ft. per minute.

The development of the pneumatic operation of handling grain has been very marked in recent years. The slow-speed horizontal reciprocating steam-driven suction pumps have been largely superseded by quick-acting vertical pumps driven by internal combustion engines which are coupled direct to the pumps. A further development is the introduction of a multi-stage air exhauster, also driven in many cases by an internal combustion engine through speeding-up gear. This type is perhaps coming mostly into favour, especially where electric power is available.

Timber Appliances.

Timber, as received at ports, is generally divided into the categories of hard wood and soft wood, the former comprising logs of greenheart, teak, oak, etc., and the latter consisting chiefly of deals, battens, and boards of pine and spruce. Logs are handled overside by crane and either floated to timber ponds or conveyed on trollies to storage sites. In Liverpool a special arrangement of tandem axles is employed to which the logs are underslung.

Cargoes of deals may consist of from 150,000 to 300,000 pieces of various sizes and marks. This renders the use of apparatus of the conveyor type difficult and unsuitable, although such appliances have been successfully installed in certain cases. Generally speaking, cargoes of this class are best dealt with by steam cranes of moderate capacity (30 cwts. to 2 tons) which, arranged to run at right angles to the quay front, convey their loads to an adjacent storage ground at speeds up to 600 ft. per minute. The lifting speed is about 160 ft. per minute, and the slewing speed about 1½ to 2 revolutions per minute.

There is an appliance in use at American ports as well as at sawmills and lumber yards which has proved very serviceable for handling timber. It is known as the Ross carrier, and is a framed motor-driven carriage, standing about 9 or 10 ft. high, the driver's seat being at the top. The framework is of arched or pierced pedestal type, so that the carriage can run astraddle of the load to be lifted, which is placed on 'bolsters,' or cross-bearers, on the ground. Suspended from the carriage chassis are two adjustable side frames with angle iron clips which can be lowered, brought under the bolsters, and then raised to a height of a foot or so to give the necessary clearance for travelling. The carrier transports its load at a speed of 12 to 15 miles per hour. The load to be carried may be 3 ft. wide, 4 ft. high, and of any ordinary length of deals; the carrier frames being about 12 ft. long, the deals project back and front. The carrier can also be used for logs and timber in baulk within the same limits of width and height. The carrying capacity is 10,000 lbs. The appliance is used in connection with a sorting table, known as the Ross break-down rig or drop table, which receives the parcel of deals from the carrier and drops, thereby releasing the deals from the bolster and placing them on a series of chains which serve to break down the load, so that it can be spread over the table and sorted to grade and size.

GRAVING AND FLOATING DOCKS.

A modern graving dock is an excavated chamber with a floor, two sides and an end wall, all of these being usually constructed of concrete, which may, in certain cases, be faced with brickwork, granite or other material. A few of the older graving docks are to be found with wooden floors over a concrete or masonry base. The remaining end of the chamber is the entrance, and is closed, as occasion requires, either by a single gate, or a pair of gates, or again by a caisson, which may be either of the sliding, rolling or floating type. When a single gate is used, it is of the 'Box' type, hinged at the sill and falling flat into a recess below the sill, when open (see page 619). Double gates are pivoted, or hinged, into the side quoins of the entrance, and move in a vertical position through the arc of a circle.

Suitable culverts are provided for emptying and filling the dock. After the entry of a ship, the entrance is closed and the water is pumped from within the dock, though, in certain cases, where graving docks are entered direct from a river or open waterway where tidal conditions prevail, the operation may be assisted, and sometimes be altogether effected, by the fall of the tide.

The floating dock is an open-ended, hollow structure of steel which, in order to receive a vessel, is sunk to the requisite depth, by allowing its interior chambers to fill with water. When the vessel has been berthed, this water is pumped out, the dock rises bodily and so lifts the vessel above the water-level.

It is not altogether easy to make an effective comparison of the respective merits of graving and floating docks, on account of the wide divergence in the conditions to which each structure is specially applicable, and, although there are strong and convinced advocates for the general superiority of one or other of the two systems, a brief summary of the most salient considerations will show that it is scarcely justifiable to decide conclusively in favour of either. Floating docks are very widely adopted in Germany and Holland and at various other Continental ports: in Great Britain there are more graving docks than floating docks.

A graving dock requires a good foundation at a reasonable depth and a disposition of site which frequently involves the appropriation of a considerable area of valuable land.

A floating dock requires a sheltered position with an adequate depth of water which, if not obtainable naturally, will have to be gained by dredging. The depth of water to be provided for a floating dock must be sufficient to cover the draught of the ship, the depth of the pontoons and the requisite clearances of a foot or so between keel and blocks and between pontoon and bed.

As a typical example, the Wellington Harbour (N.Z.) floating dock, of 17,000 tons lifting power, has a draught over keel blocks of 26 ft., and requires a depth of water of 46 ft in which to work when a ship of the maximum draught has to be lifted. Where there is tidal variation, if ships of the maximum draught are to be docked at low water, the site may require some dredging to give the necessary depth of water under these conditions, but when the range of tide is very considerable it is not always considered necessary to make provision for docking ships of maximum draught at lowest tide level.

In sites where the foundation is defective, the depth of excavation required for the floor of a graving dock may easily equal the depth of the floating dock pontoons, and the difficulties and expense in these cases are just as pronounced.

The utility of a graving dock is measured by the linear dimensions of the largest vessel which it can accommodate. The weight of the vessel is relatively a matter of little or no importance.

On the other hand, the utility of a floating dock is gauged by its maximum lifting capacity and is independent of the exact size of the vessel. Floating docks can, and often do, take vessels several feet in excess of their own length and, in the case of one-sided docks, in excess of their own width.

The size of the entrance of a graving dock is influenced by the cost of the gates, or caisson, to be used for closing it, and by the greatest width that can safely withstand the pressure of water. In floating docks these considerations do not enter into account.

Broadly speaking, apart from the physical conditions of the site, it may be said that for ships needing long and heavy repairs, the graving dock has certain advantages, while for the execution of light repairs, painting and hull inspection, such as form the major part of the work done in a dry dock, the floating dock affords quicker and more convenient service. Both classes of work, however, can be carried out in either type of dock with equal success.

Relative Costs of Graving and Floating Docks.*

Initial Cost.—The initial cost of a number of concrete and stone graving docks, including pumping and general equipment, measured in terms of their serviceable dimensions (*i.e.* internal length by breadth of entrance by depth of water over blocks or sill), has varied in the past from 1*s.* 3*d.* to 3*s.* 6*d.* per cubic foot.

The cost of floating docks is usually measured in terms of their lifting power, and varies inversely as the size of the dock; it has ranged from about 14*l.* per ton for a dock which would take small coasting ships to about 6*l.* per ton for one to take large Atlantic liners.

As regards time of construction, a graving dock of any importance could hardly be completed within less than eighteen months or even a couple of years, whereas a floating dock could be ready in half the time. The floating dock could, moreover, be constructed in any convenient locality and transported to its destination at some additional expense for towage and insurance. Many floating docks for the colonies are constructed in home shipyards.

Durability and Upkeep.—A stone or concrete graving dock is practically indestructible, and the only charge brought against it, in this respect, is that of outlasting its utility and becoming obsolete. This is, no doubt, true in regard to the class of vessel for which it is constructed; but, on the other hand, when the growth of shipping has left the graving dock too small for the larger vessels, it is still utilisable for those of smaller size. With the exception of the expenditure on the gates or caisson, which close the entrance, there is practically no maintenance outlay. It is all probably covered by an allowance of $\frac{1}{5}$ per cent. per annum.

The life of a floating dock, considered as an iron or steel structure constantly afloat, will vary with its situation, some waters being more destructive than others. It may be fairly reckoned at from 40 to 50 years, and there is no reason why, with reasonable care and attention, the longer period should not be exceeded. The annual cost of maintenance estimated from several existing examples lies between 1 and 1½ per cent. of the capital outlay. Any cost of dredging required to keep the berth clear of mud deposits must be added to this allowance.

Working Cost.—The amount of pumping to be done in the case of a graving dock generally greatly exceeds that in the case of a floating dock—it may be as much as 3 or 4 times. In a floating dock only so much work is required to be done as is necessary to raise the vessel and the deck of the pontoon above water level. The graving dock chamber, on the other hand, has always to be completely pumped out, no matter what may be the size of the vessel occupying it, and the smaller the vessel the greater the quantity of water to be pumped.

Small graving docks on the banks of a tidal river, in cases where time is unimportant, may be emptied of water by natural means. These cases are exceptional.

But even in the larger docks the cost of pumping is small in comparison with the capital charges. For one hundred dockings a year they probably do not amount to more than 10 or 12 per cent. of the whole annual expenditure.

* See footnote, page 606. The figures quoted are for comparative purposes only. They are not applicable strictly to present conditions.

Again, the pumps being only intermittently employed, one pumping station may be made to serve two graving docks, whereas each floating dock requires its own installation.

As regards general adaptability, floating docks have the advantage, already pointed out, of mobility, and they may be transferred from one port to another, whenever the circumstances require. This has already been done on several occasions to meet altered trade conditions, or for purposes of war.

A floating dock can be trimmed to take a vessel having a list which conceivably might interfere with its passage through a graving dock entrance with vertical sides. This operation was frequently performed during the war.

The mobility of a floating dock is not free from certain drawbacks. The dock may founder or be wrecked. The movement of floating docks is, however, very infrequent. It happens occasionally that a dock is constructed at a British shipyard for towage to a foreign destination.

All these varying considerations point to the conclusion that there is scope for both graving and floating docks, and that each may be employed to better advantage than the other under some particular set of local conditions.

SOME OF THE LARGEST FLOATING DOCKS IN THE WORLD.

Location.	Lifting Capacity.	Length.	Inside Width.	Depth of Water over Keel Blocks.	Owners.
	Tons.	Ft.	Ft.	Ft.	
*Southampton	60,000	960	130½	38	Southern Railway Co.
Singapore	50,000	855	126½	40	British Admiralty.
Rotterdam	46,000	695	142	35	Wilton's Eng. & Slipway Co.
Hamburg	46,000	575	123	33	Blohm & Voss.
Malta	65,000	962	140	38	British Admiralty.
Devonport	33,000	680	113	36	Ditto.
Hamburg	38,000	625	123	33	Blohm & Voss.
Portsmouth	32,000	680	113	36	British Admiralty.
Nikolaieff	30,000	656	136	30	Soviet Government.
Brooklyn, U.S.A.	30,000	685	100	31	Morse Dry Dock Company.

THE SOUTHAMPTON FLOATING DOCK.

The following particulars are extracted from a paper read before the Institution of Civil Engineers in January 1925, by Dr. E. H. Salmon (*Min. Proc. Inst. C.E.*, vol. 219):

Weights and Lifting Power.—The total net estimated weight of the dock in its final form was 18,990 tons, made up as follows:

	Tons.
Steelwork in hull, including mooring-booms	16,482
4½ per cent. for rivets	741
Joint bolts	17
Machinery	404
Fittings	736
Timber	407
Cranes, crane-rails, and collecting gear	183
Stores and spare gear	50
Proportion of weight of brow and moorings carried by dock	33
	19,053
Less half-weight of mooring-booms	63
Total	18,990

The above weight does not include any allowance for overweight and extras. From measured displacements of the finished dock its actual weight was found to be 19,330 tons. The displacement of the pontoon at Southampton (density of water = 35·09 cubic feet per ton) is 81,840 tons. At the conclusion of the test made to determine the lifting power of the dock, 1,530 tons of water remained in the pontoon. The true lifting power of the dock, therefore, is 81,840 — 19,330 — 1,530 = 60,980 tons.

Cost.—The original contract price for the dock delivered at Southampton was £375,000. The corresponding price of the steelwork in the hull comes to £17 per ton; the machinery, about £120 per ton; the fittings, about £32 per ton; and the timber, £22 per ton. A number of extra fittings over and above the original contract have been added, somewhat increasing the total cost of the finished structure. The above figures include the dock proper and mooring booms, but not the shore work, nor the preparation of the site.

* Information as to the post-war existence of this and others is lacking.

PARTICULARS OF SINGAPORE FLOATING DOCK.

(*Built* (1928) *by Swan, Hunter, & Wigham Richardson, Ltd., Wallsend-on-Tyne, for the British Admiralty for service at Singapore.*)

Lifting capacity	50,000 tons
Length	855 ft.
Breadth outside	172 ,,
Clear width between fenders	126 ft. 6 ins.
Height of side walls and pontoon	75 ft.
,, walls from pontoon deck	50 ,,
Steel weight, including joint bolts, castings, etc. . . .	21,750 tons
Number of rivets	3,500,000

Pumping capacity: 30,000 tons per hour, viz.:

Three pumps, each expelling 6,000 tons of water per hour.
Two ,, ,, ,, 4,000 ,, ,, ,,
Two ,, ,, ,, 2,000 ,, ,, ,,

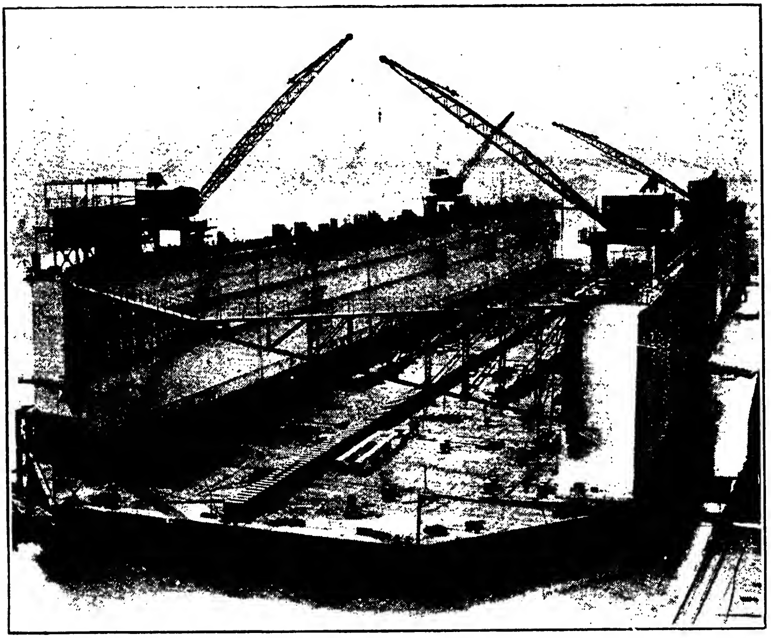


FIG. 25.—Singapore Floating Dock.

The pumps are of centrifugal type and driven by electric power. They can work at full capacity except when there is less than approximately 4 ft. of water in the tanks.

There are also four wash-down and fire-extinguishing pumps, each with a capacity of 75 tons per hour.

Electrical equipment:

(*a*) 1,000 volt, 3-phase, alternating current system for main generation and pumping.
(*b*) 220 volt, 2-wire, direct current system for auxiliary generation, lighting and power.
(*c*) 20 volt, 2-wire, direct current system for valve control in docking operations.
(*d*) Complete telephone system with central exchange.
(*e*) Provision for supplying ships in dock with electrical energy for lighting and power.
(*f*) Provision for supplying submarines in dock with electrical energy for charging large storage batteries, etc.

A view of the dock lying alongside the builder's yard is shown in fig. 25.

Notes on Floating Docks.

(Contributed by Swan, Hunter, & Wigham Richardson, Ltd.)

The development of different types of floating docks in this country is largely due to Messrs. Clark and Standfield, Westminster, who have designed docks of almost every kind, and for service in many parts of the world.

Floating docks may be conveniently divided into two classes: (*a*) those which are not self-docking, or 'box docks,' and (*b*) those which can be self-docked.

(*a*) The *box dock* is double sided and of └ ┘ section. It has two side walls, and is the only form of modern floating dock which cannot be divided into more than one part for the purpose of self-docking. The two side walls and the bottom pontoon are one permanent structure. Box docks of great size have been built: for instance, the British Admiralty floating dock of 33,000 tons lifting capacity built at the Wallsend Shipyard and stationed at the River Medway.

(*b*) By a 'self-docking' type is meant a dock divided into sections any one of which can be lifted on the remainder of the dock for painting or repairs. The following are some of the various types of self-docking docks: the *Bolted-Sectional*, the *Sectional Pontoon* or *Rennie*, the *Off-Shore*, the *Sectional Box*, the *Multi-Sectional*, and the *Depositing*.

The *Bolted Sectional Dock* combines the advantages of the great longitudinal strength of the '*Box*' type with facility for self-docking. This type is usually built in three sections of approximately equal lengths, the two end sections being stepped to form landings when carrying out a self-docking operation.

Fig. 26 shows, firstly, the whole dock assembled; secondly, the centre section lifted by the two end ones; and thirdly, one of the end sections lifted by the centre and other end section. Examples of this type of dock may be seen at Rotterdam, at Sténia in the Bosphorus, and at Walsh Island, N.S.W. (See fig. 31, page 631.)

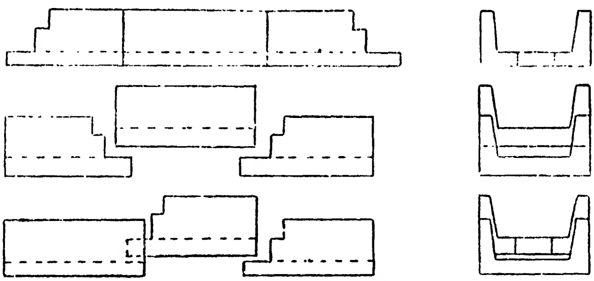


Fig. 26.—Bolted Sectional Floating Dock.

The Sectional Pontoon Dock (also called 'the Rennie Dock,' after its inventor), consists of a number of separate pontoons lying transversely to the length of the dock. Upon these pontoons and bolted to them rest two continuous side walls. The length of each pontoon, i.e. on the keel line, is such that when detached from the rest of the dock it can be passed end-wise between the walls of the latter and so lifted in the same manner as a ship. Each pontoon can be successively dealt with in the same manner. Fig. 27 shows first the whole dock assembled and then

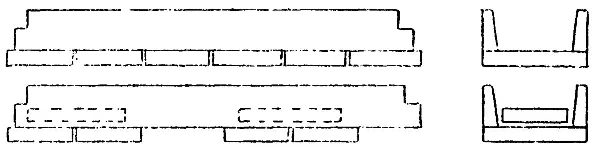


Fig. 27.—Sectional Pontoon Floating Dock.

two of the pontoons lifted by the remainder of the dock. This *Sectional Pontoon* type is very suitable for vessels of moderate dimensions, and owing to its form can easily be constructed in places where docks of other types could not be completed. The separate pontoons not being very large in size can be easily erected and launched, after which the walls can be built upon them.

Examples of this type of dock were built at Wallsend-on-Tyne for the Government of Southern Nigeria, and for service at Saratov on the River Volga.

The Off-Shore Dock is an ingenious design patented by Messrs. Clark & Standfield in 1884; a well-known example of it was built at Wallsend-on-Tyne for a ship-repairing establishment at Sunderland, and there is another good example of it also constructed by Messrs. Swan, Hunter, and Wigham Richardson, Ltd., at Penarth, near Cardiff. The off-shore dock (fig. 28), of which the end elevation is 'L' shaped, is very easily handled. The side wall is connected to the shore by booms or girders hinged at each end. When the dock is sunk to receive a vessel the

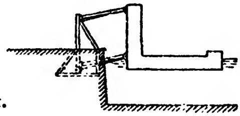


FIG. 28.—Off-shore Floating Dock.

latter can enter the dock from either end or may be warped in sideways. It is then rapidly centred by telescopic side shores and supported underneath by mechanical bilge-shores. This type of dock is divided into two sections, and as it has only one wall one-half of the dock can be lifted, so to speak, in the lap of the other. Fig. 30 on this page illustrates the self-docking operation.

The *Sectional Box Dock* (fig. 29) was originated by Messrs. Clark & Standfield, Westminster. It combines the advantage of ease and simplicity in self-docking possessed by the Sectional Pontoon dock, with the strength of the solid or *Box* dock. The dock is divided, like the ordinary sectional dock, into sections of such a length that any one section can be turned round 90 degrees and docked on the rest of the dock. In the Sectional Box dock, however, the sections are all

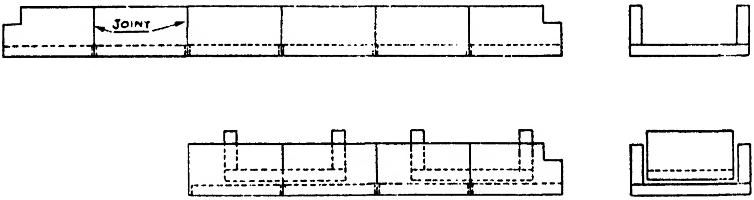


FIG. 29.—Sectional Box Dock.

joined together by a strong, rigid joint of a similar type to that used in the bolted sectional docks running round the whole profile of the dock. The 60,000-ton dock at Southampton, designed by Messrs. Clark & Standfield, the British Admiralty 50,000-ton dock at Singapore (fig. 25, p. 628), and the Wellington Harbour Dock (17,000 tons), New Zealand, both built by Swan, Hunter & Wigham Richardson, are notable examples of this type.

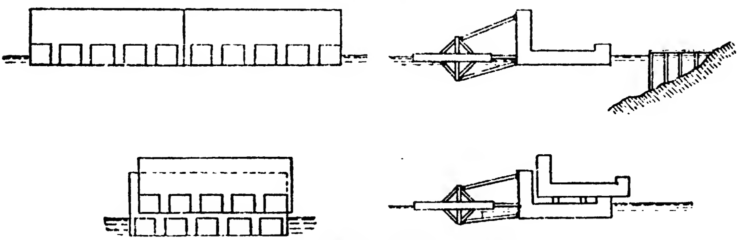
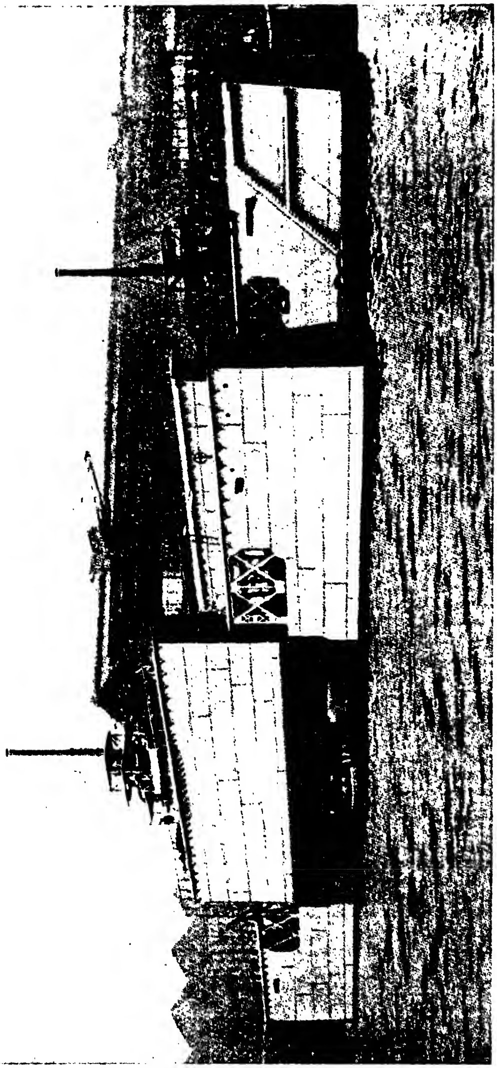


FIG. 30.—Depositing Floating Dock.

The *Multi-Sectional Dock* is a modification of the Sectional Dock type. In small docks it is not necessary to connect the bottom plating right across the width of the pontoon. The joints at the sections only occur, therefore, round the profile of the side wall, the spaces between the pontoons being decked over to form a continuous platform.



Floating Dock.

Fig. 31.—Bolted-Sectional Dock.

(Built by *Swan, Hunter, & Wigham Richardson, Ltd.*, Wallsend-on-Tyne, England.)

Self-docking operation; centre section lifted by the two end-sections.

The *Depositing Dock* was designed as early as 1877, by Messrs. Clark & Standfield. In section it is 'L' shaped. The single side wall is continuous, but the bottom pontoon is made up of a series of 'fingers' with spaces between them (fig. 30). The necessary stability is given to the dock by means of parallel booms hinged at each end and connected to a floating outrigger. This type of dock can not only dock vessels, but owing to the peculiar construction of the bottom pontoon is enabled to deposit them on a gridiron formed by groups of piles. There are spaces between each set of piles of such a width as to admit the 'fingers' of the dock. The dock carrying a ship is pushed forward so that the 'fingers' are engaged between the piles and then on sinking the dock the ship is left upon the gridiron. The wall of the dock is divided into two equal lengths, either of which can be docked by the other as in the case of the off-shore type.

Depositing docks at Kobe and Barcelona have been used for the purpose of lifting large, hollow ferro-concrete blocks for harbour construction. These were built upon the depositing grids, lifted off by the dock, and finally floated off so that they could be towed to their final position and sunk there. In the case of the Kobe dock, some of these monoliths were 120 ft. long by 41 ft. high, and weighed as much as 2,200 tons. In this way the difficulty of launching such large masses was avoided and much expense saved.

DREDGING.

Modern dredging appliances may be summarised under two classes, each with several subdivisions.

Class I.—Digging dredgers, comprising (*a*) the ladder, or elevator, dredger, with a continuous band of buckets; (*b*) the dipper dredger, with a single bucket at the extremity of a long arm, or lever; and (*c*) the grab or grapple dredger, with a drop bucket suspended at the end of a chain.

Class II.—Suction or hydraulic dredgers, comprising (*a*) the sand pump; and (*b*) the suction cutter dredger.

Rock cutters are not dredgers, but constitute an auxiliary class which will be treated separately.

Ladder Dredgers.

Ladder Dredgers, or Elevator Dredgers (sometimes in Europe called bucket dredgers), are characterised by high power and great serviceability under nearly any conditions. They are adaptable to the removal of every variety of material except the harder kinds of rock. The principle upon which they are constructed is that of an endless chain of buckets travelling in an elliptical path about two pivots, or tumblers, set one at each extremity of an inclined plane. The buckets excavate material at the lower tumbler and discharge it into a shoot while passing over the upper tumbler.

Single Ladder v. *Double Ladder Dredgers.*—Dredgers of the ladder type have either one or two ladders. In single ladder dredgers, the ladder is situated along the centre line of the vessel; in the other case, the ladders are placed at each side.

Single ladder dredgers are most in favour for ordinary operations. In machines of the same power, they have the advantage of few working parts and of less working friction. As seagoing craft, they possess a more convenient outline, and their lesser width enables them to pass through narrower waterways and lock chambers. But they cannot work so closely to a quay wall as side ladder dredgers, and if any breakdown occurs in the chain of buckets, the whole dredger is thrown out of action, whereas, with the double ladder dredger, this is not necessarily the case.

Capacity of Ladder Dredgers.—The capacity of the buckets of ladder dredgers ranges from 5 to 54 cubic ft., but, generally speaking, about 35 cubic ft. is looked upon as the maximum size consistent with convenient utility. A moderately sized dredger, with buckets of 20 to 25 cubic ft. capacity, travelling at speeds from 15 to 20 ft. per minute, and working in water of depths up to 50 ft., may be expected to raise from 400 to 500 tons of stiff clay per hour; in suitable material, such as ballast and gravel, the rate obtained may be as high as 1,000 tons per hour. The most powerful dredgers can double this performance.

Examples of Ladder Dredgers.—The following particulars relate to the 'Corozal,' a Hopper dredger, built in 1911 for the U.S. Government, for work on the Panama Canal:—

Dimensions.	Ft.
Length on load water-line	239
„ between perpendiculars	259
„ over all	268¾
Breadth moulded	45
Depth moulded	19½
Draught fully loaded	15½

Propelling speed: 10 knots.

Dredging capacity: 1,200 cubic yds. of mud and sand per hour.

Buckets: Two sets; one of 54 cubic ft. capacity for soft material and one of 35 cubic ft. for stiff clay.

Maximum dredging depth: 50 ft.

The 'Silurus,' built by W. Simons & Co., Ltd., Renfrew, is a twin-screw bucket hopper dredger of the following dimension:—

		Ft.
Length between perpendiculars		260
Breadth moulded		46
Depth moulded		2
Propelling speed	$8\frac{1}{2}$ knots.	
Dredging capacity	2,200 tons per hour.	
Buckets, two sets	one of 42 cu. ft. and one of 22 cu. ft.	
Maximum dredging depth	50 ft.	
Hopper capacity	1,500 tons.	

Both the foregoing vessels are of the hopper dredger type, that is to say, the buckets discharge into a hopper in the interior of the dredger. The following are two examples of dredgers loading into barges alongside. They are of recent construction by Ferguson Bros., Ltd., of Port Glasgow.

(1) Double side-ladder barge-loading dredger, 'Bulldog':—

	Ft.
Length between perpendiculars	156
Breadth moulded	43
Depth moulded	$12\frac{3}{4}$
Draught loaded	$8\frac{1}{2}$

Dredging capacity: 1,200 tons per hour of loose material, such as sand or ballast.
Buckets: Two sets, each of 15 cu. ft.
Maximum dredging depth: 50 ft.

(2) Central ladder dredger, 'Persevere':—

	Ft.
Length between perpendiculars	190
Breadth moulded	36
Depth moulded	$11\frac{1}{2}$
Draught loaded	7

Dredging capacity: 1,000 tons per hour.
Buckets: 30 cu. ft. capacity.
Maximum dredging depth: 50 ft.

Neither of the above vessels is self-propelling.

Utility of Ladder Dredgers.—Although the serviceability of ladder dredges does not extend to the removal of hard rock, yet they are capable of scraping away the surface of the softer varieties of sandstone and chalk, and, for this purpose, the bucket edges are sometimes fitted with spikes or tines. A ladder dredger works admirably in marl and stiff clay, and no difficulty even is found in dealing with heavy boulders which are frequently encountered in clays of glacial formation. The buckets of a first class dredger will readily lift stones weighing from 30 cwts. to 2 or even 3 tons apiece, bringing them up to the surface and within the range of a deck crane. As illustrative of the character of the material dealt with in many cases, the following particulars may be cited, relating to twelve months' work on the river Tees. A quantity of 417,963 cubic yds. was dredged at a cost of 22,432*l.*, equivalent to about 1*s.* $6\frac{3}{4}$*d.* per cubic yd. About two-thirds of this was hard, boulder clay and there were removed therefrom 1,651 stones, varying in weight from 5 cwts. to 4 tons apiece, of which 1,523 were raised by the dredgers unaided by divers; and, in addition, there were 19 tree trunks, ranging in length from 10 ft. to 20 ft., 16 tree roots and stumps, and 107 miscellaneous obstructions.

Dipper Dredgers.

Dipper Dredgers.—The dipper dredger is an American type, and though to be found in certain ports has not yet become popularised to any extent in Europe. It is mainly employed on the great lakes of the Northern Continent of America, and on river beds and channels when the working depth does not generally exceed 20 ft. or so, and where space for manœuvring is somewhat restricted. This is accountable for by the fact that the dredger is not manipulated by chain moorings, as in the case of the ladder dredger, but is maintained in position by vertical spuds, of posts, which pass down through the vessel to the bottom of the channel, and are raised and lowered as occasion may require. The dipper arm itself assists in manœuvring, being allowed to remain on the bottom while the position of the spuds is changed.

Capacity of Dipper Dredgers.—The single bucket, or scoop, at the end of the lever arm has a capacity ranging from 1 to 12 or 15 cubic yds. The bucket is operated in a curved upward sweep by suitable gearing, on the same principle as a steam navvy or land excavator. The speed of a good dipper dredger in sheltered inland waters is from 30 to 40 seconds per dipper load, and on the sea coast from 40 to 50 seconds. For hard material, the bucket, as in the case of the ladder dredger, is fitted with teeth.

Examples of Dipper Dredgers.—The following particulars relate to a very large dredger, the 'Onondaga,' which worked in New York Harbour :—

Length of hull	140 ft.
Breadth ,,	50 ft.
Depth ,,	15 ft.
Capacity of dipper bucket	12 cubic yds.
Dredging depth	50 ft.

Size of dipper arm : 80 ft. long by 36 ins. square.
Spuds : Four timbers, 80 ft. long by 5 ft. square.
The dipper arm and spuds are both of Oregon fir.
Engines : Double cylinder, condensing.
Cylinders : 20·5 ins. diameter by 24 ins. stroke.

Particulars of a dipper dredger made for Japan, by Lobnitz & Co., Ltd.

Length of hull	120 ft.
Moulded breadth	47 ft.
Moulded depth	12 ft.
Mean draught	7 ft.
Dipper capacity	8 cub. yd.
Maximum dredging depth	50 ft.
Normal reach from centre when dumping	64 ft.
Clear height of dump above water	15 ft.
Number of dips per hour, average	50
Output per hour free soil	400 cub. yd.

Two dipper dredgers, the 'Gamboa' and the 'Paraiso,' each with buckets of 15 cub. yds. capacity, were built by the Bucyrus Co. in 1914 for work on the Panama Canal.

These dredgers have a displacement of about 1,500 tons, and the hulls, which are of steel, have an over-all length of 136 ft., a width of 44 ft., and a depth at bow of 15½ ft. and at stern of 13½ ft. Each dredger has two dipper buckets, the larger, of 15 cub. yds. capacity, for soft material, and the smaller, of 10 cub. yds. capacity, for harder material. The buckets, of ¾-in. plate, have lips of manganese steel, 2 ins. thick, which are 50 ins. deep at the centre, reduced to 9½ ins. at the back edges. They have no teeth. The handles are of Oregon fir, 72 ft. long.

The rate of working is said to average 6,000 cub. yds. per day each, in two 8-hour shifts.

A dipper dredger at work in Bombay Harbour has a bucket capacity of 6 cub. yds., dredges to a depth of 40 ft., and is capable of lifting boulders of over 12 tons weight.

Comparison of Dipper Dredger with Ladder Dredger.—Compared with the ladder dredger the dipper dredger is not quite so serviceable a machine for all-round purposes, and, for the more intractable kinds of material, it has been demonstrated that the former is more efficient than the latter. Thus, in excavating a channel in indurated clay in the St. Lawrence River, dipper dredgers of the most powerful design failed and were replaced by a ladder dredger which completed the work in a satisfactory manner.

The Grab or Grapple Dredger.

The Grab or Grapple Dredger has a bucket which consists of two or more curved plates, or jaws capable of opening and closing under the action of suitable gear. The grab is suspended from the jib head of a crane which does the requisite raising and lowering. The working is actually done very largely under the influence of gravity, the bucket being allowed to fall with open jaws under its own weight and to bury itself in the ground, where the jaws are closed and the bucket with its contents is then withdrawn. The principle of the grab is simple, but there are many complex adaptations in practice. Thus, the appliance may be worked by one or by two chains, each of which arrangements admits of several variations, so that there are quite a number of different machines on the market, each possessing special characteristics and advantages.

Comparison of Single and Double Chain Grabs.—The drawback of a double-chain grab is that it cannot be fitted to an ordinary crane; but, on the other hand, it has less intricate and less delicate mechanism, and, consequently, does not run the same risk of getting out of order. With two chains, moreover, there is less likelihood of the bucket being lost through fracture of the hauling gear. There is the further consideration that in many, if not most, single-chain grabs, the full height of lift has to be made before the grab can be opened in order to discharge its contents. Double-chain grabs can discharge at any stage of the lift. This advantage is shared by a few of the single-chain systems. Its importance lies in the fact that the grab might close upon some immovable object below water, and therefore necessitate the sending down of a diver, unless the jaws can be opened mechanically in that position.

Utility of Grab Dredgers.—The grab is not a suitable appliance for extensive operations. It is best adapted to dealing with material in confined and awkward situations inaccessible to other types of dredger. It is thus excellently suited for the removal of mud and silt in the vicinity of dock entrances and passages. Silt is generally of so impalpable a character that any disturbance of the water in which it lies causes it to rise in suspension, and renders it difficult of removal. The grab proves a most effective appliance for dealing with this soft material, entrapping it with the least admixture of water and the least alteration in consistency. Moreover, the grab is an extremely convenient implement for picking up miscellaneous articles which find their way to the bottom of docks; logs of hard wood, rails, coal, bales of hay and wool, barrels of nails and cement, hoop iron, etc.

Capacity of Grab Dredgers.—Grabs are constructed of a capacity up to 10 cubic yards, but for general purposes the most useful size lies between 70 and 120 cubic ft. They work very successfully in water up to 50 or 60 ft. in depth, at a speed of about one dip per minute, raising in this way from 150 to 250 cubic yds. per hour.

Examples of Grab Dredgers.—Several excellent examples of this class of dredger are constantly at work in the Liverpool docks. The grabs are mounted on barges of 1,100 to 1,400 tons hopper capacity. There are four 5-ton or 7-ton cranes of 16 ft. and 19 ft. radius respectively on each barge, each actuating a grab of 70 or 115 cubic ft. capacity. The principal dimensions of the vessels are:—

1,100 ton barges	200 ft. long by 35 ft. beam by 15½ ft. deep.
1,400 ton barges	225 ft. long by 38½ ft. beam by 16 ft. deep.

The speed of the barges is about 11 knots.

Bag and Spoon Dredging.—This notice of dredging appliances would be incomplete without some reference to a very rudimentary form of apparatus still practised in restricted areas and where operations are on a small scale. This apparatus consists of a strong leather bag of a capacity of from one-half to one cubic yard attached to a steel rim at the end of a stout spar, or pole, from 20 ft. to 35 ft. long. The bag is kept at the bottom by means of the pole, while it is drawn along the surface by a line attached to the rim. The operation is performed by manual labour, and about half a dozen men are required to man a barge for the purpose. The quantity raised and loaded into the barge will range from 60 to 100 cubic yds. per day. The appliance is really only suitable for dealing with mud in timber ponds and in canals.

Suction or Hydraulic Dredgers.

Suction or Hydraulic Dredgers.—Turning now to the second class of dredging plant, the suction or hydraulic type may be described as consisting essentially of a continuous pipe, or tube, through which material is drawn up by suction from the bottom and discharged either into a hopper or through a long pipe line to a depositing site some distance away. The first application of the principle was to the removal of sand—hence the name Sand Pump—but it has since had a much wider application. As regards sand, the method proved extremely effective from the first, the light granular nature of the substance lending itself admirably to easy incorporation with the body of water flowing into the pump intake. Such difficulty as has been experienced has been merely in regard to securing the deposition of the sand within the hopper, the tendency being for a large proportion to flow out over the sides with the escaping water. Considerable reduction in this unremunerative pumping has been made by giving the effluent stream as long a distance to travel as possible and providing the channel with adjustable coamings.

Sand Pump Dredgers.

Sand Pump Dredgers.—For the purposes of suction dredgers, centrifugal pumps are universally employed, and of these, those with a single inlet have an advantage over pumps with a double inlet, in that there is less concussion and interference with the flow. A double inlet pump, on the other hand, disposes of any question of balancing, which has to be met in a single inlet pump by thrust bearings. The most efficient peripheral speed is stated by the Mississippi River Commission to be about 50 ft. per second. The form of the nozzle of the intake pipe materially influences the percentage of sand raised, which may be anything up to 50.

Examples of Sand Pump Dredgers.—The largest sand pump dredger in existence at the present day is the 'Leviathan' of the Mersey Docks and Harbour Board. Her dimensions are 465 ft. 9 ins. long between perpendiculars, by 69 ft. wide by 30 ft. 7 ins. deep moulded. She is fitted with four suction pumps, each of 42 ins. diameter, and is capable of filling herself with 10,000 tons of clean sand in 50 minutes from a maximum depth of 70 ft. The speed of the vessel is 10 knots. On arrival at the depositing ground, she discharges herself in 10 minutes through large cylinders extending from valve openings, 5 ft. 6 ins. diameter, in the bottom of each hopper to deck level. These valves have a lift of 4 ft. and are operated by hydraulic rams. At work on the bar and entrance channel of the Mersey, the Leviathan pumps up over 12 million tons of sand per annum.

The following particulars relate to the 'Blesbok,' the largest vessel of its type so far constructed by Lobnitz & Co., Ltd. It was supplied to the South African Railways and Harbours Administration.

Length between perpendiculars	304 ft.
Breadth, moulded	54 ft.
Depth	23 ft.
Maximum dredging depth	70 ft.
Speed	11 knots

Loading capacity: 2,600 cu. yds. of sand in 45 minutes and able to pump out the same load through a discharge pipe 2,500 ft. long in well under 60 minutes (*vide* 'The Engineer,' May 19, 1939). Triple expansion engines with cylinders 18 in., 31 in., and 51 in. × 30 in. stroke, develop 3,000 h.p. at 120 r.p.m. with steam pressure at 200 lbs. per sq. in.

Particulars of a number of other types of suction dredgers are given in the table on p. 637.

Suction Cutter Dredgers.

Suction Cutter Dredgers.—The success attending the introduction of the sand pump led to an investigation of the possibility of dealing with less tractable material by the same means. Water jets for loosening and disintegrating hard and compact strata were added, and some satisfactory results being obtained, the further step of introducing rotary cutters was taken. There was found to be little difficulty in treating in this way earth of a marly or brittle nature, easily resolvable into disconnected particles. Trouble, however, was experienced in dealing with clay and tenacious material, which clogged the cutters and choked the intake. Eventually this was overcome by the evolution of cutter blades of a suitable form and pitch.

Capacity of Suction Cutter Dredgers.—Suction cutter dredgers are now constructed capable of removing over 1,000 cub. yds. of stiff clay per hour. In material usually dealt with by bucket dredgers, the cutter suction and discharging dredger 'Lord Willingdon' at Cochin dredged and discharged, through 4,500 ft. of pipe line, 1,500 cub. yds. per hour over an extended period, and on occasion this output has been considerably exceeded.

Use of the Pipe Line.—The use of the pipe line for discharging purposes is becoming very general, especially in cases where it is practicable to use the dredged material for filling and embankment purposes. These lines are often of great length. A dredger working at Oakland, California, U.S.A., has successfully delivered material through 6,170 lineal ft. of 20-in. diameter pipe. Diameters reach as high as 42 ins. when altogether afloat, and 30 ins. to 36 ins. when partly ashore. In calm water thin sheet tubing will suffice, but in exposed situations long lengths of solidly built sections must be provided. The carrying capacity of a 36-in. pipe with 80 per cent. of solid matter conveyed at a rate of 10 ft. per second would be 7,400 cubic yds. per hour. Sand causes considerable wear and abrasion of the piping and requires a high percentage of water with greater velocity to keep it in suspension. Clay, from its unctuous nature, exercises far less frictional resistance and is sometimes rendered sufficiently fluid by 10 per cent. of water. A velocity of 7 ft. per second for clay and mud, and 10 ft. per second for sand, has been found to give good results.

Compound Dredgers.

Compound Dredgers.—A compound type of dredger has recently been introduced fitted with both the suction, or suction cutter, apparatus and a bucket ladder. This enables the dredger to be applied to a wide variety of material in the most effective manner. The bucket ladder is generally placed in the centre well and the suction apparatus at the sides of the vessel, but in some cases the suction pipe is also centrally situated.

Example of Compound Dredgers.—The dredger 'La Loire,' built for service on the river of that name, is a twin screw combined bucket and suction dredger with cutter gear and pipe line shore delivery. Her dimensions are 173 ft. 10 ins. in length by 32 ft. beam by 14 ft. deep. She is fitted with two centrifugal suction and force pumps, capable of raising material from a depth of 25 ft. and of driving it ashore through floating delivery pipes about 30 ft. long and 24 ins.

PARTICULARS OF LARGE RIVER BAR DREDGERS.

Name of Dredger	'Leviathan'	'M.O.P. 217C.'	'Rietbok'	'Pierre Lefort'	'Chien She'	'General Goethals'
Year built	1909	1929	1930	1934	1935	1938
Builders	Cammel, Laird	Werf Gusto	Simons	Deschimag	Schichau	U.S. Corps., Bethlehem, U.S.A.
Length, b.p.	465 ft. 9 ins.	325 ft. 0 ins.	374 ft. 0 ins.	337 ft. 0 ins.	360 ft. 0 ins.	476 ft. 0 ins.
Beam, moulded	69 ft. 0 ins.	52 ft. 6 ins.	57 ft. 9 ins.	54 ft. 0 ins.	60 ft. 0 ins.	68 ft. 0 ins.
Depth, moulded	30 ft. 7 ins.	23 ft. 0 ins.	25 ft. 9 ins.	26 ft. 0 ins.	26 ft. 6 ins.	33 ft. 0 ins.
Designed loaded draught	23 ft. 0 ins.	18 ft. 0 ins.	18 ft. 0 ins.	19 ft. 0 ins.	18 ft. 0 ins.	25 ft. 0 ins.
Material designed for	Sand	Light mud	Sand	Various	Firm mud	Sand
Type of dredger	Moored suction Flexible head	Drag suction Scooping head	Moored suction Flexible head	Trailing suction Flexible Head	Drag suction Scooping head	Trailgrid
Hopper capacity to deck level, cubic yds.	6,700	2,610	3,700	2,610	3,700	5,100
Dredging rate, cubic yds. per hr.	8,000	8,000 contract 14,000 trials	4,000	3,300	Normal 8,000	4,000
Number and size of suction pipes	4 at 42 ins.	1 at 37 ins.	1 at 42 ins.	2 at 25½ ins.	1 at 42 ins.	2 at 30 ins.
Pumping engines, I.H.P.	2,800	2,000	1,350	2,000	2,400	about 2,500
Propelling engines, I.H.P.	4,000	2,000	3,000	3,000	3,000	about 5,000
Total I.H.P. main machinery	6,800	4,000	4,350	5,000	5,400	about 7,500
Type of machinery	Steam	Steam	Steam	Diesel electric	Steam	Diesel electric
Speed in knots, loaded	10·5	9·0	11·5	11·0	11·0	10½
Max. dredging depth, ft.	70	50	75	65	45	about 50
Location of service	Mersey Bar	Punto Indio	Durban Bar	Bordeaux Bar	Yangtsze Bar	Ambrose Channel

From Brochure on Dredger 'Chien She,' by Dr. Herbert Chatley. With addition from 'Dredging Machinery,' paper in *Journal of Inst. C.E.*, April 1945.

diameter. The suction pipe is 26 ins. diameter and works, together with the bucket ladder, in a centre well.

The cutter has a diameter of 6 ft. 10 ins. and is driven at 12 revolutions per minute, the engine running at 140 revolutions.

The bucket ladder enables the dredger to work to a depth of 52 ft. 6 ins. The buckets have a capacity of 27½ cubic ft. They are driven at a speed of 16 buckets per minute with the engine running at 170 revolutions.

Both suction and bucket apparatus enable the dredger to excavate her own flotation in advance.

The machinery consists of two complete sets of direct acting, triple expansion surface condensing, reversible propelling engines, each arranged to drive a centrifugal pump as well as the bucket dredging gear. Each set of engines indicates about 450 h.p. when running at 170 revolutions per minute. The cylinders are 13 ins., 21 ins., and 33½ ins. by 18 ins. stroke.

The cutter gear is driven by means of belting from a separate engine of 200 h.p., which also serves for a centrifugal pump delivering spoil from the buckets into the upper part of the shoot.

For the purpose of working the bucket dredging apparatus the suction pipes and cutter gear are dismounted, but in deepening the bed of the river Loire the latter proved so effective that over 13 million cubic yds. of alluvial material have been removed without recourse to the bucket apparatus.

Hopper Service.

When dredged material has to be removed some distance to a depositing ground it is excavated by the dredger into its own hopper, or into hopper barges ranged alongside.

A combined hopper dredger costs less in initial expenditure and subsequent upkeep than a separate dredger and hopper of corresponding capacity, and it occupies less valuable space in a crowded waterway.

On the other hand, it involves the discontinuity of dredging operations and the frequent getting out and taking in of moorings. This latter consideration is of importance more to dredgers of the bucket ladder type.

On the whole, it is found expedient on undertakings of considerable magnitude, and in cases where the depositing ground is several miles distant from the dredging site, to maintain a fleet of hopper barges in attendance on bucket ladder dredgers. Suction dredgers, when they do not discharge through a pipeline, more commonly are compound vessels with self-contained hoppers.

Capacity of Hopper Barges.—Barges are constructed capable of receiving from 500 to 1,000 cubic yds. of dredged material (say 650 to 1,350 tons) and of making journeys to sea and back at speeds of 10 to 11 knots.

Example of Hopper Barge.—A typical barge of 800 tons capacity has a length of about 190 ft., breadth 30 ft. and depth 14 ft. The engines are triple expansion, surface condensing, with cylinders 16½ ins., 26 ins. and 43 ins. diameter by 27 ins. stroke, which at 110 revolutions per minute develop 820 i.h.p. and give a speed of 10 knots, steam being supplied at 180 lbs. per sq. in. from marine multitubular boilers.

Cost of Dredging Plant.*

The following figures may be useful as approximate indications of the cost of dredging plant which must, of course, vary from time to time with fluctuations in the price of material and the engagements of the shipbuilding yards.

Ladder Dredgers.—A 5 cubic ft. bucket, single ladder dredger, capable of raising 150 tons of soft material per hour from a depth of 40 ft., has cost in round figures 7,000*l.*

A 15 cubic ft. bucket, single ladder dredger, capable of raising 400 cubic yds. of sand, gravel and clay per hour from a depth of 45 ft., has cost about 27,000*l.* and was constructed in six months.

A one cubic yd. bucket, single ladder dredger, capable of raising 750 cubic yds. of ballast per hour from a depth of 55 ft., has cost about 43,000*l.*, and was built in nine months.

Dipper Dredgers.—A one to two yard dipper dredger costs from $8,000 to $12,000; say 1,600*l.* to 2,400*l.* The hull is of timber, about 60 ft. long by 25 ft. wide, with a draught of 3 ft.

A 3½ to 4 cubic yd. machine dredging to a depth of 30 to 35 ft., would cost about $80,000 to $90,000, say 16,000*l.* to 18,000*l.*

A 10 cubic yd. machine dredging to a depth of 35 ft. would cost about $130,000 to $140,000, say 26,000*l.* to 28,000*l.*

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* See footnote, page 606.

Grab Dredgers.—A twin-screw hopper dredger, with a hopper capacity of 1,200 tons, and fitted with 5 hydraulic grab cranes each of 5 tons bucket capacity, working to a depth of 47 ft., has cost about 35,000*l.*

Suction Dredgers.—A suction hopper dredger with a hopper capacity of 70,000 cubic ft., capable of raising 3,500 to 4,000 tons of sand per hour from a depth of 70 ft., has cost about 75,000*l.* A 2,000 ton suction hopper dredger would cost about 50,000*l.*

Compound Dredgers.—A sea-going hopper dredger, fitted with both ladder buckets (18 cubic ft.) and suction pumps, and capable of dredging by the former means 400 cubic yds. per hour, and by the latter means 650 cubic yds. per hour, from a depth of 50 ft., has cost about 53,000*l.*

Hopper Barges.—A steam hopper barge of 800 cubic yds. capacity, with a speed of 11 knots costs from 18,000*l.* to 20,000*l.*

Cost of Dredging Operations.*

Conditions and circumstances vary with each locality, so that it is difficult to give reliable figures for general guidance. Sand pump dredging has been done for as little as 1*d.* per cubic yd. or even less, and grab dredging from 2*d.* to 6*d.*, but a very great deal depends on the distance the spoil has to be carried. In bucket dredging, either by ladder or dipper, the range of cost is considerable, depending in addition, as it does, on the kind of material dealt with. The following are a few collected records from various sources:—

Bombay Harbour and Docks.

Total Maintenance Dredging during Five Years, 1920–1925.†

Costs exclusive of Interest and Depreciation on Dredging Plant.

Location.	Total Quantity Lifted: Cubic Yards.					Cost per Cubic Yard: Annas.				
	1920–1921	1921–1922	1922–1923	1923–1924	1924–1925	1920–1921	1921–1922	1922–1923	1923–1924	1924–1925
Alexandra Dock	24,210	297,500	20,380	144,800	388,000	9·23	7·78	8·01	5·78	6·62
Prince's Dock	50,500	263,500	68,500	17,700	474,000	7·60	6·81	7·30	4·75	5·50
Victoria Dock	159,500	114,000	48,900	344,000	49,000	6·22	6·18	6·87	5·88	10·60
Basins	206,800	131,400	3,335	—	9,330	7·78	12·26	11·60	—	9·10
Alexandra Dock Channels	650,000	149,600	615,000	591,000	1,170,000	4·40	8·56	5·76	4·97	3·60
P. & V. Docks Channels	689,800	1,098,000	931,500	857,200	1,098,800	5·37	5·01	5·38	4·54	3·83
P. & V. Channels, S.E. Flank	40,700	27,500	—	182,800	82,600	6·83	3·61	—	4·17	3·66
P. & V. Docks Channels, N. Flank	276,500	72,500	184,800	316,000	400,000	4·98	6·18	4·67	4·61	3·44

Port Natal.—The following particulars (extracted from a paper by Mr. William Brown on 'Recent Progress in Dredging Machinery'; vide *Min. Proc. Inst. C.E.*, vol. cciii.) relate to the performances of the four principal dredgers at Port Natal, Durban, within a period of several years ended December 31, 1914. The costs include all charges for working, maintenance, insurance, depreciation and interest on capital. The price of coal consumed by the dredgers was low, and seldom exceeded 8*s.* per ton. The three large vessels (Labrus, Cetus, and Nautilus) each employed fourteen Europeans, eighteen Kaffirs, and eleven Indians, or a total complement of

* See footnote, page 606.

† Extract from paper on 'Maintenance Dredging in Bombay Harbour and Docks,' by E. L. Everatt, *Min. Proc. Inst. C.E.*

forty-three hands, at an annual cost of about 3,850*l.* per vessel. 'With the possible exception of Liverpool, no port hampered by sandbars has made such progress in comparatively recent years as Durban.'

PARTICULARS AND WORKING COST OF SEA-GOING PUMP DREDGING PLANT AT PORT NATAL.

Name of Dredger.	Year Built.	Tonnage. Net.	Tonnage. Gross.	Hoppr. Capacity (Tons of Sand).	Pumping Speed (Tons of Sand per hour). Contract.	Pumping Speed (Tons of Sand per hour). Record.	Average Spoil handled per ann. (Tons).	Average number of days worked per ann.	Cost per ton handled. (*d.*)	Remarks.
Labrus .	1911	1,215	2,097	2,000	2,000	2,000	719,500	244	6·806	Worked exclusively in stiff clay, being fitted with patent clay cutter. All spoil dredged re-pumped ashore from ship's hoppers for land reclamation.
Cetus .	1905	1,327	2,414	3,000	3,000	7,430	2,151,000	232	2·227	Largely employed for bar work. Spoil mostly coarse loose sand and fine stone, all dumped at sea, two miles from harbour entrance.
Nautilus	1903	1,050	2,034	2,500	3,000	7,894	1,466,000	217	3·109	Employed chiefly on maintenance and deepening of harbour bed and channels. Material mostly fine sand, with occasional admixture of loose clay and mud, all deposited at sea two miles from harbour entrance.
Snipe .	1903	180	265	200	300	310	358,000	256	3·201	Used for clearing up wharf and quay berthage, cutting and maintaining boat channels, occasional reclamation, etc. Spoil deposited within harbour.

Panama Canal.—During 1910 the sea-going suction dredgers 'Culebra' and 'Caribbean' removed 6,083,902 cubic yds. at a cost of 3·7*d.* per cubic yd. The ladder dredger 'Marmot' removed 1,372,665 cubic yds. at 7·82*d.*, and dredger No. 5 removed 560,622 cubic yds. at 13·65*d.* per cubic yd. The small 5 cubic yd. dipper dredgers 'Cardenas' and 'Mindi' removed 1,013,485 cubic yds. at 27·89*d.* per cubic yd. and the 20-in pipe-line suction dredgers, Nos. 82 and 85, dealt with 2,472,822 cubic yds. at a cost of 14*d.* per cubic yd. During a period of twelve months ended May, 1916, the dipper dredgers 'Gamboa' and 'Paraiso' each removed over 2,900,000 cubic yds., at costs varying from 2·25*d.* to 5·45*d.* per cubic yd., including operation and maintenance.

Mississippi River.—In 1907 three pipe-line suction-cutter dredgers dredged 1,279,000 cubic yds. at 4·12*d.* per cubic yd. including the cost of idle time. In 1908 the same dredgers accounted for 989,000 cubic yds. at 5¼*d.* per cubic yd.

New York Harbour.—In 1909, the sea-going suction dredgers 'Manhattan,' 'Atlantic,' 'Raritan,' and 'Navesink,' removed 10,786,638 cubic yds. at a total cost of $424,734 (say 85,000*l.*) or at the rate of 1·97*d.* per cubic yd. made up as follows: Dredging, 1·785*d.*; surveys and examinations, ·125*d.*; contingencies and supervision, ·06*d.* The 'Delaware' removed 1,675,562 cubic yds. at a total cost of 2·39*d.* per cubic yd.

River Thames.—The contract cost of removing hard material (ballast and clay) by ladder dredger from the Thames below London Bridge has ranged at various times before the war from 1*s.* 4*d.* to 2*s.* 6*d.* per cubic yd. largely dependent on the distance the material had to be conveyed. Some work has been carried out departmentally, including all establishment and capital charges, at a figure as low as 7½*d.* per cubic yd. This was for a quantity of about one million cubic yards, and the distance conveyed was about 30 miles.

River Mersey.—At the bar and in the entrance channels of the Mersey nearly 15 million cubic yds. of sand were removed annually at a pre-war cost of ·6*d.* per cubic yd. exclusive of interest and depreciation on the plant.

Siam Irrigation Canals.—The following particulars of the working expenses of a 2½-cu. yd. dipper dredger, employed on the Siamese Irrigation Service, are given by Lobnitz & Co., Ltd.:—

'During the year 1920, working nine hours per day, the total quantity dredged was 312,130 cu. yds., measured in place, or an average of 26,010 cu. yds. per month. The best month in that year was October, with 43,700 cu. yds. at a total cost of 1·95 pence per cu. yd., and the average cost per cu. yd. for the year was 2·87 pence, including labour, fuel, supplies, repairs, and every expense except interest and depreciation. The fuel used was oil, and represented about half the total operating cost. The quantity used was about one ton per day.

'While larger outputs have been made for short periods and under more favourable conditions, this record is notable chiefly for steady running for a long period without appreciable delays or expense on account of the machinery, and this in a tropical country remote from base of supplies. The cost of repairs was 4·3 per cent. of the total cost, and consisted mainly in the replacement of some wire ropes.'

SUBAQUEOUS ROCK REMOVAL.

The removal of hard rock under water is generally effected by one or other of two means: (*a*) by drilling and blasting; and (*b*) by the use of the chisel or rock breaker. In both cases a dredger is subsequently required for the purpose of clearing the débris from the site.

Drilling and Blasting.

Drilling and Blasting.—The first is the older method and remains still a most effective means of dealing with rock of adamantine hardness in huge solid masses, such, for instance, as reefs of granite and other igneous rock, and for the harder varieties of limestone.

The holes are drilled by suitable plant at distances apart dependent upon the nature of the rock and the quantity to be removed. In very extensive operations chambers may be formed for the reception of powerful charges.

Two classes of drill are generally available for boring purposes: (*a*) diamond drills and (*b*) jumper drills. The former are hollow rods, fitted with a head or crown of diamonds, costing from 50*l.* to 60*l.*, and driven at from 200 to 300 revolutions per minute. The latter are bars or shafts of steel, furnished with bits, preferably of circular form, which are worked by percussion at the rate of from 20 to 30 blows per minute.

The blasting material is usually dynamite, or any of the numerous compounds of nitro-glycerine to be found on the market.

The following are brief notes on the methods employed in several instances.

Blyth Harbour.—The rock at Blyth harbour is sandstone of the hardness of basalt in places, and the work was done by six drills lifted by steam power and guided by hand. The distance between the shot-holes was 5 ft. in one direction and 6 ft. 2 ins. in the other. The blasting material, bellite, was lowered in canisters through the drilling tubes and fired by fuses and detonators, the holes being tamped with small gravel. The cost of drilling and blasting came to 3*s.* per cubic yd. of rock *in situ*, adding to which 2*s.* 6*d.*, the cost of dredging, and 8·2*d.* for interest and depreciation on the dredger, the total amounts to 6*s.* 2·2*d.* per cubic yd. The average quantity of rock drilled and blasted per week by one barge was 488 cubic yards. The system was afterwards abandoned in favour of the rock-cutter.

Black Rock Harbour, Buffalo, U.S.A.—Here the rock formation is a very hard limestone containing veins of flint. The blasting material was dynamite and, for a quantity of 14,450 cubic yds., the holes were drilled to an average depth of 9 ft. 9 ins. The cost of drilling and blasting only, including an allowance for interest and depreciation of 2 per cent. per month on 8,000*l.* (the value of the plant), came to 2*s*. per cubic yd. On the other hand, a smaller quantity of 333 cubic yds. with holes drilled only to 3 ft. 6 ins. deep cost 23*s*. 2*d*. per cubic yd.

The drill boat employed on this work was a steel hull, 137 ft. by 33 ft., divided into 20 water-tight compartments. It was equipped with five drill frames located along one of the gunwales each with a travel of 25 ft. on longitudinal tracks. Each frame was fitted with an Ingersoll-Rand drill 2½ ins. in diameter, with 4-in. bits, having a vertical range of 21 ft. without changing the drill bars. Each frame was also provided with an engine for hoisting the drills, two davits for handling drill steels and charging pipes and a pipe which was driven, whenever required, through loose material to the rock to prevent the drill holes from clogging. A 250 h.p. boiler supplied steam to the drills, spud engines and capstans.

Detroit River.—On three sections of the Detroit River, where the blasting material was dynamite and the holes ranged from 5 ft. to 8 ft. in average depth, the total cost is stated to have been 3*s*. 4*d*., 4*s*. 4*d*., and 4*s*. 1*d*. per cubic yd., respectively, of useful material, or about half the rates for the whole quantity removed. The holes were spaced 5 ft. apart, and the total number drilled was 94,691 at a cost of about 52,000*l*. It is not apparent that this figure includes any allowance for interest and depreciation.

Fremantle Harbour, Western Australia.—On account of the shallowness of the water, the work of drilling at Fremantle Harbour could be carried on from light wooden stages resting on four-footed trestles. In a couple of days it was possible to erect 20,000 square ft. of this staging, which carried 120 to 160 men drilling with regularity in 20 ft. or more of water. The holes were from 8 ft. to 12 ft. apart, and the charges consisted of 12 to 15 lbs. of dynamite or gelignite. One ton of explosive accounted for 5,600 cubic yds. of rock blasted. The cost of drilling and blasting 1,503.099 cubic yds. was 238,345*l*., equivalent to 3*s*. 2*d*. per cubic yd.

Fårösund, Sweden.—The cost of a navigable channel dredged in limestone at the island of Gothland in the Baltic Sea, completed in 1934, was as follows: mass dynamited, 42,000 cubic m. *in situ*; metres drilled, 31,342; mass dredged, 40,000 cubic m. *in situ*; total cost of dynamiting, 5·93 swedish crowns per cubic m.; cost of dredging, 5·85 swedish crowns per cubic m. —Report by Capt. L. Lawski to Sixteenth Navigation Congress.

Rock Breakers or Chisels.

Rock breakers, or chisels, are steel rams weighing from 15 to 20 tons, with pointed ends of hard chrome steel which are detachable for renewal. The appliance is suspended from a frame, or tripod, mounted on a barge, and is worked by a winch. A drop of from 6 ft. to 10 ft. is generally sufficient to produce disintegration. Blows can be repeated with advantage until a depth of 3 ft. is reached, with holes spaced about 3 ft. to 4 ft. apart (about eight blows will give the required penetration), when it becomes desirable to dredge the broken material before proceeding further. A net depth of 2½ ft. over the site will probably thereby be obtained. It is found that the average result in hard rock is 2 cubic ft. per blow and that about 150 blows can be given per hour. This is equivalent in uninterrupted working to about 10 cubic yds. per hour for a single-cutter machine; about 50 per cent. more will be achieved by a double cutter. In moderately hard rock the output will be greater, as about 250 to 300 blows can be given per minute. About 200 cubic yds. may be reckoned a good day's work.

There are several types of cutter, but perhaps the Lobnitz rockbreaker is that which is best known. The following are some instances of its employment. It may be added that some of the latest examples of this type of cutter have been fitted with an under-water sleeve, to afford more accurate guidance for the ram or chisel.

Suez Canal.—On the Suez Canal, where the rock consisted of limestone, calcareous agglomerate, and tufa, gypsum and alabaster, 132 blows were delivered per hour of effective working, or 83 blows per hour of whole time, allowing for interruptions from passing ships and stoppages for repairs. The average thickness of rock shattered was 2 ft. 7½ ins., and each cubic yd. of rock required a mean of 5·4 blows. The cost, exclusive of dredging but covering everything else except general charges, varied from 2¼*d*. to 7*s*. 6½*d*. per cubic yd.

Blyth Harbour.—At Blyth in hard sandstone of the coal measures, a ram of 15 tons weight, falling through a distance of 8 ft., penetrated the rock to a depth of 3 ft. on an average of 8 or 9 blows. The holes were spaced 4½ ft. apart. The plant comprised two Lobnitz rock-breakers, each consisting of a steel barge carrying sheer legs, from which the rams, 40 ft. to 50 ft. long and 17 ins. to 19 ins. diameter, were suspended. The cost of one rockbreaker was about 6,800*l*. The quantity broken per week by a single machine working night and day was 908 cubic yds., at a cost (including wages, coal, stores, repairs and insurance) of 46*l*. 11*s*. 3*d*., equivalent to 12·3*d*. per cubic yd. Adding 6½ per cent. for interest and depreciation on plant, the total cost was 14·5*d*. per cubic yd. The cost of subsequently dredging the loose material (including capital charges) was 2*s*. 9·1*d*. per cubic yd., making 3*s*. 11·6*d*. per cubic yd. in all.

Black Rock Harbour, Buffalo, U.S.A.—The cost of working two Lobnitz machines at Black Rock Harbour in hard limestone rock was 2*s*. 9½*d*. per cubic yd. for a quantity of 52,500 cubic yds. This does not cover the cost of dredging.

Panama Canal.—On the Panama Canal, a Lobnitz cutter worked mounted on a barge 100 ft. long by 28 ft. beam by 8 ft. deep. The extreme variation in the tides necessitated the use of 3 rams, weighing 15, 16 and 19½ tons respectively. The holes were spaced 4 ft. apart, and 3 ft. of penetration was made in each case. From records during 1910–11 the work done was an average of 16·2 cubic yds. per hour of actual working time, or 11 cubic yds. per hour of total time, at an average cost of 3*s*. 4*d*. per cubic yd.

Value of the method of Drilling and Blasting as Compared with the use of Rock Breaker.—While drilling and blasting is most commonly adopted for hard rock in large masses, it does not necessarily follow that the cutter always gives better results in soft rock. In some cases it is found that the cutter has a tendency to stick in the hole which it makes, and for that reason the extent of penetration should be carefully limited. On the other hand, the cutter offers less interruption to the passage of ships close to the area of operations, and it can be used in positions where blasting is inadmissible. Moreover, the rock is broken to a more uniform and more convenient size, which facilitates the action of dredging. The cost of the work done by the cutter is generally, though not invariably, less than the cost by the alternative method.

Firth of Forth.—The breaking of fine-grained granite, known locally as Whinstone, at an average depth of 43½ ft. below water-level, cost in 1923–25, about 34*s*. per cubic yd. of rock *in situ*, including repairs and replacement of plant. The total cost of breaking up and removing the broken rock varied considerably, but, for the hardest granite, worked out at about 56*s*. per cubic yd. *in situ* for the most difficult and toughest part of the work.—Report by N. G. Gedye to Sixteenth Navigation Congress.

DIVING.

Though naked diving is still practised in some parts of the world—*e.g.*, in the Greek Sponge Fisheries and on certain pearling grounds in the East—this primitive method, which has been responsible for much physical suffering and a high rate of mortality, has for many years been giving way steadily to the safer and more comfortable diving dress.

The methods of diving with apparatus in general use to-day are (1) the diving bell, and (2) the flexible closed dress originally designed in 1837 by Augustus Siebe, his principle still being in universal use. The air supply is (*a*) by pump worked manually or by electric or oil motor; (*b*) by air compressor, belt or power-driven, in connection with air reservoirs, for use in case of breakdown of the compressor; and (*c*) by banks of large cylinders of compressed air at pressures of from 2,000 to 3,000 lbs. per square inch, with reducing valves and air control panel. In all cases, care must be taken that the air sent down to the diver is clean and free from oil-vapour and other impurities.

In recent years, as a result of research and practical experiment, much progress has been made as regards depth to which the diver, equipped with the flexible dress, can work in safety, as is evidenced by the following tables compiled by Siebe Gorman & Co., Ltd., after the work of the Admiralty Deep Diving Committee of which Sir Leonard Hill, F.R.S., and Sir Robert Davis, were members. These tables include decompression times for the diver's ascent after work at various depths for different periods.

Compressed air illness—or caisson disease, or 'bends,' as it is variously called—is caused by too sudden release of the excess of nitrogen in the blood and tissues; it is absolutely essential, therefore, to decompress in accordance with the tables. It will be seen that the breathing of oxygen at certain stages during the return to normal atmospheric pressure greatly reduces the decompression period.

For decompressing from the greater depths, divers of the Royal Navy use Sir Robert H. Davis's submersible decompression chamber which is lowered to the diver's first stopping place where he enters the chamber through a door in the bottom which is closed by an attendant; the chamber is then hauled to the deck of the diving vessel where the diver is decompressed in comfort and under observation. Thus he is saved the tedium and discomfort and cold of decompressing in stages on the shot-rope as by the old system.

Another form of diving dress which has been tried at different periods is a metal armour with articulated legs and arms. While this dress has proved satisfactory in protecting the diver from the external pressure of water (the air within the dress is at normal atmospheric pressure), it allows him so little mobility that it has proved of less practical value than the steel observation chamber. In proof of this, work on the 'Egypt' treasure-recovery operations was first started with the metal armour, but had to give way to the observation chamber. The occupant of the chamber was in telephonic communication with the salvage vessel, and gave directions for the placing of explosive charges, and the working of grabs for removal of the blasted debris and the treasure itself.

In both the metal armour and the observation chamber, oxygen supplies and CO_2 absorbing apparatus are carried as the air purifying media.

Various forms of self-contained diving apparatus have been finding favour in many cases, particularly for use in ships, etc. All these are a development of the first practicable apparatus of the kind, as designed by H. A. Fleuss in 1878. The latest improved forms are adapted to the ordinary diving dress and helmet, and to several modified forms of dress. It is also used in the Davis Submarine Escape Apparatus.

For further information on the subjects of 'Diving' and breathing in poisonous atmospheres, see Sir Robert H. Davis's books 'Deep Diving and Submarine Operations' and 'Breathing in Irrespirable Atmospheres,' in which the subjects are dealt with fully.

The following table indicates the ordinary time limits in deep water, stoppages in ascent, and approximate air supply :—

Depth.		Pressure, Lbs. per Sq. In.	Time under Water from Surface to Beginning of Ascent	Stoppages in Minutes at different Depths.			Total Time of Ascent, Minutes.	No. of Cylinders needed.	Revolutions of Pump per min.
Feet.	Fathoms.			30 ft.	20 ft.	10 ft.			
0–33	0–5½	0–15	No limit . .		—		0–1	1	15–30
33–42	5½–7	15–18½	Up to 3 hours .	—			1–1½	2	15–20
			Over 3 hours . .			5	6	2	15–20
42–48	7–8	18½–21	Up to 1 hour . .		—	—	1½	2	20
			1 to 3 hours . .			5	6½	2	20
			Over 3 hours . .		—	10	11½	2	20
48–54	8–9	21–24	Up to ½ hour . .	—	—	—	2	2	20
			½ to 1½ hours . .	—	—	5	7	2	20
			1½ to 3 hours . .	—	—	10	12	2	20
			Over 3 hours . .	—	—	20	22	2	20
54–60	9–10	24–26½	Up to 20 min. .	—	—	—	2	2	25
			20 min. to ¾ hour .	—		5	7	2	25
			¾ hour to 1½ hours.	—	—	10	12	2	25
			1½ hours to 3 hours		5	15	22	2	25
			Over 3 hours . .		10	20	32	2	25
60–66	10–11	26½–29	Up to 15 min. .	—	—		2	2	25
			¼ to ½ hour . .		—	5	7	2	25
			½ to 1 hour . .	—	3	10	15	2	25
			1 to 2 hours . .	—	5	15	22	2	25
			2 to 3 hours . .	—	10	20	32	2	25
66–72	11–12	29½–32	Up to 15 min. .	—	—	2	4	2	25
			¼ to ½ hour . .	—	3	5	10	2	25
			½ to 1 hour . .		5	12	19	2	25
			1 to 2 hours . .	—	10	20	32	2	25
72–78	12–13	32–34½	Up to 20 min. .		—	5	7	2	25
			20 to 45 min. . .	—	5	15	22	2	25
			¾ to 1½ hours . .	—	10	20	32	2	25
78–84	13–14	34½–37	Up to 20 min. .	—	—	5	7	2	30
			20 to 45 min. . .	—	5	15	22	2	30
			¾ to 1½ hours . .	—	10	20	32	2	30
84–90	14–15	37–40	Up to 20 min. .	—	3	5	10	2	30
			20 to 40 min. . .	—	5	15	22	2	30
			40 to 60 min. . .	3	10	15	30	2	30
90–96	15–16	40–42½	Up to 20 min. .		3	5	11	2	30
			20 to 30 min. . .	—	5	11	18	2	30
			30 to 35 min. . .	—	5	15	22	2	30

FLOATING LANDING STAGES.

The opening of the Tilbury Floating Landing Stage in May 1930 brought into prominence the value of these facilities for passenger traffic at ports where there is an appreciable range of tide.

The Tilbury Landing Stage has a length of 1,142 ft., a width of 80 ft., and is secured to the left bank of the River Thames by means of four hinged steel booms and heavy mooring cables. It comprises a Jarrah timber deck carried on steel girders resting across sixty-three floating pontoons which rise and fall with the tide through a range of about 20 ft. At each end are two massive dolphins of timber pilework.

The Liverpool Landing Stage near the mouth of the River Mersey has been in service for many years. It is 2,534 ft. long by 80 ft. wide and has ten bridges connecting it with the shore, while, in addition, a floating approach bridge 550 ft. long, also carried on pontoons, provides an easy incline for vehicular traffic at all stages of the tide, which during springs attains a range of 30 ft.

Passenger landing stages for overseas traffic are provided with facilities for the handling and examination of baggage, waiting rooms, shelters, and other accommodation according to circumstances.

CHANNEL DEMARCATION.

(Contributed by J. P. Bowen, C.B.E., B.Sc., M.I.C.E., Engineer-in-Chief, Trinity House.)

Entrance channels to ports are outlined by buoys and lighted beacons on the river banks. Important changes in direction are also indicated by lightships and harbour entrances by minor lights.

Uniform System of Buoyage.

The general Lighthouse Authorities of the United Kingdom have adopted the following uniform system of buoyage.

Conical Buoys (those with pointed tops) denote the starboard hand, or that side which would be on the right hand of a mariner, either going with the main stream of flood tide, or entering a harbour, river or estuary from seaward.

Can Buoys (those with flat tops) denote the port hand, being the side on the left hand of the mariner under the circumstances described.

Spherical Buoys, with domed tops, mark the ends of middle grounds.

Pillar Buoys, having a tall central structure on a broad base, mark special positions.

Colour.—Starboard-hand buoys are always to be painted in one colour only ; port-hand buoys are to be painted another characteristic colour, either single or particolour. Spherical buoys are to be distinguished by white horizontal stripes. Wreck buoys are to be coloured green with the word 'wreck' in white letters across.

Types of Buoys.

Buoy Construction.—Buoys are generally hollow structures of steel or iron plating, at least ¼ in. thick. They should be constructed in two watertight sections to prevent sinking in case of penetration of one part.

The size and weight of buoys is as follows :—

		Diam. Ft.	Weight. Tons.
Trinity House Day Mark Buoys	First class	12	1
	Second class	10	2½
	Third class	8	1¾
	Fourth class	6	¾
	Fifth class	5	
Gas Buoys		10	4 to 9.

Gas buoys, in which gas is stored under pressure, should be made of welded iron or mild steel, not riveted, as under high pressures the riveted joints tend to leak and are not satisfactory.

Buoys carrying independent gas cylinders can be of either welded or riveted construction.

Buoy Moorings.—Mooring chains for large diameter and heavy buoys are about 1¼ ins. diameter, attached to sinkers weighing about 50 cwt. For buoys weighing 3 to 4 tons the chains are usually 1⅜ ins. diameter with 40 cwt. sinkers. For smaller buoys the chain is proportionately reduced. The length of a mooring chain is usually about 2 to 2½ times the depth of the water in which the buoy is laid.

Bell Buoys are provided with a bell from 3 to 6 cwts. in weight struck under the action of the sea by pendant clappers, or by a hammer actuated by carbonic acid gas.

Whistling Buoys emit a sound by means of air imprisoned in a long central tube during the period of rising in a swell and expelled through a whistle during the period of falling. The tube descends into the water to a considerable depth—about 15 to 20 ft.

Luminous Buoys.

Luminous Buoys are lighted by means of either oil gas, Blau gas, or acetylene. On account of the high pressure required for compact storage, coal gas is unsuitable for use as a naked flame, the luminosity of the gas being rapidly reduced by pressure. The ratio of calorific value is approximately as follows :—Coal gas, 5 ; oil gas, 10 ; Blau gas and acetylene, 14.

Oil gas, as manufactured on the *Pintsch* system, is produced at a cost of about 30*s.* per 1,000 cubic feet, about 75 cubic feet of gas being obtained from one gallon of oil. The average burner of a fixed light consumes about 0·75 cubic feet per hour. The gas is stored under pressure, generally of about 90 to 105 lbs. per sq. in., but reaching up to 150 lbs. A reservoir of 380 cubic feet in a buoy of 10 ft. diameter, charged to 75 lb. pressure would hold a sufficient supply of gas to maintain a 25-litre burner showing equal periods of light and darkness for about four months.

Blau gas is derived from the decomposition at a lower temperature of the raw or residuary products from which ordinary oil gas is extracted, and is richer than the latter in heavy carbides of hydrogen. It is liquefied for storage in bottles. Before use, the gas requires to be expanded in an intermediate receiver. The cost of production of Blau gas is considerably higher than that of oil gas, but the cost of transport is much less, and the increased heat of the flame gives an increase of 30 per cent. over oil gas to the intrinsic brilliancy of the mantle and the intensity of the lens.

Acetylene, a gaseous compound of hydrogen and carbon, is produced by the access of water to calcium carbide. From 1 lb. of commercial calcium carbide can be produced about 4½ cubic feet of acetylene.

Acetylene gas produced as above is not chemically pure, and gives rise to deposit on the burners, but this difficulty can be overcome by the addition of a suitable purifier between the gas-generating chamber and the burner. Unattended lights working on this system will run for six months, or even a year, without recharging.

Dissolved Acetylene.—The compression of acetylene for storage purposes is generally illegal, on account of the liability of the gas, so stored, to explode. On the other hand, compressed acetylene in what is known as the dissolved form is perfectly safe, and cylinders of dissolved acetylene, prepared in accordance with official regulations, are classed by the Home Office as non-explosive. Acetylene in this form is now most commonly used both for buoys and unattended lights, because of its convenient storage and transportation.

The solvent commonly employed is acetone, which is a liquid hydro-carbon, possessing the property of absorbing about 25 times its own volume of acetylene at atmospheric pressure, and multiples of this to correspond with the number of atmospheres to which compression is carried.

The steel cylinders for the reception of the dissolved acetylene are first completely filled with a substance having a porosity of 80 per cent., which is then caused to absorb as much acetone as would occupy 40 per cent. of the volume of the cylinder. The actual gas capacity of the cylinder is thus 40 per cent. of 25, or 10 volumes per atmosphere of pressure.

Dissolved acetylene cylinders should not be discharged below the atmosphere gauge pressure, because the rate at which the acetone is carried off by the gas greatly increases as the pressure approaches zero. Acetone vapour has no detrimental effect upon the consumption of the gas in the burner. It is generally found necessary to make up the quantity of acetone, after five to ten refillings of the cylinder with acetylene.

Economies in the consumption of dissolved acetylene have been effected by the introduction of mechanism for emitting flashing or occulting lights invented by Mr. Gustav Dalen and others whereby a wide range of intermittent light signals can be produced automatically.

To the same scientist is attributable the *Sun-valve*, or, more correctly, the *Light-valve*, which is a mechanical device sensitive to light and actuated entirely by the alternation of daylight and darkness. It opens the gas supply at nightfall and closes it at daybreak with perfect regularity, resulting in a saving of some 25 to 40 per cent. of the gas, according to climate, as compared with a constantly burning light. The use of the instrument is generally restricted to shore lights, or lights of considerable magnitude.

Mr. Dalen has further produced complete automatic apparatus for mixing in correct and constant proportions the dissolved acetylene and air and for burning the compound in conjunction with incandescent mantles, either as a constant or intermittent light. This apparatus, by reason of its nature, is more applicable to lights of considerable power, superior to those generally fitted to buoys.

Lenses.—Lenses are usually of 150 to 300 mm. (6 to 12 ins.) diameter. The candle power of lanterns fitted with an oil-gas burner with incandescent mantle or a 25-litre acetylene burner and 200 mm. diameter lens, is about 200 candles, and with 300 mm. lenses about 300 candles.

The elevation of a luminous buoy given in fig. 32 (the Gas Accumulator Co. (United Kingdom), Ltd.) has been selected as an example of buoy design with an unusually varied range of application. It has been designed primarily for use as a spare or wreck-marking buoy.

The lantern, which can be of any size up to that carrying a 300-mm. lens, is mounted upon a trestle superstructure. In this way it is possible to place any one of the three conventional types of daymark in position, namely, can, cone, or spherical, without in any way interfering with the light apparatus. The lantern carries a special type of flashing mechanism which is so designed that any one of three characters can be exhibited at will by simply turning a valve on the burner stem. The characters are single, double, or triple flashing.

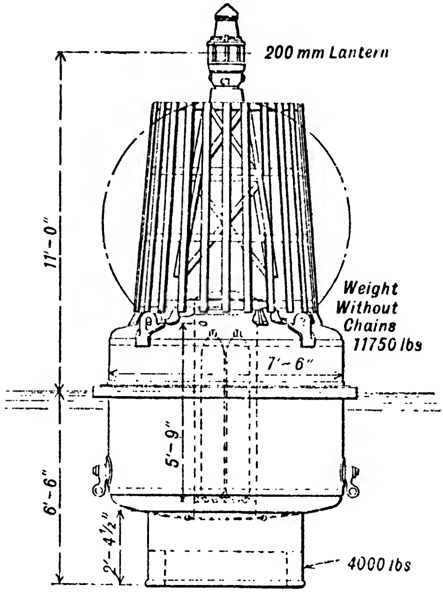


FIG. 32.—Luminous Buoy.

Two accumulators are the minimum number with which the buoy may be equipped and retain its calculated stability, but, if necessary, the number can be increased to four, and the unattended service period can be as much as twelve months.

The candle-power depends upon the size of lens and burner used. With a 300-mm. lantern and 15-litre burner this would amount to 180 s.c.p. The coloration of the light can be varied at will by the insertion of suitably tinted glass shades inside the lens and surrounding the flame. Further reference to the use of acetylene lighting will be found on page 651.

Lightships.

Lightships afford a more powerful light than luminous buoys, and the focal plane is at much higher level—generally about 35 to 40 ft. above the water surface. They are more costly both in first outlay and maintenance, and are usually manned by a crew, though this is not essential in all cases.

Small lightships, without constant attendance, are stationed on the coast of this country and of France. A vessel of this type is about 40 ft. long, 12 ft. beam, and 8 ft. deep, fitted with a storeholder containing 2,500 cubic ft. of gas, which is sufficient to maintain a light for 3 months. Many unattended light vessels in all parts of the world are equipped with dissolved acetylene lights and run without recharging for six or twelve months.

Larger vessels, with crews, range from 60 to 150 ft. in length, from 20 to 30 ft. in breadth, and from 10 to 20 ft. in depth, the draught being from 7 to 10 ft.

Lenses vary from 250 mm. to 500 mm. focal length and the candle-power from 2,000 to 500,000. Catoptric reflectors with a candle-power of 4,000 are used in the oldest types of light-vessels. Some modern light-vessels are being equipped with electric light, and fig. 33 shows a typical example of the flashing optic and pendulum revolved by an electric motor. The illuminant is an incandescent electric filament lamp and the lens is of small Third Order size (375 mm. focal length). The whole of the optical apparatus is swung on gimbals, so tha the beam is kept practically horizontal, and is entirely of Trinity House design.

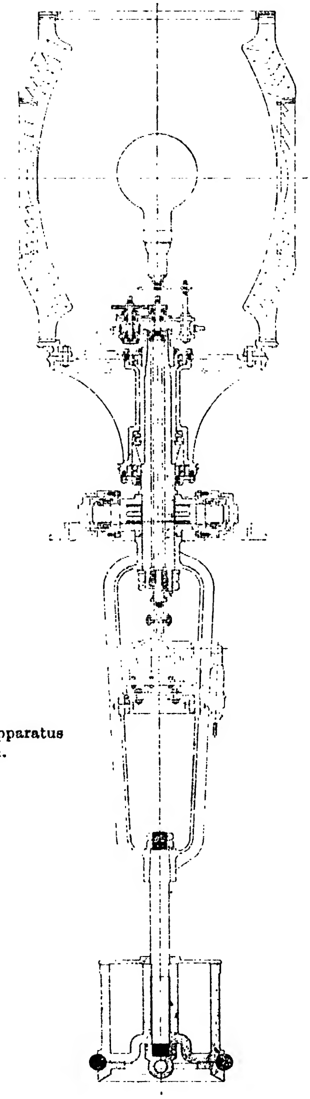


FIG. 33.
Lightship Dioptric Apparatus
and Pendulum.

LIGHTHOUSES.

Classification of Lighthouse Apparatus.

Order.	Focal distance, mm.	Order.	Focal distance, mm.
Hyper-radial	1,330	Small third order	375
Meso-radial	1,125	Fourth order	250
First order	920	Fifth "	187·5
Second "	700	Sixth "	150
Third "	500		

The Eddystone Lighthouse.

The type of foundations and general construction of rock lighthouses is well exemplified by the new Eddystone Lighthouse. In this structure the tendency of the curvilinear outline of Smeaton's and of other similar sea-towers that have succeeded it, to elevate the centre of force of each wave-stroke on the structure, induced Sir James Douglas to adopt a cylindrical base, from which base, at a level of 2½ feet above high-water spring-tides, the curved shaft of the tower commences. The difference in the rise of heavy seas on the two structures during stormy weather is remarkable. The cylindrical base has the further advantage of affording a convenient landing platform, thus adding considerably to the opportunities of relieving the lighthouse.

The base is 44 feet in diameter by 22 feet in height. The tower is a concave elliptic frustum, the generating curve having a semi-transverse axis of 173 feet, and a semi-conjugate axis of 37 feet. With the exception of the space occupied by the fresh-water tanks, the tower is solid for 25 ft. 6 ins. above high-water spring-tide level. At the top of the solid portion the wall is 8 ft. 6 ins. in thickness, diminishing to 2 ft. 3 ins. in the thinnest part of the service-room. All the stones are dovetailed both horizontally and vertically. The system consists in having a raised dovetailed band 3 inches in height on the top bed and on end joint of each stone. A corresponding dovetailed recess is cut in the bottom bed and end joint of the adjoining stones, with just sufficient clearance for the raised band to enter it freely in setting. From experiments made with blocks put together in this manner with Portland cement, it is found that the work is so homogeneous as to be as nearly as possible as strong as the solid granite.

Each stone of the foundation-courses (figs. 34, 35) is sunk to a depth of not less, at any part, than 1 foot below the surface of the surrounding rock, and is further secured by two Muntz metal bolts, 1½ inch in diameter, passing through the stone, and 9 inches into the rock below, the top and bottom of each bolt being fox-wedged.

The masonry consists of 2,171 stones, containing 62,133 cubic feet of granite, or 4,688 tons.

Cost of Construction of Various Lighthouses.

Name of Structure.	Total Cost.			Cubic Feet.	Cost per Cubic Foot.		
	£	s.	d.		£	s.	d.
Eddystone (Smeaton)	40,000	0	0	13,343	2	19	11½
Bell Rock	55,619	12	1	28,530	1	19	0
Skerryvore	72,200	11	6	58,580	1	4	7¾
Bishop Rock	34,559	18	9	35,209	0	19	7½
Smalls	50,124	11	8	46,386	1	1	7¼
Hanois	25,296	0	0	24,542	1	0	7¼
Wolf Rock	62,726	0	0	59,070	1	1	3
Dhu Heartach	72,584	9	7	42,050	1	14	6
Longships	43,869	8	11	47,610	0	18	5
Eddystone (New)	59,255	0	0	65,198	0	18	2

Different Descriptions of Lights.

(1) *Fixed.*—The light being continuously visible.

(2) *Occulting.*—A light regularly eclipsed in which the duration of light is greater than the duration of darkness.

(3) *Group occulting.*—An occulting light with two or more occultations in quick succession.

(4) *Flashing.*—A light regularly eclipsed in which the duration of light is shorter than the duration of darkness.

(5) *Group Flashing.*—A flashing light with two or more flashes in quick succession.

(6) *Alternating Light.*—A light in which successive occultations or flashes are alternate colours, usually white and red.

BASE OF EDDYSTONE LIGHTHOUSE.

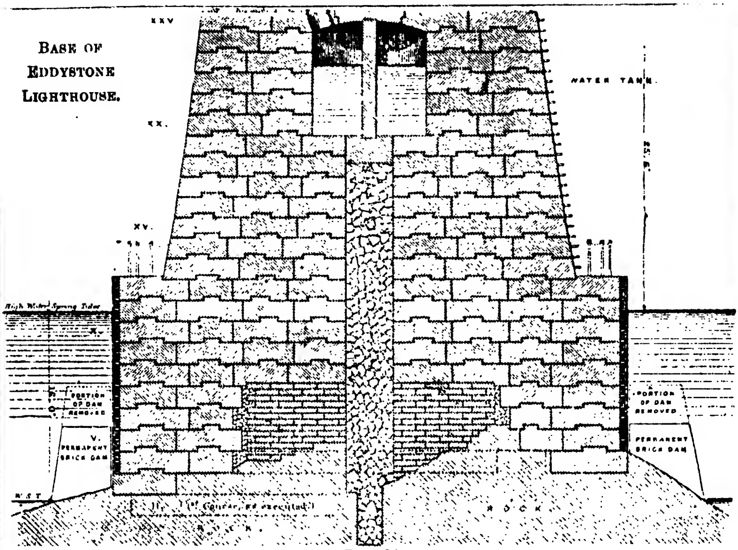


FIG. 34.

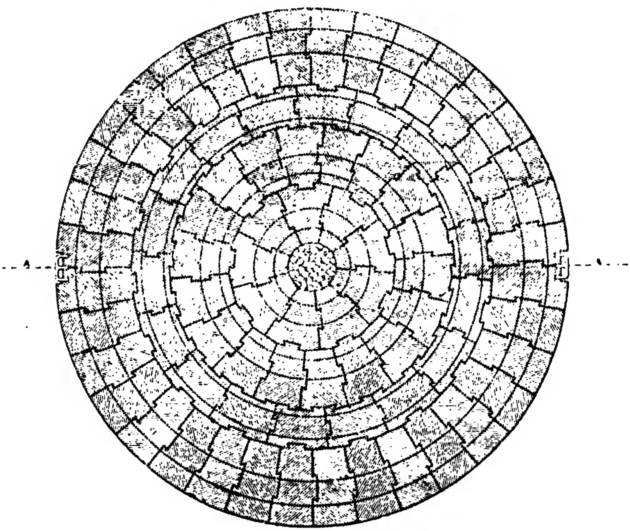


FIG. 35.

Distance visible from Lighthouses.

The following Table shows the distance of the horizon visible from towers of the heights given. In it the necessary correction has been made for refraction :—

H. Heights in Feet.	λ. Lengths in English Miles.	H. Heights in Feet.	λ. Lengths in English Miles.	H. Heights in Feet.	λ. Lengths in English Miles.
5	2·958	80	11·832	250	20·916
20	5·916	90	12·549	300	22·912
30	7·245	100	13·228	350	24·748
40	8·366	120	14·490	400	26·457
50	9·354	140	15·652	600	32·403
60	10·246	150	17·201	800	37·416
70	11·067	200	18·708	1,000	41·833

Optical Apparatus.

The essential part of a lighthouse is the optical apparatus, which is formed of glass lenses and reflecting prisms in order to concentrate the light from the burner in the required direction. The optical apparatus of a fixed light concentrates the rays in the vertical plane only, and sends out a zone of light which illuminates the horizon over the whole 360° or any less angle which may be required. That of a flashing light concentrates the rays in both the vertical and horizontal planes, and thus throws out a series of beams, the number varying with the character which the light is required to give. The whole apparatus is revolved, and by this means the beams sweep round the horizon and appear as flashes at definite intervals to the mariner.

Large lights are supported on a mercury float whereby friction of movement is reduced to a minimum ; ball bearings are satisfactory for lights up to small third order. The former enables the largest type of optical apparatus, the weight of which runs up to 10 tons, to be rotated, with the minimum friction.

It is usual to arrange the optical system so that the flashes have a minimum duration of 0·2 second ; it has been found that a shorter duration is not sufficient for the mariner to detect the flash in hazy weather. Fig. 36 illustrates an optical apparatus of 700 mm. focal distance and 500,000 candle-power, mounted on a mercury float pedestal, and also the lantern which protects it. The lens is shown with a system of prisms outside, to throw upwards a beam of 11,000 candles for the use of aircraft.

Large glass catoptric mirrors with a single light source as developed by Mr. Stevenson for the Poward Lighthouse on the Clyde, are now employed at certain lighthouses.

Sources of Lighthouse Illumination.

Where an electric supply is not available paraffin oil and dissolved acetylene gas are the most suitable and economic illuminants.

Oil and gas lights, when shown through similar lenses, are equally affected by atmospheric variations. The electric light is absorbed more largely by haze and fog, but in all weathers and at all distances its penetration has proved superior.

The luminary now in general use at large lighthouses is the petroleum vapour burner, to which paraffin oil is supplied under pressure, and is vaporised in a retort beneath the burner, being afterwards consumed in an incandescent mantle. The sizes of mantle usually employed vary from 35 mm. to 110 mm. in diameter with intensities up to 3,500 candles.

A modern burner of this type is the ' Hood ' petroleum vapour burner designed by the late engineer-in-chief to the Trinity House ; the mantle employed is of the autoform type, with an intrinsic brightness of 325 candles per sq. in. The burner is in five sizes, 25 mm., 35 mm., 50 mm., 75 mm., and 100 mm., with total intensities of 400, 660, 1,150, 2,200, and 3,300 candles, and consumptions of 0·50, 0·75, 1·25, 2·25, and 3·25 pints of oil per hour, respectively.

At the Eddystone, which is a biform light having superimposed lenses and burners, the resultant beam with a 35 mm. ' Hood ' burner in each tier, is nearly 600,000 candles and the consumption 700 gallons per annum.

The development in high-powered electric incandescent filament lamps has made possible their use for lighthouse illumination. Lamps up to 10 kw. are now employed and, combined with automatic apparatus, effect economy in maintenance by the reduction of personnel.

For minor lighthouses and portlights at the entrances to harbours, etc., the illuminant is sometimes a capillary wick lamp having one or more wicks, which can be left without attention for a single night. A distinctive character is given to such lights by means of revolving screens driven by a simple clockwork mechanism which will run for a night without rewinding ; the screen occults the light at regular intervals. For the entrance to harbours where range is not a great consideration the light may be coloured red or green. Where electricity is available electric lamps are employed and a small motor used for revolving the screens. Sometimes an electric flashing mechanism is used to give the character.

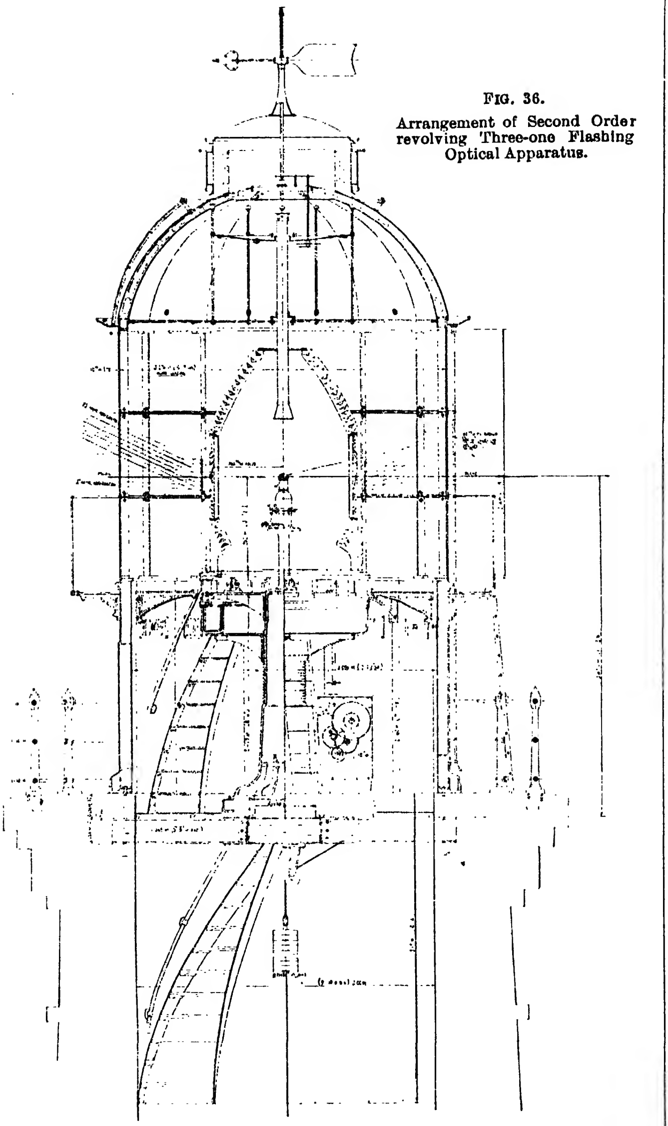


FIG. 36.

Arrangement of Second Order revolving Three-one Flashing Optical Apparatus.

Many lights are now left unattended for long periods, and the illuminant is generally compressed oil-gas, stored in receivers, or acetylene gas, manufactured in generators on the spot or on the dissolved acetylene principle; or electricity.

BAS SERANI LIGHTHOUSE, MOMBASA.

The following description of a modernised lighting equipment has been furnished by the Gas Accumulator Co. (United Kingdom), Ltd.:—

The lighthouse was originally equipped with an oil-burning, clock-driven rotating lens installation attended by light keepers.

In 1928 it was converted by the company, the light source being a 500 watt electric lamp mounted in a catadioptric section of a drum lens having an arc of illumination of 180°. The focal distance of the lens is 300 mm.

Electric current is derived from the town lighting mains, at a pressure of 240 volts, the supply being 50 periods, single-phase.

In order to render the apparatus automatic in operation and cover lamp filament and supply current failures a burner changer mechanism is installed as shown in fig. 38, p. 655. Two electric lamps are fitted, one being in the focal points of the lens, the other being carried on an arm at 120° to it; a third arm, again at 120°, carries an acetylene burner, the whole being arranged to rotate about a vertical axis. Should the filament of the lamp in service break, the cessation of the flow of filament current is utilised to operate a release gear, whereby the second bulb is brought into the focus and the defective one removed. As the new lamp comes into position, it lights up and thereby stops the action of the changing mechanism.

Should a failure of the electric supply occur due to a breakdown at the power station or for any other reason, the acetylene burner is automatically brought into focus. On renewal of the electric supply an electric lamp is replaced at the focus.

The original character of the old light of 1-in. flash plus 4-in. eclipse has been retained in the new apparatus, both electric and acetylene burners being arranged to emit a flashing light of this character. The lighting and extinguishing of the light is under the control of a Venner clock mechanism.

The candle-power developed with an electric lamp at the focus amounts to 22,000 s.c.p.; and with a focal height of 85 ft. above sea-level, the light has a geographical range of 18 nautical miles in clear weather, taking the average height of a ship's bridge as 40 ft.

The dissolved acetylene gas for supplying the standby light is provided by two gas accumulators located in the base of the tower. These supply an open type burner composed of four twin jet burners each of 25-litre capacity. The light is under the care of a member of the port staff, whose duty it is to visit the light for inspection purposes at fortnightly intervals.

Relative Absorptive Powers of Coloured Glass.

With the ruby glass the ratio between the powers of the red and white lights is as 2 to 5, and with green glass 1 to 5.

When colour is necessary in dioptric apparatus of fixed section the coloured beam can be strengthened by condensing prisms, whilst for revolving dioptric apparatus the red and white can be approximately equalised by giving to the red beam a greater area and by a special arrangement of the lens and prisms.

During thick weather red lights are not so much absorbed by the atmosphere as white.

Glazing.

The glass of lighthouses is usually ⅜ in. thick, except in peculiarly exposed situations, where it is ½ inch.

The astragals are 1 in. in section, of gun-metal, having a tensile strength of 33,000 lbs. per square inch.

Diagonal and helical astragals do not intercept light in any one azimuth throughout their whole height, and they secure great rigidity and strength.

Standard first-order lanterns are 14 ft. in diameter and 10 ft. high in the glazing.

The Bishop **Lighthouse** has a lantern of 14 ft. diameter and 15 ft. high; Fair Island, 14 ft. diameter, 6 ft. 2 ins. high; **Tory Island**, 14 ft. 9 ins. diameter, 18 ft. high; **Sule** Skerry, 16 ft. diameter, 12 ft. 2 ins. **high.**

Lightning Conductors.

These should be of ⅝-in. best quality copper rod, carried 18 ins. above highest part of ventilator of lantern, and then terminated in three branches with platinum caps or platinum wire No. 8 B.W.G., screwed and soldered into the rod; the latter should be put into metallic connection with all internal masses of metal. The earth connections should be made by copper plates, 20 ins. by 12 ins., and ½ in. thick. Conductors should be periodically inspected.

Audible Signals.

The types of sound signals in use by the Trinity House are:—(1) aerial and submarine bells; (2) explosive signals; (3) reeds; (4) sirens; (5) diaphones; and (6) electric oscillators.

Of the first of these, Aerial Bells, are actuated by the motion of the ship or buoy carrying them, or mechanically operated as the unattended lighthouse on the Menai Straits and on the **Lune Deep** unattended gas boat.

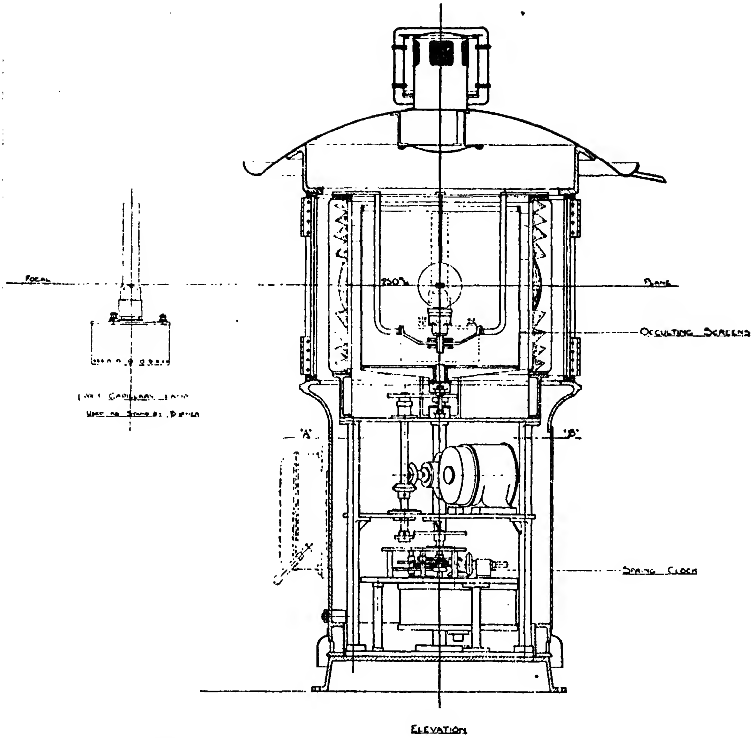
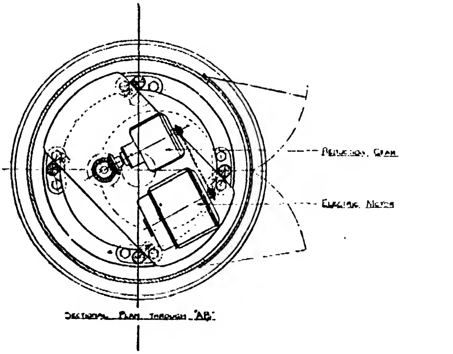


Fig. 37.

Fourth Order Occulting Portlight, with Alternative Drive of Spring Clock or Electric Motor for Screens.



Ras Serani Lighthouse, Mombasa.

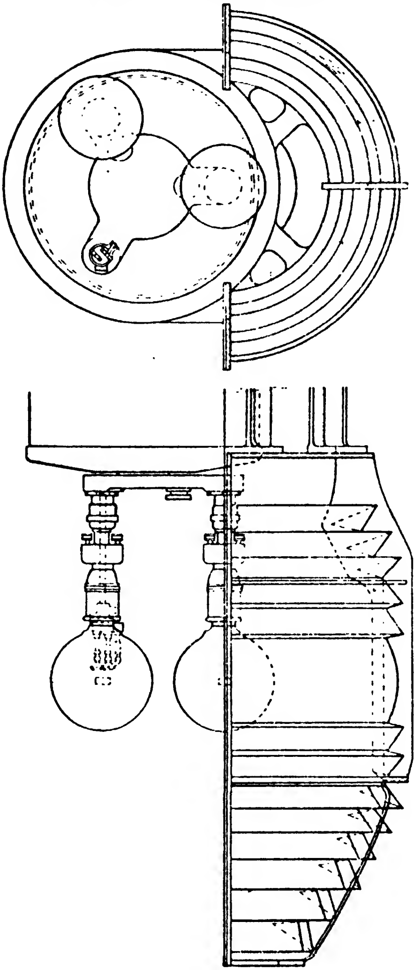


Fig. 38.—Modernised Lighting Equipment for alternative supplies of electricity or acetylene.

The bells in each case weigh 10 cwt. The striking hammer may be either interior or exterior to the bell, and the striking action is performed by the admission of carbonic acid gas under pressure to the underside of a diaphragm which actuates the hammer; the expended gas is exhausted into the atmosphere and the hammer returns preparatory to another impulse. A battery of 24 cylinders of CO_2 gas is capable of maintaining a bell signal with a character of 1 stroke every 15 seconds for twelve months without re-charging.

Submarine Bells are similarly operated by a regulated pressure of air or CO_2 gas, in order to impart a definite character to the signal.

The bells used on Trinity House light-vessels are suspended outboard from a davit, and when lowered are approximately 18 feet below the water line. They are operated by compressed air at a pressure of 15 lbs. per sq. in., which is led from a receiver on board the vessel through a rubber pipe to the striking mechanism actuating the bell hammer.

Explosive Signals consist of the firing of a gun, explosion of acetylene gas, or the detonation of gun-cotton charges from a rising and falling jib on the lantern roof, of which last system there are seven instances in the Trinity House services; a special safety gear is fitted by which premature firing is obviated while the jib is in its lowered position for loading. This is effected by leading the wires from the battery switch on the lantern pedestal to terminals outside the lantern roof, and fixing to the trunnion of the jib spring contact strips which make and break contact between the battery and detonator leads, when the jib is raised and lowered; by this means, after the jib has been moved through a small angle out of the vertical, no firing of the charge is possible until it is again raised, and consequently no danger to the keeper, or damage to the lantern glazing, can accrue.

Explosive signals are now sometimes controlled from a distance by wireless. A representative installation on the Clyde consists of a gun which is charged with an admixture of air and acetylene gas and is fitted with wireless receiving apparatus. A synchronous transmitting set is installed on a pier 1½ miles distant. When fog is observed, the transmitting installation is put into operation and the impulses sent out are received on the apparatus attached to the fog signal gun. The synchronising arrangement renders the installations immune from atmospherics and interference from other wireless waves.

The next three important types of sound signals will be dealt with conjointly, as each is operated by compressed air, and the only difference lies in the type of instrument producing the sound.

Reed Signals are, as a rule, sounded at a pressure of 15 lbs. per sq. in. from a vertically fixed conical horn with a bent top. The reed tongues are of steel 3 ins. long by ½ in. wide, and are tuned to give a note of about 150 vibrations per second, with a consumption of ⅜ cu. ft. of free air per second of blast.

Manual reed signals are sounded at 5 lbs. per sq. in., and the character is produced by a hand-operated valve.

Siren Signals are sounded at a pressure of 25 lbs. per sq. in. from horizontal tapered elliptical trumpets, and from vertical conical horns with bent tops, and the siren may either be automatic or motor driven. A note of about 187 vibrations per second is generally employed.

The siren at Trevose Head is 12 ins. in diameter, with only one port to the trumpet, and is driven by a small Brotherhood air motor. In this instance there is only one trumpet, which is a specially constructed Rayleigh elliptical steel horn.

'Diaphones' are sounded at a pressure of 30 lbs. per sq. in. and have a consumption up to 45 cu. ft. of free air per second of blast. The signal terminates with a distinctive grunt.

Electric oscillators consist of a diaphragm vibrated by an alternating electric current, and are used both for aerial and submarine signals.

Audibility.

Under favourable conditions a powerful siren or diaphone may be heard at a distance of 14 miles, but sometimes the range does not exceed 2 miles. A reed horn may have a range of 3 miles.

A compressed-air siren (0·125 metre in diameter) has been heard at a distance of 13 miles, but the range is sometimes as low as 1·75 mile.

A 4-oz. charge of cotton powder can be heard under favourable conditions at a distance of 20 miles, and under unfavourable conditions at a distance of 2½ miles.

The distance traversed by the sound from submarine bells has reached 15 miles, but 1 to 3 miles is a serviceable distance. The range of a submarine oscillator is about twice this distance.

Wireless Signalling.

A modern aid to navigation is the wireless fog signal, known as a radio-bearer, which enables the mariner to obtain bearings when lights are invisible and sound signals inaudible, the system of transmission being by means of valves for generating continuous or modulated continuous waves within a band of 938 to 1,034 metres. Some light vessels equipped with wireless beacons are also provided with a submarine sound signal, and the combination of the two is utilised for sending out at regular intervals synchronised signals for distance finding as well as for taking bearings.

Wireless telephones are installed at lighthouses and on light vessels at sea for communicating with shore stations.

SECTION XVIII

WATER ENGINEERING

PART I

RAINFALL—RUN-OFF — ABSORPTION AND EVAPORATION — FLOOD FLOWS (pp. 659–670)

(Contributed and revised by R. A. Ryves, M.Cons.E.)

PART II

THE FLOW OF WATER — FORMULAE AND DATA—WEIRS— SLUICES AND CHANNELS (pp. 671–682)

(Revised and mainly contributed by R. A. Ryves, M.Cons.E.)

PART III

EARTHEN AND MASONRY DAMS—TYPICAL AND NOTABLE WORKS — EXPERIMENTAL DETERMINATIONS — RECENT PRACTICE (pp. 683–700)

(Revised by R. A. Ryves, M.Cons.E.)

PART IV

IRRIGATION — AVAILABLE SUPPLY — EVAPORATION — ABSORPTION—EVAPORATION FROM SOILS—LOSSES FROM IRRIGATION CANALS — CRITICAL VELOCITY — DUTY OF WATER (pp. 701–717)

(Contributed by R. A. Ryves, M.Cons.E.)

PART V

THE DEVELOPMENT OF WATER-POWER (pp. 719–728)

(Revised by R. A. Ryves, M.Cons.E.)

PART VI

WATER SUPPLY (pp. 729–740)

(Revised by R. A. Ryves, M.Cons.E.)

SECTION XVIII

PART I

RAINFALL—RUN-OFF—ABSORPTION AND EVAPORATION—FLOOD FLOWS.

RAINFALL.

(Contributed and revised by R. A. Ryves, M.Cons.E.)

Rainfall in the British Isles.

For valuable data and studies relating to rainfall in the British Isles reference may be made to a paper on 'The Distribution of Mean and Extreme Annual Rainfall over the British Isles,' by H. R. Mill (*Min. Proceedings Civil Engineers*, vol. clv., 1903–1904); also to 'Variations in the English Climate during the Thirty Years 1881–1910,' by W. Marriott (*Quarterly Journal R. Met. Soc.*, July 1911). See also recent issues of *British Rainfall.*

The rainfall of each of the three divisions of the British Isles decreases from 100, 80, or 60 inches in the mountains and moorlands near the Western coasts to about 25 inches (over a few areas down to 20 inches) near the Eastern coasts of Scotland and England, and in the Midlands, and to about 35 inches on the East coast of Ireland. Over more than a third of England the rainfall ranges from 25 to 30 inches, and over a considerable total of smaller areas the fall is from 30 to 40 inches. There is a similar range, 30 to 40 inches, over about half Scotland, and a large area in the Western Highlands has a rainfall ranging from 60 to 100 inches. Over the Eastern half of Ireland the fall is from 35 to 40 inches, and over most of the Western half from 40 to 50 inches. For the period 1870–1899 the average annual falls were as follows: England 31·62; Wales, 49·53; Scotland, 46·85; Ireland, 42·28; British Isles, 39·25. For the period 1881–1910 the average for England was 31·90.

Average, Maximum and Minimum Rainfalls.

It is generally agreed that a period of 30 to 35 years (*Binnie*) is long enough to establish for the average annual rainfall a value which will be nearly the true average, if such there be, and anyhow will be the mean of probabilities.

In the British Isles for the period 1870–1899 the rainfall of the 'wettest' year exceeded the average fall by 43 per cent., and that for the 'driest' year was 29 per cent. below the average (*Mill*). The following, expressed in percentages of the averages, are the figures for the different parts of the British Isles and for the whole area, taking in every case the wettest and driest years for the area referred to, period 1870–1899:—

England, 146 and 70; Wales, 147 and 75; Scotland, 147 and 72; Ireland, 133 and 69; British Isles, 143 and 70.

For the period 1881–1910 (*Marriott*) the percentage figures for England are 131 and 76·7. This period seems, however, to have been somewhat abnormal. The actual rainfalls compare thus:—

England.	Maximum.	Average.	Minimum.
Period 1870–1899	46·98	31·62	22·83
Period 1881–1899	41·78	31·90	24·45

This suggests that although the 30 years' period may be satisfactory as regards the average annual rainfall it may not include a fall approximating to the maximum, nor one approximating to the minimum.

For the period 1881–1915 the results (inches) are as follows:—

	England and Wales.	Scotland.	Ireland.	British Isles.
October–March . . .	19·61	29·2	24·28	23·42
April–September . . .	15·62	21·12	19.02	17·99
Year	35·23	50·32	43·30	41·41

At stations in the British Isles the following annual rainfalls in percentages of the average have been recorded: 58·8, 59·8, 61, 60·5, 63·6, and 64·5, showing defects from 35½ to 41 per cent. of the averages.

Mr. John Glasspoole, from 40 stations in the British Isles, for the 58 years, 1870–1927, furnishes a diagram showing the means of extremes for any period up to 12 months: for four months, min. 11, max. 71; for 6 months, min. 23, max. 95; for 8 months, min. 36, max. 114; for 12 months, min. 63, max. 154; these percentages based on the average rainfalls at the individual stations.

The Driest Group of Three Years Running.

Experience has shown that the most serious shortage in water supplies usually occurs as the result of the defect of rainfall in the driest group of three years running, and this consideration is usually one of the factors in the design of water supplies and irrigation works. The principle is to be accepted in all cases, inasmuch as there is, for every place, some period which is that towards the end of which the effects of a drought will probably be most severely felt.

The estimated average annual rainfall for the three years' period is sometimes put at 75 per cent. of the general average. At Cape Town the average rainfall for the driest three years is 73 per cent. of the general average. In any case it must be remembered that the available run-off in the three years may be a much smaller percentage of the average available run-off. If, out of 50 inches of rain, 15 are lost by evaporation and absorption, we have 35 inches available. With a 40 inch, or 80 per cent. fall, we should have 25 inches available, or about 71 per cent., and the loss in the dry year might be greater than the average.

England and Wales: 1932–35.

The following data are from a paper (*Inst. Water Eng.*, December 1935), by Dr. J. Glasspoole.

The dry period, 1932–35, was the fifth since 1727. The mean annual rainfall changed from 39 ins., in the preceding wet period, to 31 ins. in 1933–34. Beginning from November 1932, the 5 months' rainfall in the extreme South-East of England and in Northumberland was less than 70 per cent. of the local averages. In the 11 months, to September 1933, the rainfall was only two-thirds of the area averages, in South-West Wales, the neighbourhood of the Cheshire Plain, from Northamptonshire to Kent, and in the south of Northumberland. In the 17 months' period from November, 1932, the rainfall was below average over a long area between London, Oxford and Wellingborough, as well as in South Wales and Southport; in London, 64 per cent. of the average. In that period, the drought during the 12 months ending March 1934, was unprecedented. In the 23 months' period, the rainfall was below average over the whole of England and Wales. In October 1934, the variation was from 150 per cent. of the average over North Wales and North-West England to 50 per cent. over South-East England; in November, 25 per cent. in North-West England and 125 per cent. along the coasts of Northumberland and Sussex. For the 25 months' period, the rainfall was little more than two-thirds the average over an area between Wellingborough and London, also near Cardiff. In the 3 years ending September 1935, the rainfall was 85 per cent. of the average over a large area stretching from the southern half of Wales across central England to London and the Wash, the smallest rainfall being 81 per cent. at Wellingborough. In the Thames Valley above Teddington, the 25 months fall from November 1932, was 12·5 ins., the average being 28·2 ins.

Variations of Annual Rainfall.—Percentages in successive years, 1910 to 1936, taking the average for 1881-1915 as 100. Year '10 (1910), 113; '11, 93; '12, 125; '13, 98; '14, 108; '15, 110; '16, 114; '17, 98; '18, 107; '19, 105; '20, 109; '21, 69; '22, 105; '23, 113; '24, 120; '25, 106; '26, 102; '27, 123; '28, 115; '29, 100; '30, 117; '31, 109; '32, 103; '33, 81; '34, 95; '35, 113; '36, 109. In those 27 years, therefore, only 6 yielded less than the standard average, while 20 years yielded more. Although including the drought of 1933–34, the 10 years, 1927–36, yielded 6 per cent. above the average. Since 1891 we have never had more than two years dry in succession; but the percentages of the standard average in the 5 years 1854–58 were 77, 87, 94, 97 and 81. (E. G. Bilham.)

Altitude and Rainfall.

Rainfall in Scotland.—Altitudes and average annual rainfalls, 1881–1915. *East Scotland* Edinburgh, 441 ft., 26·3 in.; Marchmont, 498 ft., 32·2 in.; Loch Leven, 360 ft., 35·8 in. *North-east Scotland:* Aberdeen, 79 ft., 29·5 in.; Ardross Castle, 450 ft., 38·5 in.; Gordon Castle, 104 ft., 29·8 in.; *West Scotland:* Rothesay, 200 ft., 49·0 in.; Toch Thom, 643 ft.; 65·4 in.; Poltalloch, 135 ft., 52·7 in. *North-west Scotland:* Stornoway, 79 ft., 49·9 in.; Applecross, 70 ft., 57·3 in. (A. H. R. Goldie and H. E. Carter, Sec. Lit.)

Heavy Station Rainfalls in the British Isles.

Inches in 24 hours: Bruton, Somerset, 1917, 9·56; Cannington, Somerset, 1924, 9·4; Loch Quoich, Inverness-shire, 8·40; Seathwaite, 8·03 and 7·00; Ben Nevis, 7·74 and 7·29; Rosthwaite, 6·94; Angerton Hall, 6·70; Lyme Regis (' in the night ' July), 3·58 ins.; Ballycumber, 3·8 ins. For shorter periods:—Norwich, 0·52 in. in about 5 mins.; Dover, 0·44 in. in 12 mins.; Margate, 0·51 in. in 20 mins.; Hellingley, Sussex, 1·21 ins. in 50 mins., and 0·73 in. in 12 mins.; Oxted, 2·14 ins. in 24 hours.; East Grinstead, 1·80 ins. in 1¼ hrs.

Banstead, 3·59 ins. in about 90 mins.; Wallington, 2·77 ins. in 65 mins., and 3·50 ins in 60 mins.; Wimbledon, 1·58 in. in 80 mins.; Croydon, 1·85 in. in 2 hours 48 mins., and 0·8 in. in 15 mins.; Ponders End, 1·05 in. in 19 mins.; Birmingham, 1·04 in. in 27 mins.; Burnham, 1·73 in. in 40 mins.; a fall of 1·2 in. in 15 mins. seems to be the quarter-hour record for the British Isles.

Sheffield, August 4, 1922, 2·45 ins. in 18 hours, 1·04 in. in 2¼ hours, and nearly 1·0 in. in 30 mins. Great Britain, February 11, 1927: at 10 stations the falls exceeded 2 ins. in 1 hour; Balham High Road, 5 ins. in 1 hour (report of L.C.C. main drainage committee).

Heavy Single Falls of Rain over Considerable Areas in the British Isles.

London, whole area, 2·5 ins. in 24 hours.
Lake District, 714 sq. miles, 5·63 ins. } single fall in less than 3 days
Eastern Ireland, 1,160 sq. miles, 4·76 ins. } single fall in less than 3 days

Aug. 1912	East Anglia	18 sq. miles, over	8 ins. in about	24 hours
	,, ,,	238 ,,	7 ,, ,,	24 ,,
	,, ,,	705 ,,	6 ,, ,,	24 ,,
	,, ,,	1,039 ,,	5 ,, ,,	24 ,,
	,, ,,	3,463 ,,	3 ,, ,,	24 ,,

Before 1912 the maximum fall for 24 hours recorded in Norfolk was 4·48 ins.

In the South of England storm of June 1917 the falls and areas covered were: more than 9 ins., 2 sq. miles; 8 ins., 13 sq. miles; 7 ins., 29 sq. miles; 6 ins., 85 sq. miles; 5 ins., 288 sq. miles; 4 ins., 809 sq. miles. The storm lasted from 2¼ to 20 hours, and is classed as a 16-hour storm.

Heavy Station Rainfalls, Canada and United States.

Toronto, 0·5 in. in 10 mins.; and 0·6 in. in 15 mins.; New York, 1·2 in. in 10 mins.; Philadelphia, 1·6 in. in 20 mins.; Chicago, 0·97 in. in 7 mins.; Taylor, Texas, September 1921 23 ins. in 24 hrs.; July 1916, at 12 stations in the U.S., more than 10 ins. in 24 hrs.; Orpid's Camp, California, 0·95 in. in 1 min., 1·17 in. in 10 mins., 2·2 ins. in 1 hr.; Campo, California, August 1891, 11·5 ins. in 80 mins.; New Orleans, 1926, 1927, and 1929, 5-hr. falls of 7·45, 7·88, and 10·38 ins., also near New Orleans, 20 ins. in 24 hrs., and 4·63 ins. in 60 mins.

The New England Storm of November 1927.

Comparing with the storms of August 1912 and June 1917 (above): New England, November 3 and 4, 1927, the values being very slightly less than and closely corresponding to those for the New England storm of July 1897, in 24 hrs.: Spot maximum, 8·7 ins; 500 sq. miles, 7·4 ins.; 1,000 sq. miles, 6·8 ins.; 2,000 sq. miles, 6·1 ins.; 3,000 sq. miles, 5·6 ins.; 4,000 sq. miles, 5·3 ins.; 5,000 sq. miles, 5·3 ins.; 7,000 sq. miles, 5 ins.

New England Floods of 1936.—A large area received 12 ins. in 8 days, with two peak periods Maximum one-day station falls included 6·46 5·07 and 5·46 ins. The discharges of the rivers corresponded to the following coefficients in the Ryves (R.) and Dickens (D.) formulas, Connecticut river, 4 stations: R., 416, 546, 550, 625; D., 208, 264, 260, 291. Merrimac river, 3 stations: R., 552, 570, 644; D., 358, 310, 321. Androscoggin river, one station: R., 1,480 or 1,615; D., 794 or 867. Three other streams: R., 241, 259, 281; D. 146, 167, 192. (See *Civil Engineering* (U.S.A.), May 1936; abstract, *The Engineer*, July 17, 1936.)

Ohio River Floods, January 1937.—At Murray, Kentucky, 5·88 ins. of rain fell in 24 hours. In the 12-day period; average for the Ohio basin, 8·0 ins. Run-off at Louisville (607 m. down-stream from Pittsburgh), 1,206,000 cusecs.

Heavy Station Rainfalls in or near the Tropics.

Central America, 0·6 in. in 10 mins.; 1·1 in. in 30 mins.; 1·7 in. in 60 mins. Panama Isthmus, 6 ins. in 95 mins.; 4·9 ins. in 60 mins.; 5·86 ins. in 60 mins. Philippines, 27 ins. in 24 hours, also (1913) 48 ins. in 48 hours, and 24 ins. in 8 hours (same storm). Manila, 17 ins. in 15 hours. Near Johannesburg, 3·2 ins. in 15 mins. Near Durban, 15·6 ins. in 15 hours; 17·6 ins. in 24 hours; 6 ins. in 3 hours. Mauritius, 18 ins. in 24 hours, and the mean of 15 stations 9·86 ins. in 24 hours; 99 ins. in 6 days; 24 ins. in 24 hours. Ceylon, 21 ins. in 24 hours; 8 ins. in 4 hours.

In 2 days: Santa Cruz, Teneriffe, 12·58 ins.
In 24 hrs.: Newlands, Cape Peninsula, South Africa, 16·5 ins.
In 12 hrs.: Beaumont, Texas, 13·54 ins.
In 8 hrs.: Santa Cruz, Teneriffe, 3·47 ins.
In 4½ hrs.: Beaumont, Texas, 12·76 in

Georgetown, British Guiana.—In 1933, 104·3 in.; in 1934, 80·3 in.; in 1935, 85·16 in. There was rain on 202 days, in 1935; 1 in. or more on 25 days; 2 in. or more on 5 days; maximum day's fall, 3·13 in.

RAINFALLS IN INDIA.

NORMAL RAINFALLS, JUNE–SEPTEMBER. INCHES.

Burma: 83·8; Bay Islands, 67·3; Lower Burma, 118·4; Upper Burma, 34·3.
Assam: 61·1. *Bengal:* 60·9. *United Provinces:* 36·1 (East, 36·1; West, 36·1).
Bihar and Orissa: 45·5; Orissa, 47·8; Chota Nagpur, 43·7; Bihar, 44·8.
Punjab: 15·7; East and North, 19·9; South-west, 8·3. *Kashmir:* 6·2.
North-West Frontier Province: 5·0; Baluchistan, 2·4. *Sind:* 4·7.
Rajputana: 18·1; West, 10·7; East, 21·7. *Bombay:* 37·9; Gujerat, 22·4; Deccan, 21·9.
Central India: 33·8; West, 28·5; East, 39·1; Berar, 27·4. *Mysore:* 15·5. *Malabar:* 74·2.
Central Provinces: 40·5; West, 41·9; East, 45·0; Konkan, 88·8.
Hyderabad: 26·7; North, 29·7; South, 24·5.
Madras: 16·3; South-east 10·8; Deccan 16·3; Madras Coast, North, 22·4.

NORMAL RAINFALLS, OCTOBER–MAY. INCHES.

Burma: 26·03; 7·98 in October, 11·1 in May. *Assam:* 33·05; 8·74 in April, 12·00 in May. *Bengal:* 21·83; 5·42 in October, 8·51 in May. *Bihar and Orissa:* 11·06; 3·65 in October, 3·00 in May. *United Provinces:* 5·42. *Punjab:* 5·47. *North-West Frontier Province:* 6·30. *Sind:* 1·40. *Rajputana:* 2·30. *Bombay:* 4·63. *Central India:* 3·65. *Central Provinces:* 6·05; 2·05 in October. *Hyderabad:* 6·23; 2·39 in October. *Mysore:* 15·02; 5·56 in October; 2·59 in November, 4·20 in May. *Madras:* 23·59; 8·06 in October, 6·69 in November, 2·59 in December, 3·13 in May.

RAINFALLS ELSEWHERE IN THE BRITISH EMPIRE.

South Africa.—At 550 ft. above King William's Town, in the Amatola Mountains, head-waters of Buffalo River, average annual rainfall for 39 years to December 1925, 73·99 ins.
Blijde River, 15 ins. Calitzdorp, less than 12 ins. Bon Accord, Transvaal, 28 ins., 21 ins. November–February. Van Ryneveld's Pass, Sunday River, about 14·5 ins.

The following are actual inches of annual rainfall, maximum, average and minimum, for a period of 24 years, in the Madras Presidency, 7 stations:—

70, 47, 31—78, 40, 17—48, 29, 9—60, 50, 17—43, 28, 17—165, 117, 87—182, 131, 95.

Rainfall in Nigeria.—In four zones; (1) coastal to (4) a plateau about 2,000 ft. in altitude. Percentages of mean falls, driest single years; (1) 61·5; (2) 66·9; (3) 67·6; (4) 66·7; means of driest three consecutive years: (1) 80·2; (2) 81·4; (3) 83·2; (4) 83·3; wettest single years: (1) 187·0; (2) 143·7; (3) 159·6; (3) 159·6; (4) 137·4. (*G. H. Ivory.*)

Hartebeestpoort Dam (near): altitude over 3,800 ft.—Monthly rainfalls and evaporations (approximate), inches: January, rainfall 4·99 and evaporation 6·51; and in that order, February, 4·11 and 5·65; March 3·31 and 5·07; April 0·80 and 5·13; May 0·69 and 5·10; June 0·06 and 4·35; July 0·27 and 4·91; August 0·73 and 6·60; September 0·48 and 7·91; October 2·26 and 8·41; November 4·66 and 7·63; December 3·98 and 7·40; the year 26·34 and 74·67.

RUN-OFF AND LOSSES.

The available supply and the losses by evaporation and absorbtion are often expressed as percentages of the rainfall, but the amount of run-off or of loss bears no definite relation to the amount received by the catchment. The yield of a catchment area may better be reckoned as proportionate to the rainfall less some constant quantity which may be called the 'handicap' (*Lapworth*).

Rivers Thames and Severn.—(E. G. Bilham.) From the data for the period 1914–36, there derives, for the Thames basin, the equation: Run-off = 0·57 rainfall *minus* 6·05, the run-off being expressed as the depth in inches over the whole catchment. From the gaugings of the Severn (R.M., p. 719) there derives: Run-off = 0·67 rainfall *minus* 6·66. For the periods, Thames, 1914–36, and Severn, 1921–36, the averages were: Thames, rainfall, 29·74 ins.; run-off, 10·71 ins.; Severn, rainfall, 38·14 ins.; run-off, 18·90 ins.; the respective losses being 19·03 ins. and 19·24 ins.

It is necessary to find the particular 'handicap' in every case, since its amount necessarily depends upon topographical and geological features, upon the area of the catchment and the manner in which the water is collected. Further, it must be noted that the method is by no means generally applicable, and notably fails when the effect of one year upon the next is strongly marked. In the Ajamenta Valley, Gold Coast Colony, the following, in inches, were year's falls and the corresponding yields (*Hunter*):—

38·8, 1·800—59·9, 8·280—47·7 2·540—42·4, 0·140—30·8, 0·560—29·6, 0·004—40·2, 0·006.

Again, if we take the figures for the Sweetwater catchment basin, California, we find that the following comparisons may be made (*Folwell*):—

A year's rainfall of 21 inches yielded 12 per cent., but a fall of 23·4 inches yielded only 9 per cent., while another year's fall of over 23 inches yielded less than 0·4 per cent. A fall of 17·2 inches yielded 4 per cent., but falls of 18 to 19 inches yielded less than a quarter per cent. The average fall of nearly 21 inches yielded less than 6 per cent., but an actual year's fall of 20 inches yielded 8 per cent.

The yield from the Coolgardie catchment, in a period of six years (*Palmer*) ranged from 0·20 per cent. to 3·5 per cent., while the rainfall ranged only from 19 to 33½ inches.

In a period of ten years the worst discharge of the Guadalquivir (Spain) was 9 per cent. of the rainfall, and in the best year it was 50 per cent. In the Karroo, S. Africa, a yield of 2 or 3 per cent. would be expected from a year's fall of 8 to 10 inches, and from a fall of 20 inches the yield would be about 8 or 10 per cent. (*Legg*).

The General Method.—The data necessary to determine a 'handicap' can be used—whether a 'handicap' can be found or no—to establish curves of run-off as proportional to rainfall; the units of rainfall times being such as hours, for computations of flood flows, and weeks or months with respect to water supplies and irrigation. The graphs should show curves for, respectively, soaked, damp and dry catchment, or, as regards long periods, the three cases of wet, normal, and dry conditions—precedent as :—

Miami River.—At Dayton, Ohio. Percentages of rainfall ; under the conditions :—

(1) Catchment under snow and ice ; frozen ground ; followed by rain and thaw : Dec.–March, 100 + ; April–June, 80 to 90.

(2) Wet ground, heavy preceding precipitation, or thaw after cold weather : Dec.–Mar., 60 to 75 ; April–June, 70 to 80 ; July–Sept., 30 to 40 ; Oct.– Nov., 45 to 60.

(3) Moderately wet, some preceding precipitation: Dec.–Mar., 50 to 60 ; April–June, 50 to 70 ; July–Sept., 20 to 30 ; Oct.–Nov., 30 to 45.

(4) Moderately dry, little preceding precipitation : Dec.–Mar., 35 to 50 ; April–June, 25 to 45 ; July– Sept., 10 to 15 ; Oct.–Nov., 20 to 30.

(5) Very dry : Dec.–Mar., 30 to 35 ; April–June, 5 to 20 ; July–Sept., 1 to 5 ; Oct. –Nov., 5 to 20.

Generally, in dry countries with a small rainfall the ratio of evaporation to the rainfall, or to a part of the rainfall, increases rapidly with a decrease in the actual amount of the fall. Conversely, in wet countries with a fairly large proportion of rainy or cloudy days the effect of relatively dry conditions or of scanty rainfall is not felt to nearly the same extent. In some cases it is found that the rainfall to be considered is that of only a part of the year, and it is then necessary to study maximum and minimum rainfalls, and the worst three years, as applying only to this season. In Natal about ⅞ of the rainfall occurs in the wet season, September to April. In Southern Rhodesia, about $\frac{11}{12}$ from November to March. In other cases, especially when the area does not feel the full effects of any of the seasonal rains, it is necessary to examine the rainfall records in detail for the worst three years, and to eliminate all falls of less than some predetermined amount (single falls or day's rain). Next, the yield from each of a series of falls should be estimated as, in each case, a proportion depending partly upon the amount of the fall and partly upon the interval which had elapsed since the previous fall. The maximum percentage yield will be assigned to continuous rain on soaked ground and soaked subsoil. In dry areas it may be necessary to adopt such a method of estimating the yield, when, owing to the nature and distribution of the rains, no general rule can be applied to the whole rainfall of the year, or to that of a season. For a rough estimate it may be sufficient to eliminate small or isolated falls and to calculate the yield of the remainder as a diminishing proportion of the remaining rainfall, less some constant. In some conditions the proportion need not be a diminishing one.

Absorption and Evaporation.

While it is possible, by the exercise of judgment, to form some idea as to the extent of losses by absorption for a particular area, it is clear that the large number of factors involved makes it impossible to lay down any general rules. Under some circumstances all the rain runs off, under other circumstances it is all absorbed, and there is every conceivable intermediate condition.

A few points are, however, worth noting. Clay soils, after the surface is wet, absorb water only very slowly. Some light soils are absorbent up to a given rate of absorption and rainfall in excess of that rate runs off. If the water absorbed by the soil percolates to places where it is available for use, the loss by evaporation from the soil will largely depend upon the average depth of the subsoil water. A dusty surface to a great extent prevents evaporation from the soil.

In practice, the loss from evaporation and percolation taken together is found by gaugings of the streams from the catchment, or, provisionally, is based upon data obtained from similar catchments, with such modifications as seem to be desirable owing to differences in the conditions. The estimate of the loss by evaporation and absorption, obtained from one set of data, should agree with the estimate of total yield based upon another set of data, or upon general considerations. References to evaporation from free water surfaces and from soils will be found under 'Irrigation.'

Dr. Walter Leather gives the following formula for estimating evaporation from water surfaces:—

$$E = 2\cdot0\,(\log t - 1\cdot74) + 0\cdot33\,(\log D - 1\cdot00) + 0\cdot36\,(\log W - 0\cdot125)$$

where,

E = Evaporation in millimetres for 24 hours; D = Dryness = 100 − (humidity at 8 A.M.); W = Mean wind velocity for the 24 hours in miles per hour; t = Mean temperature in degrees F. for the 24 hours.

Rainfall Quantities and Rates.

One inch of rain is equal to:—

2,323,200 cub. ft. per square mile; 14,485,000 gallons per sq. mile;
3,630 cub. ft. per acre; 22,633 gallons per acre. (Taking 1 cub. ft. = 6·235 gals.)

One inch of rain per hour is at the rate of:—

645 cub. ft. per second on a square mile, nearly 1 (1·01) cub. ft. per second on an acre;
241,419 gallons per minute on a square mile;
377 gallons per minute on an acre.

I. Table showing Equivalent Rainfall in Cubic Feet per Minute.

Areas in Acres. (A_1)	Rainfall in Decimals of an Inch in 24 Hours. (r) 0·1	0·2	0·3	0·4	0·5	0·6	0·7	0·8	0·9
1	0·252	0·504	0·756	1·008	1·260	1·513	1·765	2·017	2·269
2	0·504	1·008	1·513	2·017	2·521	3·025	3·529	4·034	4·537
3	0·756	1·513	2·269	3·025	3·781	4·537	5·294	6·050	6·806
4	1·008	2·017	3·025	4·034	5·042	6·050	7·059	8·067	9·075
5	1·260	2·521	3·781	5·042	6·302	7·562	8·823	10·083	11·343
6	1·513	3·025	4·537	6·050	7·562	9·075	10·587	12·100	13·612
7	1·765	3·529	5·294	7·059	8·823	10·587	12·352	14·120	15·881
8	2·017	4·034	6·050	8·067	10·083	12·100	14·120	16·134	18·150
9	2·269	4·537	6·806	9·075	11·343	13·612	15·881	18·150	20·420
10	2·521	5·042	7·562	10·083	12·604	15·125	17·646	20·166	22·687

Cubic feet per minute $= A_1 r \times 2\cdot521$.

II. Table showing Equivalent Rainfall in Cubic Feet per Minute.

Area in Square Miles (A_2)	Rainfall in decimals of an inch in 24 hours. (r) 0·1	0·2	0·3	0·4	0·5	0·6	0·7	0·8	0·9
1	161·3	322·6	484·0	645·3	806·6	968·0	1129·3	1290·6	1452·0
2	322·6	645·3	968·0	1290·6	1613·3	1936·0	2258·6	2581·3	2904.0
3	484·0	968·0	1452·0	1936·0	2420·0	2904·0	3388·0	3872·0	4356·0
4	645·3	1290·6	1936·0	2581·3	3226·6	3872·0	4517·3	5162·6	5808·0
5	806·6	1613·3	2420·0	3226·6	4033·3	4840·0	5646·6	6453·3	7260·0
6	968·0	1935·9	2904·0	3872·0	4839·9	5808·0	6776·0	7743·9	8712·0
7	1129·3	2258·6	3388·0	4517·3	5646·6	6776·0	7905·3	9034·6	10164·0
8	1290·6	2581·3	3872·0	5162·6	6453·3	7744·0	9034·6	10325·3	11616·0
9	1452·0	2904·0	4356·0	5808·0	7260·0	8712·0	10164·0	11616·0	13068·0
10	1613·3	3226·6	4842·0	6453·3	8066·6	9680·0	11293·0	12906·0	14520·0

Cubic feet per minute $= A_2 r \times 1613\cdot7$.

Flood Discharges.

For every area, however small or however large, there is some period of time at the end at which the effect of continuous rain reaches the maximum. What this period is depends upon all the circumstances, the size of the area in question, the nature of the rainfall, the physical character of the catchment, and the nature of the vegetation or of artificial works all affect the result.

Computations of maximum discharge are most reliable when based upon this 'period of total contribution.'

Investigations, practically confined to the more humid parts of Rhodesia, showed (*R. H. Roberts*) that for catchments ranging from 78 to 2,040 sq. miles their lengths from 12 to 76 miles, and the ratios of length to mean breadth from 1·85 to 4·5, the period of total contribution ('critical time') ranged from 4 to 29 hrs.

Merits of the Empirical Formula.

In most cases the maximum capacity of the catchment for surface water is reached only for short periods after the heaviest rushes of rain, and the amount of such storage affecting the result is the average over a relatively long period, and cannot be estimated with any degree of accuracy. A period longer than the critical period will give a lower rate of discharge, because the rate of rainfall will be less, and a period shorter than the critical period will give a discharge lower than the maximum, because the water from the more distant parts of the catchment will not have time to reach the outlet. It is usually necessary to have recourse to empirical formulæ, or formulæ partly empirical and partly rational, and the engineer has to exercise judgment in choosing one coefficient affecting the whole quantity or several coefficients affecting different factors. In some large river basins the effects of short storms over several tributary valleys may, by the simultaneous arrival of the resulting floods, yield a larger discharge than that due to the lower rate of rainfall for the critical period corresponding to the whole area. In such cases the purely empirical formula may be better than the rational formula, because the engineer makes a note of every real coefficient established by experience, and all these coefficients are strictly comparable.

Further, in the case of a steep, rocky valley, the river of which drains flat uplands, or glens under dense jungle, the heavier rainfall for the shorter 'period of total contribution' for the main valley may give a larger flood discharge than the smaller rainfall for the longer period of total contribution for the whole catchment. Even in normal river basins, areas projecting beyond the average distance of the watershed of the distant parts of the catchment may be eliminated, the loss in area being more than made up for by the gain in rate of rainfall for the reduced period of total contribution (*Ryves*).

The following formulæ are used in calculating flood discharges of rivers and, in some cases, are also employed for quite small catchments.

Ryves' formula—

$$Q = CM^{\frac{2}{3}}.$$

Q = discharge in cu. ft. per sec.; M = area in square miles, C is a coefficient usually ranging from 450 to 1,000, according to the nature and situation of the catchment; rainfall data are also taken into consideration, by comparisons with established results.

This formula is largely used in Southern India. (Discharge in cu. metres per sec. = the same coefficient $\times \frac{15}{1,000} \times$ (area in sq. kiloms.)$^{\frac{2}{3}}$).

For a considerable range of catchment areas the coefficient 1,000 is a good provisional figure; it applies to many parts of the tropics in hilly country. For small catchments in such districts the discharge may correspond to higher values, up to 2,000 for the maximum in an indefinite period of time. At the site of the Gatun dam, catchment 168 sq. m., the maximum flood, estimated from records, corresponded to a coefficient of 1,525. In the New York State flood of July 1935 a catchment of 2·91 sq. miles yielded a short-period run-off corresponding to a coefficient of 1,200, and another of 2·34 sq. miles, a coefficient of 1,153.

Dickens' formula—

$$Q = 825\ M^{\frac{3}{4}}.$$

This is the form in which the equation is usually presented, but it is doubtless advisable to vary the coefficient as in the case of the Ryves' formula.

The Dickens' formula is probably more particularly suited to the conditions of Bengal and other parts of Northern India, but both formulæ may be usefully employed in other countries.

Chamier's formula—

$$Q = 640 \times R \times C \times \frac{M^{\frac{3}{4}}}{M}.$$

R = average rate of greatest rainfall in inches per hour for such duration as will allow of the flood-water flowing to the outlet from the farthest extremity of the catchment. C = coefficient of surface discharge, giving the proportion of rainfall that may be expected to flow off the surface. M = area of catchment in square miles.

In order that the length of the catchment might be taken into consideration the following formula was proposed:—

Burge's formula—

$$Q = C.\frac{M}{(L)^{\frac{2}{3}}}$$

where L is the length of the main river or stream in miles ; C is usually taken as 1,300

A modification of this formula has been proposed, as follows (*Jackson*):—

$$Q = C \times \frac{B}{L} 100\ M^{\frac{3}{4}},$$

where L is the extreme length and B the extreme breadth of the catchment area.

It may be suggested that when it is possible to make a fair estimate of the period of total contribution, and, therefore, of the maximum rate of rainfall, such an estimate should form the basis of the calculation for maximum flood discharge, and when no such estimate can be made an empirical formula with a single coefficient should be used.

Storm-water Discharge of Sewers.

The same principles that govern the discharges of large river basins hold in the case of the storm-water discharge of sewers, but it is usually much easier to estimate the average rate of flow from the most distant parts of the catchment (measured in time) and, therefore, to estimate the period of total contribution, the rainfall for which can be found from local records.

The following are the formulæ employed:—

Bürkli-Ziegler's formula—

$$Q = R \times C \times \sqrt[4]{\frac{S}{A}}.$$

Q = discharge in cubic feet per second per acre.
R = average rainfall in cubic feet per second per acre 'during heaviest rainfall.'
C = 0·75 for paved streets; 0·31 for rural districts; average 0·625.
A = area of district drained in acres.
S = general fall of area per thousand.

It is clear that R will depend upon the period of total contribution which will have to be found in each case in order that the formula may be applied. The factor C would, as regards unpaved areas at any rate, also depend upon the period of tota contribution

Kuichling's formula—

$$Q = A \times a \times t \times (b - c \times t).$$

A = area drained, in acres ; a, b, c, empirical constants which vary for different localities; = time required for the concentration of storm water at the outlet.

This formula seems needlessly complicated, since if t be known the rainfall can be found and a single modifying coefficient could be applied to it, as well as one to be applied to the quantity as a whole, if considered necessary.

Lloyd-Davies' formula—

$$Q = \left(60 \cdot 5 \times \frac{60}{Tc} \times R\right) \times Ap.$$

Q = cubic feet per minute ; Tc = time of concentration in minutes (time of flow through the longest sewers) ; R = rainfall in inches during Tc ; Ap = percentage of impermeable area in acres.

The equation is as given by the author, *Min. Proc. Inst. C.E.*, vol. clxiv, p. 47. The expression 'percentage of the impermeable area in acres' implies that p is the percentage and A the area of the district in acres. The formula seems to depend upon a proportion of the area being regarded as fully impermeable and the remainder being regarded as having an assumed average run-off coefficient. Therefore, if 'percentage impermeability' be substituted, the basis of the equation seems to be disturbed and the estimation of maximum discharge to be suitably made by means of a formula such as those of the type

$$Q = C \times A^n$$

The Ministry of Health advises the use of :—

$$R = \frac{30}{T_c + 10}, \text{ where } T_c \text{ is 5 to 20 min.}$$

$$R = \frac{40}{T_c + 20}, \text{ where } T_c \text{ is 20 to 100 min.}$$

The use of a curve drawn through two points so found for $T_c = 12$ and $T_c = 60$ min. would seem to be more logical.

Heavy Station Rainfalls in India.

Madras, 4·95 ins. in 3 hours ; 2 ins. in 18 minutes ; 3 ins. in 60 minutes 10, 11, 12, 13, 18, and 20 ins. in 24 hours. Madras Presidency (South Arcot), 7, 8 9, and up to 11·3 ins. in 24 hours. Central Provinces (Talgaon), 7·75 ins. in 6 hours. West Coast, Bombay Presidency, 8 ins. in 2 hours. Bombay Town, 7 to 9 ins. in 24 hours. Sholapur, 10 ins. in 7 hours. Calcutta, Alipore Observatory, May 6, 1928, 3·25 ins. in 40 mins. (record intensity for this station or 4·88 ins. per hour) ; the whole fall during the storm being 3·39 ins.

In 24 hours : Karwar, Konkan, 13 ins., June ; Hukitala, Orissa, 15 ins., July ; Burdwan, Bengal, 10 ins., June ; Saugor Island, 10 ins., June ; Kyankpyu, 9 ins., June. In 48 hours : Bombay, 17 ins., June. In 72 hours : Cherrapanji, 31 ins.

Rivers of Western Bengal and Orissa : Maximum Flows.

Affluents of the Houghly.	Catchment area. Sq. Miles.	Max. Discharge. Cusecs.	Coefficient with the Ryves Formula.
Ajai	2,544	270,000	1,450
Damodar	7,211	650,000	1,743
Cossye	2,154	228,540	1,369
Flowing into the sea.			
Subarnarekka	6,473	300,000	863
Brahmini	13,700	642,510	1,122
Bytarni	3,761	400,000	1,653
Mahanadi	49,800	1,571,000	1,153

Forests and Rainfall.

A point which is often overlooked is the effect of trees in increasing the total precipitation considerably beyond that recorded by rain gauges. A large proportion of the rime which collects on the twigs of trees in frosts afterwards reaches the ground as water, and, in climates such as those of the British Isles, the total amount of water deposited on the twigs from fogs and drifting clouds is considerable, and most of it reaches the streams or underground storage or at least replaces losses from subsequent rainfall.

Of more importance, however, to hydraulic engineers is the effect of woodlands in modifying the run-off. The rush of water from bare hillsides is exchanged for the slower delivery from the matted carpet of woodland, losses by evaporation may be much diminished and the melting of snow usefully retarded. In catchments from which flood waters are largely lost, woodlands may increase the available run-off by extending the period of surface flow. The maximum floods of rivers are reduced, and the lowest summer flow increased. Woodlands are usually much more effective than minor vegetation, such as gorse and heather, in preventing the soil from being carried from the land into an open reservoir.

To protect a reservoir from silting, it may be unnecessary to plant large areas, the silt being arrested by suitable planting of narrow belts of woodland, or by the protection of natural growth, along the margins of the streams.

Some engineers consider that in the case of small reservoirs the shelter afforded by a belt of trees along the margins is of value in reducing the amount of scour of the banks caused by wave action. Afforestation over considerable areas in large river basins would, in many cases, reduce the amount of silting in navigable rivers and estuaries.

A matter which does not receive sufficient attention in connection with hydraulic engineering is the effect of judicious planting or woodland conservation over small areas. A narrow belt of woodland along the foot of a slope will arrest the soil brought down by rains from the hillside. The encouragement of dense vegetation along the bottom of a narrow valley or glen may check the rate of flood discharge to a useful extent. The planting of suitable trees along ridges, and for a little way down the slope facing the rain-bearing and damp winds, will produce the maximum of certain desired effects, in proportion to the area occupied. Suitable tree and bush growths in swampy areas and around their margins will increase their effect in checking flood discharge, and may prevent these areas from contributing large quantities of silt to the streams during very heavy rains. Areas of soft, cultivable soil liable to denudation may similarly be protected. Generally, a country which is, in the ordinary English sense of the words, 'well-timbered' is, from the point of view of the hydraulic engineer, a favourable country; and in the development of new lands the future effects of a proposed agricultural policy should be considered from this point of view, and in consultation with hydraulic engineers.

Forests and Run-off.—In order to carry out large-scale field tests of the effect of forest cover on the run-off and erosion catchments, three reservoirs, each of 10,000 cu. ft. capacity and concrete-lined, are being constructed in the Los Angeles National Forest, U.S.A. Measuring flumes and weirs will be installed immediately above each reservoir. After 5 years' records have been obtained, one of the areas will be burned and be maintained in a barren state, another will be burned and left to return to its natural condition (or take on the naturally-ensuing character, resulting from the burning), while the third area will be kept with its present cover, as a control.

RIVERS: CONTROL OF FLOW AND FLOODS.

Flow Control.—Seine Basin.—The Department of the Seine will have control of 128 million cu. m. of water in the four reservoirs:—the Crescent and the Bois de Chamauçon, together impounding about 22·5 million cu. m.; the Champaubert-aux-Bois, 23 million; these having been completed; and the Pannesière-Chaumard, under construction, 82·5 million cu. m., the water-spread of which will be 1,300 acres, and the greatest depths 150 ft. Dry-weather flow of the Seine in Paris will be increased by 50 per cent. The flood-control efficiency of the reservoirs will be reduced by the necessity for maintaining minimum levels for power production, 18 million kW.h. a year.

The Nile Basin.—A series of articles, *Civil Engineering* (London), October and December 1933, January, March, April 1934. Largely based on official and semi-official information. A study of the topography and hydrology of the regions, in respect of problems of control and improvement of the Nile and its tributaries and the extension of irrigation. Proposals for storage in the Nile Basin are reviewed.

The Sudd Region. In the fifth article, by R. A. Ryves, the problem of the Sudd region is discussed in two aspects, as related to the *régime* of the river, and in respect of engineering works. The character of proposed works is illustrated by a map to the scale 110 miles to an inch. (See also, *The Engineer*, 'Irrigation Works since the British Occupation of Egypt,' Sept. 20, 1895, and following articles; also, 'Irrigation Schemes in Egypt and the Sudan,' May 23, 1919, and following articles: an abstract, June 9, 1933, of H. E. Hurst's paper (*Soc. of Arts.*, May 30) on 'The Sudd Region of the Nile,' and 'The Training of the Upper Nile.' By F. Newhouse. Review, *The Surveyor*, February 24, 1939.

Floods.—'Flood Flows: a Study of Frequencies and Magnitudes.' By Allen Hazen, M.Am.Soc.C.E. Reviewed in *The Engineer*, October 31, 1930. The data are entirely American.
'Measures of Flood Control.' By R. A. Ryves, *The Surveyor*, June 3, 1932. Relates to the broader aspects of the subject.

Flood Control.—'The Flood Danger to London.' By R. A. Ryves. Article, *Civil Engineering* (London), June 1933. A study of the natural conditions causing abnormally high tides in the Thames estuary; a review of the reports and of some papers relating to the subject; a commendation of A. B. Buckley's proposals for the creation of reservoirs in the Thames basin and condemnation of the works of river flow alteration as carried out and proposed by the Thames Conservancy. The part played by the winds in respect of the high tides of January 7 and 8, 1928, is considered in respect of the whole of the North Atlantic Ocean.

The Critical River Width.—(K. P. Pillai). On 10 miles of the River Cauvery the width varied from 700 ft. to 2,000 ft., and after a theoretical bed slope between controlled points had been plotted, it was found that the bed level, averaged on a cross section, was above that so given wherever the river exceeded 900 ft. in width and was below it where the width was less than 900 ft. Accordingly, that dimension was taken to be the proper width for the river if trained.

River Euphrates.—Works put in hand in June 1939 will provide for flood control on the Euphrates by storage of water in Lake Habbaniyah, which lies to the south of the river, about 30 miles from Baghdad, as proposed by the late Sir William Willcocks in 1911. The works will include the excavation of the Ramadi inlet channel, about 7½ kilom. from the river to the lake; cutting an escape channel about 8 kilom. long, leading to a depression in a desert region, and the provision, on the inlet channel, of a regulator with 12 six-metre openings and, on the escape channel, a regulator with 8 openings of 6 m. It is proposed to excavate, at some future time, a channel for returning stored water to the river. The works are described, with a map, *The Engineer*, June 30, 1939.

Rivers Mersey and Irwell.—In an article, *The Surveyor*, December 22, 1939, by S. Pearson, the adjoining river basins are described, and an account given of their history and their present régimes. Works carried out for the control of the rivers and their tributaries are described.

Rain Gauges.

The smallest diameter for a serviceable rain gauge is 5 ins. Trials with gauges of various diameters ranging from 1 in. to 2 ft. have shown that if they are set perfectly level, and observed with great care, exactly the same rainfall has been registered by all of them.

The standard height above ground for rain gauges is 1 ft.

The higher a gauge is fixed above the ground the smaller is the quantity of rain caught, owing to the eddies carried by the wind about any prominent upstanding object. A curve, given in Beardmore's Hydrology, gives the divisor to be used for increasing the quantity of rain observed at any given height above ground level, so as to make the observation equal to that at ground level. The divisors are: for 10 ft., 0·93; 20 ft., 0·88; 30 ft., 0·84; 40 ft., 0·80; 50 ft., 0·77.

It is more important to have rain gauges placed at a uniform height above the ground than to have them of uniform diameter.

Literature.

Thames Floods.—Letters to the Editor of *The Engineer*; from Major F. Newhouse, published December 10, 1937, from Mr. R. A. Ryves, December 17, 1937, and from Mr. E. J. McCaig, January 21, 1938. Major Newhouse pointed to the fact that the greatest rainfall yet recorded, for the period of total contribution, falls far short of the maximum as estimated by means of comparative data. Mr. McCaig drew attention to the effects of the enclosure of the marshes of the lower reaches of the Thames, and to the influence of low atmospheric pressure in increasing the heights to which tides rise.

'The Maximum Probable Flood and its Relation to Spillway Capacity.' By S. M. Bailey and G. R. Schneider. Article, with two isohyetal maps, *Civil Engineering* (*U.S.A.*), January 1939.

'Conservation of Water Through Recharge of the Underground Supply.' By A. T. Mitchelson; Paper, 1938, Water Conservation Conference in Salt Lake City. *Civil Engineering* (*U.S.A.*), March 1939. Four principal methods of 'water spreading' are described.

'The Climate of the British Isles.' Book: by E. G. Bilham.

'The Trend of Annual Rainfall in Scotland.' By H. E. Carter. Paper, Inst. Water E.

'England's Rainfall Problem: a Survey of Rainfall, Drought and Distribution.' By H. Spence-Hales and John Bland. Reviewed, *The Surveyor*, Aug. 18, 1939.

'The Accuracy of Run-off Calculations.' By L. B. Escritt. Article, *The Surveyor*, August 4, 1939.

'Flood Discharge and Damage to Bridges on the Snowy River, Victoria, Australia.' By M. G. Dempster and W. A. Ozanne. Article, *The Commonwealth Engineer*, March 1934. Abstract, *The Surveyor*, June 1, 1934, and No. 220, *Engineering Abstracts*, Vol. 61.

'Stormwater Drainage in Flat Country.' By E. Dixon Grubb. Article, *The Surveyor*, June 6, 1941.

'Land Drainage and Reclamation. By Q. C. Ayres and D. Scoates. Book (second edition).

'Ganges Flood and its Lesson.' By C. C. Majumdar. *Journal Inst. E.* (*India*), April 1941.

'Reliability of Station-year Rainfall Frequency Determinations.' By Katherine Clarke-Hafstad. *Proc. Am. Soc. C.E.*, November 1940. Discussion, January, February, March and April, 1941.

'Maximum Probable Floods in Pennsylvania Streams.' By C. F. Ruff. *Proc. Am. Soc. C.E.*, September 1940. Discussion, December 1940, January, February and May, 1941.

Addenda 1948.

Heavy Rainfalls.

Jamaica.—At Silver Hill Plantation, in the Blue Mountains, 135 ins. fell in 8 days, including 114·5 ins. in 5 days. At Farm Hill Plantation, 114·4 ins. fell in 7 days, including 95·88 ins. in 4 days and 28·5 ins. in one day. Both falls in August 1909. The mean annual rainfall over a period of 70 years was, for the whole island, 77·76 ins.

The Fenland.—The rainfall for March 1947 was 254 per cent. of the average for the period 1881–1915. At some stations more than 300 per cent. of the average was recorded.

Fenland Flooding.—The rainfalls of November and December, 1946, were well above the average. In January, February and March, 1947, they were, allowing for melted snow, 3·4 ins., 2·1 ins. and 6·8 ins., respectively. At St. Neot's Bridge, over the River Ouse, the water level was higher than at any time since 1823. (At the watergate near Worcester Cathedral the flood was the highest since 1770. At Gloucester the highest previous flood, in 1852, was exceeded by 6 ins.).

Area of Flooding.—Officially given as 690,000 acres, but, in the opinion of Mr. E. A. G. Johnson, the total flooding exceeded 1,000,000 acres, about 325,000 acres of which was arable land—the flooding of which probably did not last longer than 10 days, whilst on the remainder of the area it was caused by embankment failures and lasted for some weeks. (See 'Flooding in the Fens and Remedial Measures Taken.' A paper (*Journal Inst. C.E.*) by E. A. G. Johnson.)

Fen Flood Prevention Scheme.—Apart from the heightening of many miles of river embankments, which was done soon after the flooding, a scheme of preventive measures has been prepared. In November 1948 the Great Ouse Catchment Board decided to promote in Parliament a Bill for a scheme which it is believed will protect the Fens from flooding for 50 years. The estimate of the cost is £6,500,000.

Water engineers may wonder why works which will prevent flooding for 50 years should not prove effective indefinitely. Weirs for delaying the flow of tributaries and that of upper reaches of the Great Ouse, the water being released at low tide, can be so designed that silt deposited upstream of them can be scoured away. Such weirs will, presumably, be important features of the scheme. Provisions for scour of silt can be provided, also, at the dams of reservoirs, and in so far as they would fail to remove silt from the upper part of a reservoir, the cost of dredging would be very small compared with the losses caused by floods.

Other Flooding, 1948.—The March rains caused the River Trent, at Nottingham, to attain a level which was the highest since 1875. On the River Lea flood levels were records for the 100 years for which records have been kept. As in the case of the Thames, a factor affecting flood levels is diminution of the area of the river's flood channel by the creation of large-area embanked reservoirs, a number of which have been created in recent years.

The level of the Yorkshire Ouse,— at York, was the highest since 1831.

An August Flood.—The flood of August 1945 in northern Northumberland was one of great severity. Some 1,500 acres of grassland were rendered useless for the rest of the season. About 300 acres of grain crops were destroyed and 200 to 250 acres seriously damaged. Hay crops amounting to 150 tons were lost, and 500 sheep and 550 head of poultry drowned. Railway embankments were breached and rail services suspended for many days.

Extremes of Rainfall.—British Isles, 1870–1935. At Llyn Llydaw, Carnarvon, 246·92 ins. and at the Stye, Cumberland, 247·3 ins. in 1923. At Margate, 9·29 ins. in 1921.

Heavy 24-Hour Falls.—British Isles, 1922–1944. Belfast, August 27, 6·00 ins., Abergwesyn, July 18, 6·05 ins. Loch Carron, April 2, 6·50 ins. Buttermere, July 29, 7·14 ins. Rhondda, November 11, 8·31 in. Brymore, August 18, 9·40 in.

Recent Literature.

'The Flood Problem: Its Philosophy and Its Facts.' By R. A. Ryvers, M.Cons.E., *The Engineer*, May 30, 1947.

'Soil Erosion in New Zealand.' Book, by K. B. Cumberland. Second edition, 1947. Whitcombe and Tombs, Ltd.

'The Flow of the River Nene.' By R. F. Wileman and H. W. Clark. Paper, *Journal Inst. C.E.*, April 1946.

'Floods and Flood Control.' Article in the publication 'Ten Eventful Years,' of *Encyclopædia Britannica* (inc.).

A shorter article with the same title in the *Encyclopædia Year-Book*, 1946.

SECTION XVIII

PART II

THE FLOW OF WATER—FORMULAE AND DATA—WEIRS, SLUICES AND CHANNELS.

(Revised and mainly contributed by R. A. Ryves, M.Cons.E.)

Discharge of Water through Submerged Orifices.

The theoretical velocity, v, with which water flows from an orifice in the side of a vessel at a depth h from surface is the same as that of a body falling freely by gravity from height, h, viz.

FIG. 1.

FIG. 2.

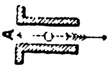


FIG. 3.

$v = \sqrt{2gh}$, but practically the converging currents produce a contraction of the jet, which assumes the form of the vena contracta, d, e, f, g, fig. 2.

h = height from surface of water to centre of orifice, in feet. v = velocity of issuing water

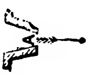


FIG. 4.

FIG. 5

in feet per second. A = area of orifice, in feet. Q = cubic feet discharged per second. g = coefficient of terrestrial acceleration.

$v = m\sqrt{2gh}$. $\quad Q = mA\sqrt{2gh}$. $\quad 2g = 64{\cdot}4 \quad \sqrt{2g} = 8$ practically

m.

·625. Where the orifice is a thin plate, bc (fig. 1), with thickness less than ½ smallest dimension of orifice.

·97. Orifice shape of vena contracta, d, e, f, g (fig. 2); A is measured at f, g.

·82. Orifice with an adjutage (fig. 3); O = 2½ to 3 times least dimension at A.

·9. Sides converging at angle of 13½° (fig. 4). m depends on diameter and angle of convergence.

1·46 Sides diverging at angle of 5° 6′ (fig. 5). O = 9 times least dimension at A, which is of form of vena contracta. (Seldom used.)

·548. Where 2 sluices in thin plates are so close as to interfere with each other; m being reduced from ·625.

·62. For sluices of moderate size in lock gates, etc.

·82. For narrow-bridge openings.

·92. For very large sluices and bridge openings

Flow of Water over Sharp-edged Notches and Weirs.

Flow over a triangular notch ; see equations 12, 13 and 14.

Flow over any trapezoidal notch = flow over a rectangular weir of equal length with two end contractions, plus the flow through a triangular notch of corresponding angle.

Flow over a rectangular weir with end contractions varies as—

$$Q = 3 \cdot 10\ L^{1 \cdot 02}\ H^{1 \cdot 47} \qquad (1)$$

for all weirs up to at least 19 ft., and for head up to half length of weir, provided depth of pool below sill of weir is not less than twice the head. L = length in feet.

General law for trapezoidal weirs :

$$Q = 3 \cdot 10\ L^{1 \cdot 02}\ H^{1 \cdot 47} + 2 \cdot 48 n H^{2 \cdot 47} \qquad (2)$$

For n, see equation (16).

(*H. J. F. Gourley and B. Santo Crimp*, '*Proceedings Inst.C.E.*,' vol. cc. p. 388, Nov. 1915.

Hydraulic Radius.—The hydraulic radius, or hydraulic mean depth, is the area of the water cross-section divided by the length of wetted perimeter, and is denoted by H.M.D., R or r. For a pipe flowing full, diameter divided by 4. For many purposes the indices for h are taken as 1·5, 2·5, instead of 1·47, 2·47.

Rectangular Weirs and Notches.

According to the different cases we have the following formulæ giving the discharge in cub. ft. per sec. :

With end contractions and a velocity of approach,

$$Q = 5 \cdot 347\ CL \sqrt{(H + 1 \cdot 4h)^3} \qquad (3)$$

where water has no velocity of approach,

$$Q = 5 \cdot 347\ CL \sqrt{H^3} \qquad (4)$$

without end contractions but with velocity of approach,

$$Q = 5 \cdot 347\ CL \sqrt{(H + 4/3h)^3} \qquad (5)$$

where water has no velocity of approach,

$$Q = 5 \cdot 347\ CL \sqrt{H^3} \qquad (6)$$

in which

L = Length of weir in ft. ;
H = Head in ft. ;
V = Velocity of approach in ft. per sec. ;
h = Head equivalent of velocity of approach = $0 \cdot 01555\ V^2$
C = Coefficient of discharge ;
Q = Discharge in cub. ft. per sec.

Values of 'C.'

Effective Head in ft.	For Weirs with Contracted Ends. Length of Weir in ft.						
	·66	1	2	3	5	10	19
·10	·632	·639	·646	·652	·653	·655	·656
·15	·619	·625	·634	·638	·640	·641	·642
·20	·611	·618	·626	·630	·631	·633	·634
·25	·605	·612	·621	·624	·626	·628	·629
·30	601	·608	·616	·619	·621	·624	·625
·40	·595	·601	·609	·613	·615	·618	·620
·50	·590	·596	·605	·608	·611	·615	·617
·60	·587	·593	·601	·605	·608	·613	·615
·70	—	·590	·598	·603	·606	·612	·614
·80	—	—	·595	·600	·604	·611	·613
·90	—	—	·592	·598	·603	·609	·612
1·00	—	—	·590	·595	·601	·608	·611
1·20	—	—	·585	·591	·597	·605	·610
1·40	—	—	·580	·587	·594	·602	·609
1·60	—		—	·582	·591	·600	·607

VALUES OF 'C'—*continued.*

Effective Head in ft.	For Weirs without Contracted Ends. Length of Weir in ft.						
	19	10	7	5	4	3	2
·10	·657	·658	·658	·659	—	—	—
·15	·643	·644	·645	·645	·647	·649	·652
·20	·635	·637	·637	·638	·641	·642	·645
·25	·630	·632	·633	·634	·636	·638	·641
·30	·626	·628	·629	·631	·633	·636	·639
·40	·621	·623	·625	·628	·630	·633	·636
·50	·619	·621	·624	·627	·630	·633	·637
·60	·618	·620	·623	·627	·630	·634	·638
·70	·618	·620	·624	·628	·631	·635	·640
·80	·618	·621	·625	·629	·633	·637	·643
·90	·619	·622	·627	·631	·635	·639	·645
1·00	·619	·624	·628	·633	·637	·641	·648
1·20	·620	·626	·632	·636	·641	·646	—
1·40	·622	·629	·634	·640	·644	—	—
1·60	·623	·631	·637	·642	·647	—	—

OTHER FORMULÆ.

Velocity of Approach.—The velocity with which the water approaches a weir or other passage in a stream or channel is allowed for in the discharge measurement by converting it to head, in feet thus:

$$h_a = \frac{v_a^2}{2g} = 0\cdot01555\, v_a^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (7)$$

For gauging purposes the increased section, if the weir be made for the purpose, gives the velocity of approach. For design purposes, it may usually be assumed that the river will silt up to its previous sectional area.

Drowned Notch.—If h_1 be the head to the bottom of the notch and h the difference between water levels, an approximate measurement is given by:

$$Q = C.l.\sqrt{2g}.\sqrt{h}\left(h_1 - \frac{h}{3}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (8)$$

the same coefficient being used for both portions of the opening.

Drowned Weir.—For a drowned weir, or a rectangular notch, a coefficient C_1, such as $0\cdot7$ to $0\cdot8$, may be taken for the lower portion, and C_2, such as $0\cdot577$, for the upper portion. If d be the depth of tail water above the sill and h the difference between head water and tail water, the discharge is given by:

$$Q = Q_1 + Q_2 = C_1 \times l \times d \times \sqrt{2gh} + \tfrac{2}{3} \times C_2 \times l \times h \times \sqrt{2gh}, \text{ or}$$

$$= l \times \sqrt{2gh}\,(C_1 \times d + \tfrac{2}{3} \times C_2 \times h) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (9)$$

Clear Overfall Weir, with Velocity of Approach.—If the coefficient, for a broad-crested weir, be taken as $0\cdot577$ the discharge may be approximately calculated by:

$$Q = 3\cdot1 \times l\left\{(h + h_a)^{\frac{3}{2}} - h_a^{\frac{3}{2}}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (10)$$

Drowned Weir with Velocity of Approach.—Similarly, a useful working equation for a broad-crested weir when drowned, taking c_1 as $0\cdot8$ and c_2 as $0\cdot577$, is

$$Q = l \times \left[6\cdot4 \times d(h + h_a)^{\frac{1}{2}} + 3\cdot1\left\{(h + h_a)^{\frac{3}{2}} - h_a^{\frac{3}{2}}\right\}\right] \quad . \quad . \quad . \qquad (11)$$

SYPHON WEIRS AND BELL-MOUTHED WEIRS. See PART III.

To raise a River a certain Height by a Weir.

Q = discharge of river in cubic feet per second ; h_1 = height by which river is to be raised in feet ; h_2 = depth of top of weir below original surface of water necessary to raise surface by height h_1 ; h_a = head corresponding to velocity of approach ; l = length of weir in feet ; m = coefficient of contraction = 0·628 ; $2g$ = 64·4.

$$h_2 = \frac{Q}{ml\sqrt{2g(h_1 + h_a)}} - \frac{2}{3}\,\frac{(h_1 + h_a)^{\frac{3}{2}} - h_a^{\frac{3}{2}}}{\sqrt{h_1 + h_a}} \quad . \quad . \quad . \quad . \quad 12)$$

If h_a is not taken into account, then :

$$h_2 = \frac{Q}{ml\sqrt{2gh_1}} - \frac{2}{3}h_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

Gauging Streams.

Flow of Water over Sharp-edged Notches and Weirs.

The flow of a small stream, or of a sluggish channel or canal, is conveniently and accurately measured by building a weir across the stream and measuring the flow through a notch, or a number of notches, in the sill-board of the weir. It is most convenient, and is likely to provide the more accurate measurements, if the depth and width of water upstream of the weir be great enough for velocity of approach to be practically eliminated. The triangular, or Vee, notch is to be preferred, since the relation of wetted perimeter to area remains constant for all heads. Care should be taken that the sides of the notch are equally inclined to the vertical. If a series of Vee notches be used, their apexes must be in exactly the same horizontal line, and they should be seven times the head apart. If they are not less than 3½ times the head apart the error will not exceed 0·5 per cent. (*H. S. Rowell.*)

' V '-Notch Method.—The ' V '-shaped notch, fig. 6, is cut in the sillboard, and is of sufficient size to allow the full volume of water to discharge through it. When the flow is steady from the upper side of the notch, as shown in the plan, then, from still water,

$$Q = \frac{4}{15}\,kwh\sqrt{2gh}, \text{ cubic feet } \textit{per second} \quad . \quad . \quad . \quad . \quad (14)$$

w, h, and g being in *feet*.

The value of k varies from 0·59 to 0·62, according to the angle a, being 0·59 for a right angle 90°).

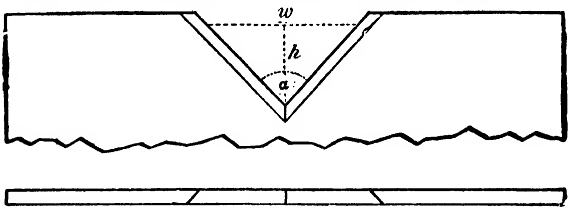


FIG. 6.

When a is a right angle, then $w = 2h$, and formula becomes

$$Q = \cdot 3h^2\sqrt{h}, \text{ cubic feet } \textit{per minute} \quad . \quad . \quad (15)$$

h being in *inches* and $k = 0·59$.

More convenient for some measurements is the formula

$$Q = 2·48nH^{2·47} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

Q = cub. ft. per sec. ; n = tangent of half the angle, a, of notch ; H = head in feet ; the implied coefficient being about 0·581. This is included in equation (2), for the notch portion of a trapezoidal weir.

Measurement of Head.—The dispositions, generally, are as in the following method, for both the Vee notch and the rectangular notch. The bottom and sides of the notch are bevelled on the downstream side, leaving the upper edge sharp. A stake driven into the bed of the stream some feet upstream of the weir, with its top level with the sill of the notch, provides for measurement of head. More accurate measurements, as now usually made, are obtained by providing a stilling chamber—if only a glazed, earthenware pipe set on end in a bed of loose stones—and employing a hook gauge, measuring downward from a fixed level above the weir sill. The hook gauge is essentially a rod with a 180° bend, or hook, which dips below the surface of the water, and is adjusted by a screw which brings the point of the hook up to the surface. The height is then read by a vernier, or by a micrometer screw. If a metal scale be used, the head can be read to within one-hundredth foot or less.

Equations (4) and (6) may be employed, and the table following equation (6). This method may be employed on any scale, and is often applied to exact models of proposed works, simulating the conditions which affect the coefficient: as, broad crests, end contraction, degree of elimination of end contraction, drowning, and velocity of approach. The sharp-edged notch weir is a *contracted weir*, but if the sides be made continuous with the channel it is a *weir without end contractions*, the coefficient being greater. A yet greater coefficient may be obtained by fluming the weir, an economy in respect of construction in such works, and at canal outlets to branch canals.

Automatic Recorders.—These are so designed as to mark and record the water-levels, or the corresponding computed discharges. (See p. 197.)

Fluming.—By making the sides of a weir, sluice, bridge opening, canal bifurcation, canal outlet to a branch, or any other water passage, continuous with the sides of the channels, or channels adjacent to them, end contraction is suppressed. Similar treatment of the bottom, in some of those cases, aids in increasing the coefficient of discharge. Such dispositions amount to fluming only when the curves approach such as the water would make for itself if flowing through a plastic but adhesive material, not having weight. Thus the design of the side walls for an existing sluiceway which is narrower than the channel, and is to be flumed, may often be indicated by the form of the mud banks in the approach to the sluiceway.

Sluices

Coefficients for the discharges through submerged orifices and sluices will be found on the first page of this Part. The coefficient for a sluice depends on the design, which is often directed to obtain a high value. In the design of the Stoney gates of the Sukkur Barrage a provision has been made whereby, when the gate is raised, the grooves below water-level are covered by plates. The coefficient of discharge is improved and the entry of mud or sand into the grooves is prevented.

Sometimes, however, a low-level sluice under a great head of water, or any sluice under some conditions, is designed rather in view of small erosive effect than in regard to high discharge capacity. Toothed sills, baffle chambers, and other devices are thus employed.

Waterway of Bridges.—See p. 490.

Sluice Discharge Formula (J. H. Jones).—Confirmed (L. G. Puls) by measurements of the flow through the sluice-gates of the Wilson Dam, 38 ft. in span, and holding up water to 18 ft. above sills. The formula is

$$Q = cDL\left\{2g(H - cD)\right\}^{\frac{1}{2}} \quad . \quad . \quad . \quad . \quad . \quad (17)$$

where c is the coefficient; D the depth of gate opening; L the span; H the depth of water to the sill.

Flow of Water in Pipes, Flumes, and Channels.

The principal formula of flow in pipes, flumes and channels is based on the formula for velocity of flow known as *Chesy's* formula, which is as follows

$$V = C\sqrt{rs}$$

where,

V = velocity in ft. per sec.;
C = a coefficient of discharge;
A = area of cross section considered, in sq. ft.;
p = wetted perimeter in ft.;
r = hydraulic radius = $\dfrac{\text{area of section}}{\text{wetted perimeter}} = \dfrac{A}{p}$;
s = hydraulic gradient or slope of channel = $\dfrac{\text{difference of level}}{\text{distance}} = \dfrac{h}{l}$
h = fall in ft. between points considered;
l = distance in ft. between points considered.

The formula for the flow is $\quad Q = AC\sqrt{rs},$
where

Q = discharge in cub. ft. per sec.

The value of the coefficient C can be determined either by Kutter's formula or Bazin's formula.

Kutter's formula is as follows:

$$C = \frac{41{\cdot}65 + \frac{0{\cdot}00281}{s} + \frac{1{\cdot}811}{n}}{1 + \left(41{\cdot}65 + \frac{0{\cdot}00281}{s}\right)\frac{n}{\sqrt{r}}} \quad . \; . \; . \; . \; . \; . \; . \; . \; (18)$$

where,

n = the coefficient of roughness.

Robt. E. Horton gives in *Engineering News*, of February 24, 1916, the following category of coefficients of roughness for use in Kutter's formula:

TABLE OF '*n*' FOR KUTTER'S FORMULA.

(*R. E. Horton.*)

Surface.	Perfect.	Good.	Fair.	Bad.
Uncoated cast iron pipe	0·012	0·013	0·014	0·015
Coated cast iron pipe	0·011	0·012*	0·013*	—
Commercial wrought iron pipe, black	0·012	0·013	0·014	0·015
Commercial wrought iron pipe, galvanised	0·013	0·014	0·015	0·017
Smooth brass and glass pipe	0·009	0·016	0·011	0·013
Smooth lockbar and welded 'OD' pipe	0·010	0·011*	0·013*	
Riveted and spiral steel pipe	0·013	0·015*	0·017*	—
Vitrified sewer pipe	0·010 / 0·011	0·013*	0·015	0·017
Glazed brickwork	0·011	0·012	0·013*	0·015
Brick in cement mortar; brick sewers	0·012	0·013	0·015*	0·017
Neat cement surfaces	0·010	0·011	0·012	0·013
Cement mortar surfaces	0·011	0·012	0·013*	0·015
Concrete pipe	0·012	0·013	0·015	0·016
Wood-stave pipe	0·010	0·011	0·012	0·013
Plank flumes:				
Planed	0·010	0·012	0·013	0·014
Unplaned	0·011	0·013	0·014	0·015
With battens	0·012	0·015	0·016	—
Concrete lined channels	0·012	0·014*	0·016*	0·018
Cement-rubble surface	0·017	0·020	0·025	0·030
Dry-rubble surface	0·025	0·030	0·033	0·035
Dressed-ashlar surface	0·013	0·014	0·015	0·017
Semi-circular metal flumes, smooth	0·011	0·012	0·013	0·015
Semi-circular metal flumes, corrugated	0·022	0·025	0·0275	0·030
Canals and ditches:				
†Earth, straight and uniform	0·017	0·020	0·0225*	0·025
Rock cuts, smooth and uniform	0·025	0·030	0·033*	0·035
Rock cuts, jagged and irregular	0·035	0·040	0·045	—
Winding sluggish canals	0·0225	0·025*	0·0275	0·030
Dredged earth channels	0·025	0·0275*	0·030	0·033
Canals with rough stony beds, weeds on earth banks	0·025	0·030	0·035*	0·040
Earth bottom, rubble sides	0·028	0·030*	0·033*	0·035
Natural stream channels:				
(1) Clean, straight bank, full stage, no rifts or deep pools	0·025	0·0275	0·030	0·033
(2) Same as (1), but some weeds and stones	0·030	0·033	0·035	0·040
(3) Winding, some pools and shoals, clean	0·035	0·040	0·045	0·050
(4) Same as (3), lower stages, more ineffective slope and sections	0·040	0·045	0·050	0·055
(5) Same as (3), some weeds and stones	0·033	0·035	0·040	0·045
(6) Same as (4), stony sections	0·045	0·050	0·055	0·060
(7) Sluggish river reaches, rather weedy or with very deep pools	0·050	0·060	0·070	0·080
(8) Very weedy reaches	0·075	0·100	0·125	0·150

* Values commonly used in designing.

† In British Indian practice the values would be shifted one column to the left, 0·017 being hardly attainable with earthen channels 0·020, perfect; 0·0225, good; 0·025, fair; 0·030, bad. See Part IV.

Maximum Discharge.—When there is no head over the pipe the water running with a velocity depending on the slope at which the pipe is laid, the greatest discharge is not with the pipe flowing full, but is given by angle at the centre of the pipe subtended by the water surface, about 54°.

TABLE OF COEFFICIENTS C FOR VARIOUS HYDRAULIC RADII (r), ACCORDING TO KUTTER'S FORMULA.

(For the same *actual* r (such as 3·28 ft., or 1 metre) the coefficient for metric units is the coefficient for British units $\times \frac{100}{181}$, or $\frac{5}{9}$, nearly.)

Slope S = ·000025 per Unit of Length, = 1 in 40,000, = ·132 Foot per Mile.

Hydraulic Radius, r, in Feet.	Coefficients n of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	65	57	50	44	40	3	28	23	17	14	12	10
·2	87	75	67	59	53	45	38	31	24	19	16	14
·4	111	97	87	78	70	59	51	42	32	26	22	19
·6	127	112	100	90	81	69	60	49	38	31	26	22
·8	138	122	109	99	90	77	66	55	43	35	30	25
1	148	131	118	106	97	83	72	60	47	38	32	28
1·5	166	148	133	121	111	95	83	69	55	45	38	33
2	179	160	144	131	121	104	91	77	61	50	43	37
3	197	177	160	147	135	117	108	88	70	59	50	44
4	209	188	172	158	146	127	113	96	78	65	56	49
6	226	206	188	174	161	142	126	108	88	74	64	57
8	238	216	199	184	171	151	135	117	96	82	71	63
10	246	225	207	192	179	159	142	124	102	87	76	68
12	253	231	214	198	186	165	149	129	107	92	81	72
16	263	242	223	208	195	174	157	138	115	100	88	79
20	271	249	231	215	202	181	164	144	121	106	94	84
30	283	261	243	228	215	193	176	157	133	117	104	95
50	297	274	257	241	228	207	190	170	147	130	117	107
75	306	284	267	251	238	217	200	180	157	140	127	117
100	312	290	273	257	244	223	207	187	163	147	134	124

Slope S = ·00005 per Unit of Length, = 1 in 20,000, = ·264 Foot per Mile.

Hydraulic Radius, r, in Feet.	Coefficients n of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	78	67	59	52	47	39	33	26	20	16	13	11
·2	100	87	77	68	62	51	44	35	26	21	18	15
·4	124	109	97	88	79	66	57	46	35	28	24	20
·6	139	122	109	98	90	76	65	53	41	33	28	24
·8	150	133	119	107	98	83	71	59	46	37	31	27
1	158	140	126	114	104	89	77	64	49	40	34	29
1·5	173	154	139	126	116	99	87	72	57	47	40	34
2	184	164	148	135	124	107	94	79	62	51	44	38
3	198	178	161	148	136	118	104	88	71	59	50	44
4	207	187	170	156	145	126	111	95	77	64	56	49
6	220	199	182	168	156	137	122	105	85	72	68	56
8	228	206	189	175	163	144	129	111	91	78	68	61
10	234	212	195	181	169	149	134	116	96	82	72	64
12	238	217	200	185	173	153	138	120	99	86	75	68
16	245	223	206	191	180	160	144	126	106	91	81	73
20	250	228	211	196	184	165	149	131	110	96	85	77
30	257	236	219	204	192	172	157	139	118	103	92	84
50	266	245	228	213	201	181	165	148	127	112	101	93
75	272	250	233	218	207	187	171	153	138	119	108	99
100	275	254	237	222	210	190	175	158	137	123	112	104

Slope S = ·0001 per Unit of Length, = 1 in 10,000, = ·528 Foot per Mile.

Hydraulic Radius, r, in Feet.	Coefficients n of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	90	78	68	60	54	44	37	30	22	17	14	12
·2	112	98	86	76	69	57	48	39	29	23	19	16
·4	136	119	106	95	86	72	62	50	38	31	25	22
·6	149	131	118	105	96	81	70	57	44	35	30	25
·8	158	140	126	114	103	88	76	63	48	39	33	28
1	166	147	132	120	109	93	81	67	52	42	35	31
1·5	178	159	144	130	120	103	89	75	59	48	41	35
2	187	168	151	138	127	109	96	81	64	53	45	39
3	198	178	162	149	137	119	104	89	71	59	51	45
4	206	186	169	155	143	125	111	94	76	64	55	49
6	215	195	178	164	152	134	119	102	84	71	61	54
8	221	201	184	170	158	139	124	107	88	75	66	59
10	226	205	188	174	162	143	128	111	92	78	69	62
15	233	212	195	181	169	150	135	118	98	85	75	68
20	237	216	200	185	173	154	139	122	102	89	79	71
30	243	222	206	191	179	160	145	128	108	95	84	77
50	249	227	211	197	185	166	151	134	114	100	91	83
100	255	234	218	204	191	172	158	140	121	108	98	91

Slope S = ·0002 per Unit of Length, = 1 in 5,000, = 1·056 Feet per Mile.

Hydraulic Radius, r, in Feet.	Coefficients n of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	99	85	74	65	59	48	41	32	24	18	15	12
·2	121	105	93	83	74	61	52	42	31	25	21	17
·4	143	125	112	100	91	76	65	53	40	32	27	23
·6	155	138	122	111	100	85	73	60	46	37	31	26
·8	164	145	131	118	107	91	79	65	50	41	34	29
1	170	151	136	123	113	96	83	69	54	44	37	32
1·5	181	162	146	133	122	105	91	77	60	49	42	36
2	188	170	154	140	129	111	97	82	64	54	45	40
3	200	179	163	149	137	119	105	89	72	59	51	45
4	205	185	168	155	143	135	111	94	76	63	55	48
6	213	193	176	162	150	132	117	100	82	69	60	53
8	218	198	181	167	155	137	122	105	87	73	64	57
10	222	201	185	170	158	140	125	108	89	76	67	60
15	228	207	190	176	164	145	131	113	95	82	72	65
20	231	210	194	180	168	149	134	117	98	85	76	68
30	235	215	198	184	172	154	139	122	103	89	80	73
50	240	220	203	189	177	158	143	126	108	94	85	78
100	245	224	208	194	182	163	148	131	113	99	90	83

TABLE OF COEFFICIENTS C FOR VARIOUS HYDRAULIC RADII (*continued*).

Slope S = ·0004 per Unit of Length, = 1 in 2,500, = 2·112 Feet per Mile.

Hydraulic Radius, r, in Feet.	Coefficients n of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	104	89	78	69	62	50	43	34	25	19	16	13
·2	126	110	97	87	78	65	54	44	32	25	21	18
·3	138	120	107	96	87	73	62	50	37	30	24	21
·4	148	129	115	104	94	79	68	55	42	33	27	23
·6	157	140	126	113	103	87	75	62	47	38	31	27
·8	166	148	133	121	110	93	81	67	51	42	35	30
1	172	154	138	125	115	98	85	70	55	45	37	32
1·5	183	164	148	135	124	106	93	78	61	50	42	37
2	190	170	154	141	130	112	98	83	65	54	45	40
3	199	179	162	149	138	119	105	89	71	59	51	45
4	204	184	168	154	142	124	110	94	76	63	55	48
6	211	191	175	161	149	130	116	99	81	69	60	53
10	219	199	183	168	157	138	123	107	88	75	66	59
20	227	207	190	176	164	146	131	115	96	83	73	66
50	235	215	198	184	173	154	139	123	101	91	82	75
100	239	219	203	189	177	158	143	127	108	96	87	80

Slope S = ·0010 per Unit of Length, = in 1,000, = 5·28 Feet per Mile.

Hydraulic Radius, r, in Feet.	Coefficients n of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	110	94	83	73	65	54	45	36	27	21	17	14
·2	129	113	99	89	81	66	57	45	34	27	22	18
·3	141	124	109	98	89	74	63	51	39	30	25	21
·4	150	131	117	105	96	80	69	56	43	34	28	24
·6	161	142	127	115	104	88	76	63	48	39	32	27
·8	169	150	134	122	111	94	82	68	52	42	35	30
1	175	155	139	127	116	99	86	71	56	45	38	33
1·5	184	165	149	136	124	108	93	78	62	50	43	37
2	191	171	155	142	130	112	98	83	66	54	46	40
3	199	179	163	149	138	119	105	89	71	59	51	45
4	204	184	168	154	142	124	110	93	75	63	54	48
6	211	190	174	160	149	130	116	99	81	68	59	52
10	218	197	181	167	155	136	122	105	87	74	65	58
20	225	205	188	175	163	144	129	113	94	81	72	65
50	232	212	196	182	170	151	137	120	101	89	79	72
100	236	216	200	186	174	155	141	124	105	94	85	77

Slope S = ·01 per Unit of Length, = 1 in 100 = 52·8 Feet per Mile.

Hydraulic Radius, r, in Feet.	Coefficients n of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
·1	110	95	83	74	66	54	46	36	27	21	17	14
·2	130	114	100	90	81	67	57	46	34	27	22	19
·3	143	125	111	100	90	76	64	52	39	31	25	22
·4	151	133	119	107	98	82	70	57	44	35	29	24
·6	162	143	129	116	106	90	77	64	49	39	33	28
·8	170	151	135	123	112	95	82	68	53	43	35	31
1	175	156	141	128	117	99	87	72	56	45	38	33
1·5	185	165	149	136	125	107	94	79	62	51	43	37

Hydraulic Radius, r, in Feet.	Coefficients n of Roughness.											
	·009	·010	·011	·012	·013	·015	·017	·020	·025	·030	·035	·040
	Coefficients C.											
2	191	171	155	142	130	112	99	83	66	55	46	40
3	199	179	162	149	138	119	105	89	71	59	51	45
4	204	184	167	154	142	123	109	93	76	63	55	48
6	210	190	173	160	148	129	115	99	81	68	59	52
10	217	196	180	166	154	136	121	105	86	74	65	58
20	225	204	187	173	161	143	128	112	93	80	71	64
50	231	210	194	181	168	150	135	119	100	87	78	71
100	235	214	197	184	172	153	139	122	104	91	82	75

To find the coefficient C, having the slope S, the hydraulic radius r, and the degree n of roughness. Turn to the division of the Table corresponding to the given slope, S. In the first column find the given hydraulic radius, r. In the same line with this r, and under the given n, is the proper value of C.

To find the actual, or the greatest permissible, degree n of roughness of channel, having the slope S, the hydraulic radius r, and either C, or the actual or required velocity, v.

If the velocity is given, and not C, first find $C = \dfrac{\text{velocity}}{\sqrt{\text{slope} \times \text{hydraulic radius}}}$. Turn to the division of the Table corresponding to the given S, and in the first column find the given r. In the same line find the value given, or just obtained, for C; over which will be found the required n.

To find the actual or necessary hydraulic radius, r, having the slope S, the degree n of roughness, and the actual or required velocity, v. Assume an hydraulic radius; and from the division of the Table corresponding to the given S take out the value of C corresponding to the given n and the assumed r. Then say

$$v = \text{C so found} \times \sqrt{\text{assumed hydraulic radius} \times \text{slope}}.$$

If this v is the same as the given velocity or near enough to it, take the assumed r as the proper one. Otherwise, repeat the whole process, assuming a new r, *greater* than the former one if v *is less* than the given velocity, and *vice versa*.

Manning's Formula is:

$$V = \frac{1 \cdot 486}{n} R^{\frac{2}{3}} S^{\frac{1}{2}},$$

the values of n being the same as for Kütter's formula. It gives nearly the same values of V.

Direct Measurement of the Flow of Streams and Canals.

Cross Section.—The basis of the direct measurement of the flow of an open stream or canal is the measurement and plotting of the cross section of the stream-bed with reference, in respect of heights and lengths, to firmly driven stakes. This having been done, even if one set of readings of water depths be taken at the same time, further depth readings can be taken, either by direct measurement from the stream-bed or, as may be more convenient in the case of a muddy bottom or a very swift current, by differences in the readings of water levels referred to the datum stakes. To a width which is that to which a wire may be stretched taut, such a wire furnished a base for measurements, including those to the bed. In the case of a wide stream, theodolites may be used, or boats. Direct measurement by wading is to be preferred where it is possible. Whichever of the following methods be employed, the number of measurements required in the width may be judged from the degree of difference between one velocity measurement and those next to it. It may be necessary to space the points or lines of measurement somewhat closely, (*a*) where a stream develops a reverse current near the bank; (*b*) where a well-defined main current flows through relatively slowly moving water.

Capacities of Canals: Mean Velocity.

The discharge or capacity of a canal or river is the product of the cross-sectional area and the mean velocity. The mean velocity is usually found from Kütter's formula (see p. 676), a coefficient of roughness, n, being chosen with regard to the nature of the soil and the probable behaviour of the canal with respect to silting and erosion. Since the formula takes account of the water slope or fall, and of the hydraulic mean radius,

$$\frac{\text{area}}{\text{wetted perimeter}}$$

every measurement or calculation of discharge by other means tends to fix the value of n for the particular conditions of the bed of the canal or stream. Small discharges may sometimes be measured directly in basins or symmetrical pools, and at notches or specially prepared weirs (pp. 672–674), but calculations for discharge over large weirs are themselves subject to disturbing factors and their value in the determination of n is limited.

For irrigation canals in favourable soils and working under favourable conditions the value $n = 0 \cdot 0225$ is considered suitable, and this value has been adopted in the design of very important works. For canals in less favourable conditions the value 0·025 is often adopted, and for canals distinctly below average conditions n may be taken as 0·0275. The value 0·030 may be applied to some rivers and to canals in 'defective condition,' whilst for channels which are much obstructed n may be taken as 0·035. The first two values above are those of most importance in irrigation engineering. For aqueducts of rubble masonry n may be taken about 0·017 to 0·020, and for ashlar and brickwork, 0·013.

Bazin's Formula.

Bazin's new, 1897, formula is often used for channels, especially for those which, in irrigation systems, are relatively small. It is derived from the general formula

$$V = c\sqrt{RS},$$

by making in:

$$\text{British units, } c = \frac{157 \cdot 6}{1 + \frac{m}{\sqrt{R}}} \qquad \text{Metric, } c = \frac{87}{1 + \frac{\mu}{\sqrt{R}}} \quad . \quad . \quad . \quad (19)$$

where c is the coefficient, and

$m = 0 \cdot 109$;	$\mu = 0 \cdot 06$	for smooth cemented surfaces.
$= 0 \cdot 29$;	$= 0 \cdot 16$	for ashlar and brickwork.
$= 0 \cdot 83$;	$= 0 \cdot 46$	for rubble masonry.
$= 1 \cdot 54$;	$= 0 \cdot 85$	for very smooth earthen channels, pitched, or partly pitched.
$= 2 \cdot 36$;	$= 1 \cdot 30$	for ordinary earthen channels.
$= 3 \cdot 17$;	$= 1 \cdot 75$	for canals encumbered with weeds or boulders.

Parker states ('The Control of Water,' Second Edition, 1925) that experience led him to use Bazin's formula exclusively, in accurate calculations. On p. 474 of that book he gives the results of comparisons of the two formulæ, founded on very numerous observations.

Direct Measurement of Mean Velocities.

The Use of Floats.—Floats of various kinds have been employed for the measurement of the velocities of canals and rivers, being sent down the reach so as to maintain as far as possible their proper positions in the cross-sections, the average velocity being calculated from float speeds and depths in each strip of the reach. Double floats are often used, the lower and larger float which is a little heavier than water, being held at a determined depth below the surface by a smaller surface float to which it is attached by a cord. Thus the velocities at different depths and at different distances from the centre-line of the stream can be measured. The essential difficulty in using these floats is that if the lower float has a specific gravity considerably greater than that of the water the surface float must be relatively large, and affect the result appreciably, while if the specific gravity of the lower float be but little greater than that of the water it is liable to be lifted towards the surface by small upward currents.

For the lower floats both cylinders and spheres have been employed; for instance, cylinders 6 ins. in diameter and 12 ins. long, or 6 to 8 ins. in diameter and 9 to 12 ins. long; and spheres 2 ins. and 3 ins. in diameter.

Various forms of surface floats have been used, including very thin discs of considerable area, intended to slip easily along the surface without seriously affecting the speed of the lower float. This form seems to be suitable in calm weather, while a deeper float of smaller surface area may be preferable when there is a breeze. Calm days are best for float observations of any kind, but if a river or canal flows with or against prevailing winds of considerable velocity observations on calm days may be to a small extent misleading.

The float in general use at the Great Ouse Catchment Board sites is a light pinewood rod, 9 inches long, weighted at one end so as to float with 8 inches submerged. This may, it seems, call for changed values of K, in the next table.

Rivers Ouse and Nar.—(W. M. Griffith.) Maximum surface velocities in ft. p.s.: by meter (m) and by float (f), percentage difference (d). (1) m, 1·96; f, 2·29; d, 14·4. (2) m, 1·66, f, 2·01; d, 17·3. (3) m, 2·30; f, 2·60; d, 11·50. (4) m, 1·65; f, 1·64; d, 0·6. (R.M., 681–683.)

Rod Floats.—For canals, and for reaches of rivers in which the water varies but little in depth, a good form is the rod float, which may be a tube weighted at the bottom so as to float upright, or a wooden pole similarly weighted. In the Ganges canal experiments both one-inch poles and one-inch tubes were used, the latter proving to be the better. The rods should float with their bottom ends just clear of the bed of the canal.

Correct the observed velocity and reduce it to the mean velocity by using the following formula, due to Francis:

$$V_m = V_r\,(1-0{\cdot}116\,(\sqrt{d}-x)), \qquad (20)$$

where V_m = mean velocity for the vertical section considered;

V_r = observed velocity of the rod float;

$d = \dfrac{\text{distance between bottom of rod and river bottom}}{\text{depth of river}}$;

x is a function of $\frac{l}{d}$, l being the length of the rod and d the depth of the water.

The following shows the values of x corresponding to various values of $\frac{l}{d}$:

$\frac{l}{d}$	0·9	0·8	0·7	0·6	0·5
x	0·10	0·05	0·00	−00·5	−0·10

When $\frac{l}{d}$ = 0·9 or more, the formula is that given by Francis:

$$V_m = V_r\,(1-0{\cdot}116\,(\sqrt{D}-0{\cdot}1)) \qquad (21)$$

Current Meters.—Current meters are often used for measuring the velocity of streams and canals, and must be preferred to floats when there is no reach of sufficient regularity and length for the successful use of the latter. With a current meter the velocity at a particular cross-section may be found, and the measurement of that cross-section will then suffice, instead of the number which the irregularity of a reach may render necessary. It is not always easy in practice to keep the current meter at the desired depth, but in some cases it is held there by a rod. In a sluggish stream, the meter may be towed behind a boat, the speed of which is measured. Current meters should be calibrated, and from time to time tested, by towing them at a measured speed in still water, or, failing that, up and down the same reach of the canal or river. The mechanism of the meter can be stopped and started by a brake, on the stop-watch principle, this brake being worked by a cord from the boat. But some meters are fitted with an electrical apparatus, the wires of which form part of the tow line, and record the revolutions visibly in the boat.

The Pitot Tube.—If a tube be partly immersed, vertically, in a current of water, the lower end being suitably bent and remaining open, the water in the tube rises above the surface of the flowing water. The Pitot tube, designed on this principle, gives approximately the relation, $V = \mu\sqrt{2gh}$, where V = the velocity of the current, and h is the height of the water in the tube above the surface of the stream. Instruments which have been employed have given the results $V = 1{\cdot}7\sqrt{h}$. A much more elaborate instrument on the same principle (*Darcy*) has also been employed, and has given good results in small channels with fairly high velocities.

Perrodil's Hydrodynamometer.—This instrument is a device for measuring the velocity of a current, by the torsion which is produced in a wire by the pressure of the water against a disc mounted on an arm at right angles to the wire. The velocity is found by the equation $V = c\sqrt{a}$, where a is the angle of torsion, and c a coefficient found by experiment, or calculated from the coefficient of torsion of the material of the wire. The instrument is very sensitive and can be used for very slow currents.

Mean Velocity in a Vertical.—The mean velocity of the water in a given vertical in the cross-section of a stream is also the actual velocity at some proportion of the total depth in that vertical. What this proportion is will depend upon the nature of the stream, and will not be the same for all the verticals. For streams of considerable width and not very variable depth at the cross-section, measurements, by means of a current meter. of the velocities at the assumed depth of mean velocity may yield a fair approximation to the mean velocity of the stream, and a closer approximation may be obtained by making preliminary tests, with rod floats for instance, to determine the depth of mean velocity in the particular case. Values given usually lie between 0·60 and 0·75, or 0·60 and 0·70 of the depth.

Relation between Mean and Maximum Velocities.—For a rough estimation of the velocity of a stream the maximum surface velocity may be measured and the mean velocity deduced from it. If V be the maximum velocity, the mean velocity is KV, where K is a coefficient usually ranging from 0·60 to 0·80. In the following table values of K are given for earth channels and rubble masonry channels.

VALUES OF 'K' IN THE EQUATION 'V MEAN = KV MAXIMUM.'

Hydraulic mean depth (feet)	1	2	3	4	5	6	7	10	15	20
Rubble masonry channels	·77	·79	·80	·81	·81	·81	·81	·82	·82	·82
Earthen channels	·65	·71	·73	·75	·76	·77	·78	·78	·79	·80

Safe Bottom Velocities.—According to Bazin the relation between the bottom and the mean velocities in channels is the following:

$$V_b = V - 10{\cdot}87\sqrt{rs} \qquad (22)$$

where,

V_b = bottom velocity in ft. per sec.; r = hydraulic radius;
V = mean velocity in ft. per sec.; s = hydraulic gradient or slope of channel.

The following table, calculated from the foregoing formula, gives the safe bottom velocities for different materials:

TABLE OF SAFE BOTTOM VELOCITIES.

Material of Channel.	Safe Bottom Velocity (V_b) in Ft. per Sec.	Mean Velocity (V) in Ft. per Sec.
Soft brown earth	0·249	0·328
Soft loam	0·499	0·656
Sand	1·000	1·312
Gravel	1·998	2·625
Pebbles	2·999	3·938
Broken stone, flint	4·003	5·570
Conglomerate, soft slate	4·988	6·564
Stratified rock	6·006	8·204
Hard rock	10·009	13·127

RECENT LITERATURE.

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SECTION XVIII

PART III

EARTHEN AND MASONRY DAMS—TYPICAL AND NOTABLE WORKS — EXPERIMENTAL DETERMINATIONS — RECENT PRACTICE.

(Revised by R. A. Ryves, M.Cons.E.)

THE MAIN TYPES.

Earthen Dams.

The most widely adopted form of this type, and one employed in the full range of heights for which earthen dams have been built, has an impervious core wall and trench.

Instead of the core wall, and especially when the dam is built by the hydraulic method, a layer of impervious material, usually protected by pitching, may be provided on the upstream face.

Gravity Masonry Dams.

Suitable for any height up to, say, 200 ft., or to greater heights in narrow gorges. May or may not be economical for small heights.

Multiple Arch Dams.

(1) *Parallel Sluice Dams.*—The design is the same as for a liquid of specific gravity depending upon the proportion of opening to solid, and the calculations for stability can be thus made as for a gravity dam and water. The arches are vertical, or nearly so, a slight batter allowing of their weight being included in the calculations for stability. A river barrage, with sluice-gates replacing the arches, is an example of the type, though the form of the piers is changed, to provide a horizontal top, or one at two levels, and to suit conditions in respect of flow.

(2) *The Thrust Buttress Dam* (*Ryves*).—Designed to transmit the water load directly to the foundations. This type allows of precise computations of stresses being made and for two almost uniform stresses from base to crest, one in the arches and one in the buttresses, or for the same stress in both. American, and some other, practice in the building of dams of the type next to be described has for many years gradually approached this type, but no example of the full adoption of direct thrust has yet come to notice.

(3) *The Usual Multiple Arch Form.*—The base is of such a width and the water face so inclined, that, while the water load up to some level may be considered as directly transmitted to the foundation—though the dam is not usually so built that direct thrust is ensured—the upper part is a gravity dam, depending on weight-moment to resist water-load moment. There have been

many serious defects, and some extravagances, in the designs of such dams. For quite moderate heights they are economical. An almost common defect is that the base is not wide enough, or the water face is at too steep a slope.

THE SINGLE ARCH DAM.

Where, as in most gorges which provide promising sites for high dams, the width of the gorge at depths such as 200 ft. below the water level of the reservoir is small, this type is suitable for great heights, and is the only type economically applicable to very great heights, unless, perhaps, where the conditions favour the building, with materials from the adjacent strata, of a very wide-crested or wide-bermed dam of the earthen type.

The weight of a single-arch dam and its resistance as a wall are taken into account in computations of stresses, and the type may be regarded as in two divisions, thin arches and thick arches.

Gravity Dam Sharply Curved in Plan.—The border line between this form—including dams considerably but not sharply curved—and thick, single-arch dams is not definite, for the degrees of reliance upon weight and upon arch resistance depend on the materials employed, and other conditions.

Gravity Dams slightly Curved in Plan.—That slight curving adds to the strength of the dam is disputed. That it adds directly to the stability, if but a little, is hardly to be disputed. That it reduces stresses due to changes of temperature and tends to watertightness cannot be disputed. It is important to remember that where hard and firmly resisting strata, through which the river valley has been carved, extend only a little way upstream and downstream of a centre line drawn across the site, curving in plan, or the building of any single-arch dam, may have to be rejected in favour of a straight dam.

Tangential Gravity Buttresses.—The width, measured upstream and downstream, of the firm strata may suffice for the lower part of a single-arch or a curved gravity dam, while the curving in the wider part of the cross-section, towards the top, brings the ends of an arch downstream of the firm strata or, in the case of an arch of small radius, the ends may meet firm rock, but at an angle inconvenient for making springings. Any of these conditions is met by designing the dam with an arched portion about the middle, usually, of the upper part of the dam, and providing, up to crest-level, walls which play the part of gravity buttresses in a curved gravity dam, or of tangential struts in an arch dam. In the latter case additional walls may be provided upstream, to relieve these struts of the bending effect of water load. See Tujunga Dam and Grimsel Dam, *Engineering News-Record*, Aug. 6, 1931, and Jan. 19, 1933 Grimsel Dam *Civil Engineering* (London), July 1934.

Earthen Dams.

The top width is usually in the range 6 to 20 ft., very often about 10 ft., and, for high dams, seldom less than 15 ft. The upstream slope is often 1 in 2½; if there is a berm it may be 1 in 2 above the berm and 1 in 2½ below it. The downstream face is often 1 in 2½ to 1 in 3. These figures are, however, almost meaningless, for the dam has to be designed for stability of the upstream slope under water, requiring perhaps 1 in 2 for a pitched slope or 1 in 3 to 1 in 4 for an unpitched slope, also for the hydraulic gradient, which depends on the nature of the earth and on whether there is a core wall.

Watertightness is generally obtained by means of a core of puddled clay in the centre of the bank, the puddle being carried right down through the solid ground below the bank and well bonded into the impervious material on which the reservoir is built.

If pure clay be employed for the puddle, it must never be exposed to evaporation or to drying by capillary action. It is, therefore, often good practice to add sand to the clay, or some earth which will give a proportion of about 20 per cent. of sand in the mixture. Some engineers prefer an even larger percentage. It is important that there be no right angles in the cross-section of the puddle wall or trench, as these, or other irregularities, produce fissures in the puddle. (*Parker.*)

The puddle wall should generally be about 4 ft. thick at the top and both sides batter outwards about 1 in 40 to ground level, below which the thickness of the puddle should be quickly reduced to about 6 ft. and carried down at this as far as necessary. This applies to good puddle; the thickness must be increased if the puddle is of poor quality.

The puddle is often replaced by a concrete wall; care must, however, be taken to avoid the possibility of unequal stresses upon such a wall, as these might lead to cracking and possible failure.

Another method is to fill the trench in the solid ground with concrete and to build on this a wall of puddle. In this case the puddle must be well tongued into the concrete so as to ensure a watertight joint.

Where the earth available for the dam is of varying quality it is usual to place selected earth next to the puddle trench and to use ordinary earth for the outer portions. On this principle, and employing the most clayey material in the middle of the cross section, earthen dams are successfully built without core walls. A trend of practice is to locate the impervious prism with its centre of mass well upstream of the crest vertical; but not near enough to the water face for the clay, expanding when a depleted reservoir is being filled, to overcome the resistance, to shear or bursting, of the material between it and the water face.

Distribution of Materials.—The more and the less pervious materials must not be so distributed, generally or locally, in layers or in radiating prisms, that water will be trapped so as to develop uplifting or bursting pressure, or seepage so concentrated as to cause erosion.

Placing.—It is usual to place the material in layers inclined, slightly or considerably, towards the middle of the dam.

Compacting.—The materials are compacted by various means, trampling by the labourers, as in India, or by animals, having proved effective in the past. Punning has been employed, but earthen dams are now often consolidated by rollers or, to an increasing extent, by the machines used in their construction. The use of water is often desirable, but excess water should be squeezed out, layer by layer. Layers should not exceed 12 ins. in thickness, or with some materials 6 ins.; generally, the thinner the layers the better the compacting.

***Uniformity.*—Especially when rollers are used, it is important to remember that, while the dam will eventually consolidate itself by settlement, unequal distribution of solid content as between one cubic foot and another can never be put right after it is built. Especially when consolidation is by rolling, stubborn humps should be scraped away, not pressed down, and parts that seem somewhat hollow should be tested for resistance, and, where necessary, replenished before the next layer is spread. The engineer should study the manner in which the work is being done, reducing the thickness of the layers in proportion as the spreading and equal compacting are lacking in thoroughness.**

Compaction of Cohesionless Soils by Explosives.—In a paper, *Proc. Am. Soc. C.E.*, May 1941, A. K. B. Lynman describes how explosives were used to consolidate deposits of loose, fine sand—on which one end of an earthen dam was to be founded—in order to avert danger of 'liquifaction caused by earthquakes, demolition, or blasting in the vicinity.' The best consolidation was with charges of 8 lb. of 60 per cent. dynamite at 15 ft. depths. Twenty-one charges at that depth, distributed in four coverages with final grid intervals of 10 ft., produced 1·25 ft. final settlement and 36 per cent. degree of consolidation. Another test, with 5 coverages and final grid intervals of 5 ft., produced 2 ft. ground settlement and compaction to an estimated depth of 20 ft. Degrees of compaction, per cent., before and after blasting were: at depths, 5 ft., 18–47; 8 ft. 14–55; 10 ft., 26–70; 12·5 ft., 37–60; 15 ft., 35–53. Tests at other sites are described. (Abstract, *The Surveyor*, August 15, 1941.)

The Hydraulic Fill Method.

(From 'Earth Dam Projects,' by J. D. Justin.)

Definitions.

***Hydraulic Fill Dam.*—An earth dam in the construction of which the materials are transported** on to the dam by water and distributed to their final position in the dam by water. Hydraulic fill dams have not been built in British or Overseas British practice, and the method seems to be discredited in the United States.

***Semi-Hydraulic Fill Dam.*—An earth dam in the construction of which the materials are transported on to the dam by some other means than water and dumped within the section of the dam, some of this material being moved to its final position by the action of water.**

In hydraulic fill construction, water under high pressure is directed by means of a large nozzle mounted on a ball-and-socket joint, against the bank which is being excavated, the jet undercutting and then breaking up the materials, which then, mixed with the water, are guided to flumes, or pipes, through which they flow to the dam and are deposited.

Rock Fill.—When an embankment or, as in a greater range of useful dam construction, **a part or parts of an embankment, are occupied by large stones or pieces of rock, or such materials including large pieces, the spaces so occupied, or the materials so placed, are described as 'rock-fill.' The parts of an earthen dam near the toes, or one of the toes, are often thus constructed, to increase stability, assist drainage, and prevent erosion from tail water, or from low water wave action on the reservoir side of the embankment. The base of the rock mound thus provided may** be as much as 50 ft., width of cross section. Rock fill dams have largely been built in the United States. A number have been built in North Africa.

MASONRY DAMS

Rankine gives for low masonry walls to retain water that the thickness at the base should be about $\frac{6}{10}$ of the height either for rectangular walls or for those of trapezoidal form where the thickness at the top is $\frac{1}{2}$ that at the base.

Another rule for low dam walls is: Thickness at bottom 0·7 height; thickness at middle 0·5 height; thickness at top 0·3 height.

For high masonry dams Molesworth gives H = height of dam in ft.; x = any depth below the water surface in ft.; y = offset from vertical line to outer face of dam at any depth x; z = ditto ditto to inner face; b = width of dam at top in ft.; a = width of dam at 1/4H from top in ft.; P = limit of pressure on masonry in tons per sq. ft.

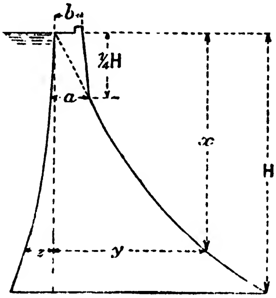


FIG. 1.

$$b = 0{\cdot}4a\,; \quad y = \sqrt{\frac{\cdot05\,x^3}{P + \cdot03\,x}}\,; \quad z = \left(\frac{\cdot09\,x}{P}\right)^4;$$

y at any section must not be less than 0·6 x, or x in fig. 1.

In modern practice a common typical section is triangular, the inner face being given a slight batter, and the back a batter which will generally approximate to 1 in 1½ but depends upon the weight of the masonry and the exact form of the wall. To some extent this form has been adopted in British practice, but the normal French and British section is usually, in British practice, adopted for high, straight dams of considerable length.

The graphical method of computation, proceeding from crest to base, with the given crest width to start with and adjustment to the within-the-middle-third rule, layer by layer, gives a section such as fig. 1, and corresponding closely with sections arrived at by other and recently devised methods of design.

The dam must be so proportioned that the resultant thrust acts within the middle third of the base, both with the reservoir full and empty; this provision should also apply to any horizontal section of the dam. The theoretical apex of the triangle must be high enough to allow, if necessary, for the water overtopping the wall, also for wave action.

Uplift.—A usual rule for taking uplift into account is to assume that the water pressure at every level may be exerted, in a degree diminishing from the head to zero, to a distance beyond the water face measured to the point at which the vertical pressure on the masonry is equal to the water head. It must be remembered, however, (*a*) that the maximum uplift is that due to a horizontal crack at the base, or seam in the rock foundation; (*b*) that, as regards the dam itself, the full theoretical uplift is reduced in proportion to continuity of solid material across the crack; (*c*) that, subject to the effect of (*b*), no higher crack affects the total uplift force as exerted by the lowest crack; (*d*) that if building in courses be avoided, it is not likely that the uplift will approach the full theoretical value, especially if watertightness in an increasing degree towards the base be attained, and (*e*) that if the face be fairly impervious, the material beyond it less pervious and decreasingly so as the downstream face is approached, there will be no uplift.

Maximum Stress at Down-stream Toe.—The pressure exerted vertically in any layer near the down-stream toe being designated by p, the maximum stress is usually computed as p sec.^{2} α (*Bouvier*) or p sec.^{2} ϕ (*Unwin*), where α is the angle between the resultant and the vertical, and ϕ the angle between the face and the vertical. (These stresses, as computed for the Cauvery Metur Dam, before it was decided to make it higher, are given with a drawing of the cross-section in *The Engineer*, February 24, 1911.)

THE CROTON DAM.

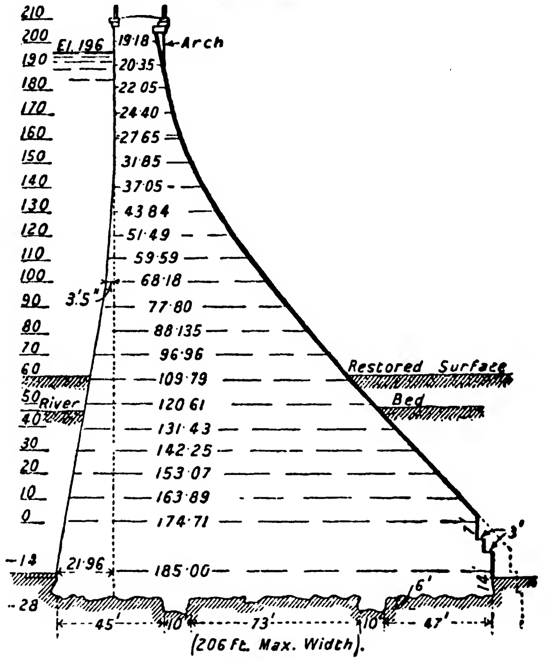


FIG. 2.—Cross-section of Croton Dam.

The cross-section of the Croton Dam (fig. 2) fairly accords with French and British practice (Chatrain, Periyar, Cauvery Metur, etc.), but it is probable that a straight line would be adopted for the downstream face if a dam were built under the same conditions to-day, in the United States. The straight line is now preferred by some British engineers.

CAUVERY METUR DAM.

Catchment, 16,700 sq. miles; reservoir, waterspread, 59·25 sq. miles, effective capacity, 93,500 million cu. ft.; minimum flow of the Cauvery, 800 cusecs, average flow about 20,000, maximum flood, 456,000 cusecs; newly irrigated area, about 320,000 ac. The dam: length of crest, 5,300 ft.; height above average riverbed level, 176 ft.; maximum height above deepest foundations, 214 ft.; volume of masonry in the dam, 54 million cu. ft.; basic economy, 1731. The basic economy would be higher but for the abrupt ending of the reservoir at a waterfall.

The Aswan Dam.

The Aswan Dam, figs. 3 and 4, is a granite rubble masonry structure founded on a similar rock which is traversed by dykes of softer igneous material. It is situated at the head of the first cataract of the Nile.

As originally constructed it was just under two kilometres in length, the top level of the structure was R.L. 110·00, the water being held up to a level of 106·00.

It is pierced for the greater part of its length by sluices which are closed by steel gates, most of them working on the Stoney principle. There are 180 sluices in all, of which 22 are at a level of 100·00 and 18 at 96·00; these sluices are 3½ metres high by 2 metres wide. Of the others, which are 7 metres high by 2 metres wide, 75 are at R.L. 92·00 and 65 at R.L. 87·50; 30 of the lowest sluices are lined with cast iron, the rest smooth-faced granite ashlar.

The original dam was finished in the summer of 1902.

It was found that the rush of water through the sluices was such as to erode what has appeared to be solid rock, and it was therefore necessary to build masonry aprons along practically the whole length of the pierced dam.

Between 1907 and 1912 the dam was raised so as to impound water to a level of 113·00 metres, and it was at the same time thickened by a blanket wall five metres thick on the downstream side.

The original reservoir impounded 980,000,000 cubic metres of water, or 34,608 million cu. ft.

The heightened reservoir (first heightening) contains 2,420,000,000 cubic metres, or 85,462 million cu. ft. See fig. 4.

The reservoir stores the flood waters of the Nile for distribution, from the barrages downstream, to the irrigated areas.

Second Heightening.—Successfully completed in 1934. In order to eliminate stresses due to settlement (including elastic compression) of the added masonry, as well as stresses due to shrinkage or temperature changes, the new work, other than the extension of the upper and nearly rectangular portion of the dam, is in the form of buttresses between which and the old masonry, dressed smooth, is a sheet of non-corrosive steel plating. This plating extends as far as the curve in the downstream face, along and above which there is a gap between the old masonry and the upper part of the buttress, the weight of which is thus carried by the lower part only. See fig. 5.

The original water maximum level was R.L. 106, with the roadway at 109; the first heightening gave water level 113 and roadway 114; the third heightening provides for water level 121 and roadway 123. The storage capacity is now about 4,800 million cu. metres, or 169,512 million cu. ft., and the normal summer supply has been increased by about 66 per cent.

Ryburn Dam.

(*J. N. Wood.*)

For the water-supply of Wakefield this dam impounds 220 million gals. Greatest height over foundations, 121 ft.; maximum water depth, 95 ft. Strata, alternating bands of millstone grit and shale, with a dip downstream. *Underdrains* are provided beneath the foundations in the middle of the valley. The dam is of mass concrete (so-called, really masonry), and is curved in plan. During the building of the dam the stream was diverted through a *wood-stave* pipe, 170 ft. long. Gritstone 'plums' of ¼ to 1 cu. yd. were embedded in the concrete, as well as smaller stones, the larger plums being not less than 9 ins. apart. *Cementation of fissures* involved the drilling of 151 holes through which 2,278 tons of cement were injected. At first the rock was grouted to a depth of 100 ft. below the cut-off trench, later to a depth of 200 ft. (*Inst. Water Engineers*).

Woranorah Dam.

New South Wales, on the Woranorah River. Dimensions: crest length, 1,280 ft., and width, 20 ft.; radius of upstream face at crest, 1,200 ft., maximum height above foundations, 216 ft.; maximum depth of water, 204 ft.; maximum depth of cut-off trench, 36 ft.; maximum breadth, 178·53 ft.; length of overflow weir, 570 ft.; maximum depth of excavation for weir, 110 ft.; length of weir channel, 1,250 ft.; full supply level, R.L. 552; crest level, R.L. 560; sill level of weir, R.L. 552; waterspread of reservoir, 937 acres. This dam impounds 15,000 million gals., or 2,400 million cu. ft., allowing of a daily supply of 20 million gals., or 3·2 million cu. ft. The area of the catchment is 29 sq. miles. The volume of masonry work is about 370,000 cu. yds. The approximate cost was £1,688,000. The whole of the foundation area was grouted to a depth of about 30 ft., the primary holes on a 40-ft. grid and additional holes at intervals of 20, 10, or even 5 ft., where necessary.

The Aswan Reservoir

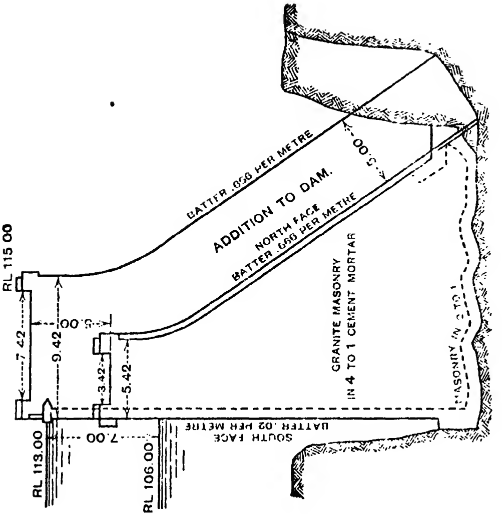


Fig. 4.—Cross-section of Solid Dam.

Mean High Flood Level at Aswan 92.80.
Mean Low Water Level..........85.10.

Contents of Reservoir-Original Dam 980,000,000$^{M^3}$
" " Heightened Dam 2,300,000,000$^{M^3}$

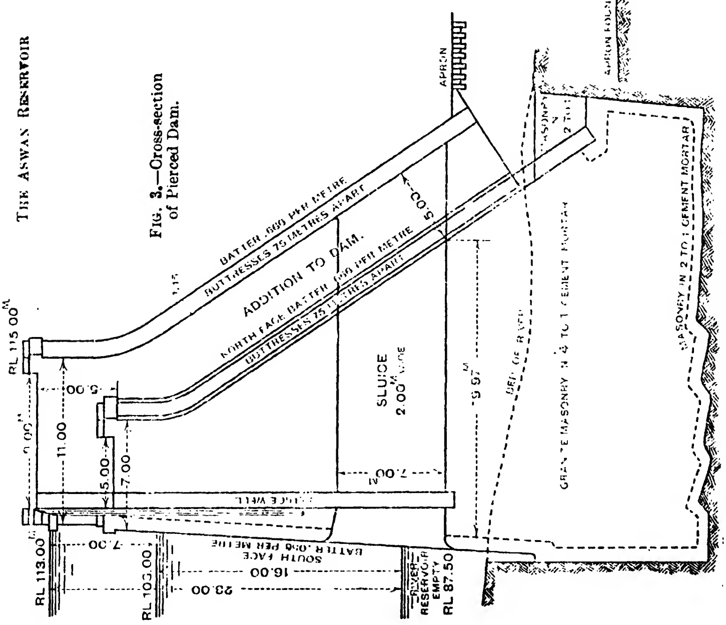
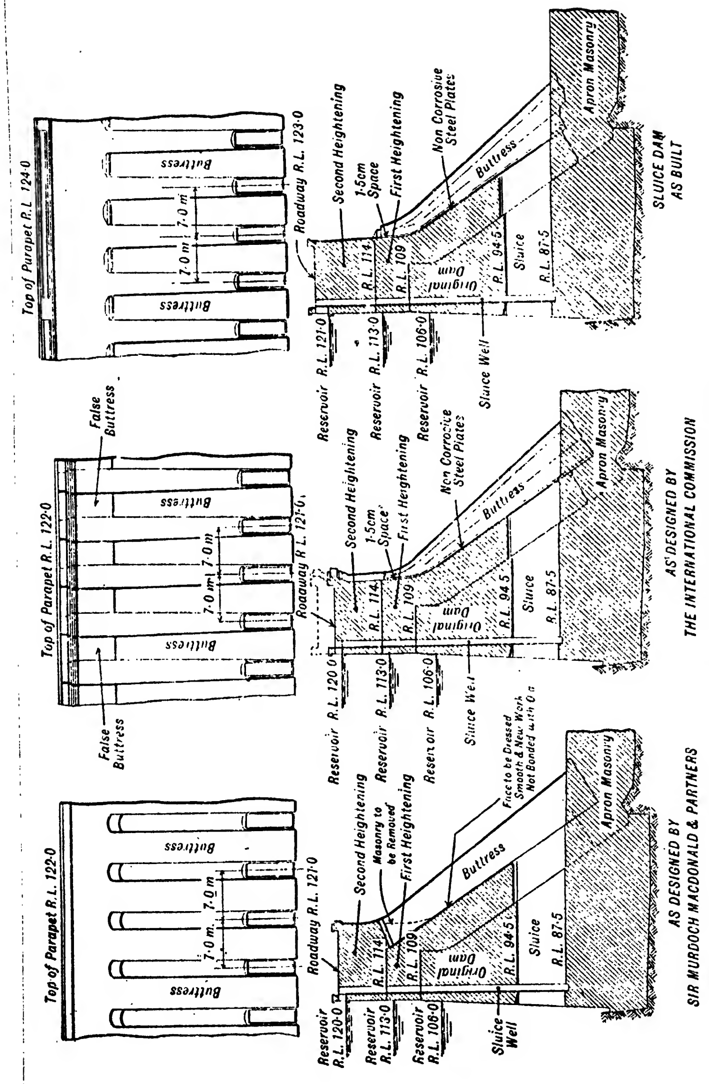


Fig. 3.—Cross-section of Pierced Dam.



"THE ENGINEER"

FIG. 5.—Second Heightening of the Aswan Dam.

Boulder Dam, California.

Boulder Dam, Colorado River. A concrete dam of gravity type, curved to a radius (axis) of 500 ft. Maximum height plus maximum depth of foundation, 726 ft.; crest, length about 1,180 ft.; top width 45 ft.; width at base, 660 ft.: volume of dam, 3·4 million cu. yds.; excavation in foundation, 400,000 cu. yds.; reservoir ('Lake Mead') 115 miles in length, area 227 sq. miles; capacity 30·5 million acre-feet, or 1,328,580 million cu. ft.; main canal for irrigation in the Imperial Valley, 200 ft. wide by 22 ft. deep. Maximum discharge, 15,000 cusecs; aqueduct capacity, 1,500 cusecs; hydro-electric plant, 1,200,000 h.p.

The Colorado River has a length of 1,700 miles, and drains a basin of 244 sq. miles. Its discharge at about 120 miles from its mouth ranges from 66 to 200,000 cusecs. The site of the Boulder Dam is about 370 miles from the mouth (not counting minor bends of the river). The reservoir receives about three-quarters of the run-off of the river basin. The four diversion tunnels were 15,905 ft. in aggregate length. The dam was artificially cooled by a net-work of 2-in. piping, totalling about 800,000 linear ft., permanently embedded in the concrete. (See *The Engineer*, January-June 1935, pp. 154, 170, 196, 222, 227.)

Laggan Dam.

The Laggan Dam, on the River Spean (Lochaber water power development), impounds 1,500 million cu. ft. of water from a catchment of 150 sq. miles. It is 700 ft. long and 170 ft. high from foundation to weir crest and is built in mass concrete; upstream face nearly vertical; downstream face in two straights, the lower curving at a radius of 44 ft. at the toe. The foundation, in rock, is sloped upwards to the downstream face and the concrete was similarly laid in layers not less than 3 ft. thick, at a slope of 1 in 12. (Descriptions, drawings and photographs, *The Engineer*, May 22 and 23, 1936, and *Civil Engineering (London)*, February and March 1937.)

Columbia River Project.—For water-electric power, ultimately 8 million kilowatts, utilising an annual run-off of 146 million acre-feet, 9·3 ins. from a catchment 1,210 mi. long and 259,000 sq. mi. in area. Height of proposed dams, feet: Grand Coulee, 430; Foster Creek, 200; Chelsea, 120; Rocky Reach, 100; Friest Rapids, 150; Umatilla Rapids, 330; John Day Rapids, 258; the Dalles, 150; Bonneville, 72.

Design of Spillway Dams.

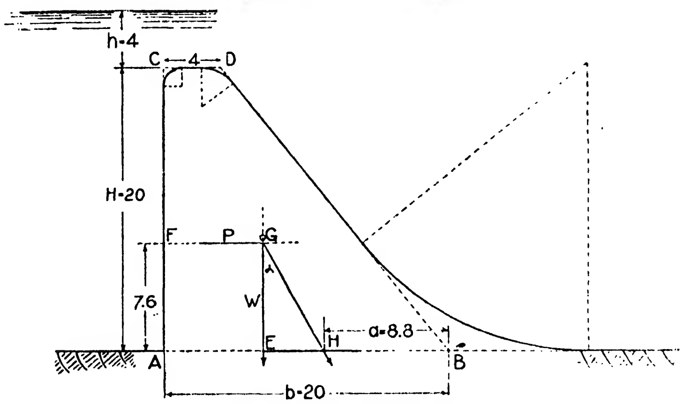


Fig. 6.—Design of Spillway Dam.

Spillway dams, or weir dams, are designed so as to allow the water to flow over their crest. The two first points to decide before beginning the design of such a dam are the elevation of the crest of the dam and the elevation of the highest possible flood.

Assuming that the elevation of the crest is 20 ft. above the bed of the river, and that the greatest flow of the river can be passed over the dam when the water level is 4 ft. above its crest, we can now proceed to design a plain concrete dam to meet these conditions, fulfilling at the same time the requirements as to stability enumerated above.

The width of the crest OD (fig. 6) can be taken as 0·2 of the height of the dam, and the width of the base b equal to the height H.

Referring to the figure we have the following:

$$A = \text{area of section of dam} = \frac{4+20}{2} \times 20 = 240 \text{ sq. ft.};$$

$$w = \text{weight of 1 cub. ft. of concrete} = 130 \text{ lbs.};$$
$$W = \text{weight of section of dam 1 ft. wide} = 240 \times 130 = 31{,}200 \text{ lbs.}$$

This force W acts vertically through the centre of gravity of the section G, whose position is determined graphically.

The water load P tending to overturn the dam has for value

$$P = 62\cdot5 \times \frac{(H+h)^2 - h^2}{2} = 62\cdot5 \frac{H}{2}(H+2h),$$

and in the case in question.

$$P = 62\cdot5 \times \frac{20}{2}(20+2\times4) = 17{,}500 \text{ lbs}$$

This force P acts horizontally, and is applied at a height AF above the bed of the river given by the following formula:

$$AF = \frac{H}{3} \cdot \frac{H+3h}{H+2h} = \frac{20}{3} \times \frac{20+12}{20+8} = 7'\cdot6.$$

Taking the moments about the downstream toe of the dam B we have the following results:

$$\text{Resisting moment} = W \times EB = 31{,}300 \times 13 = 405{,}600 \text{ ft. lbs.}$$

$$\text{Overturning moment} = P \times AF = 17{,}500 \times 7''6 = 133{,}000 \text{ ft. lbs.}$$

$$\text{Factor of safety} = \frac{\text{Resisting moment}}{\text{Overturning moment}} = \frac{405{,}600}{133{,}000} = 3\cdot05.$$

A factor of safety of 3 is considered sufficient to fulfil the first condition of stability.

The second condition is that there should be no danger of the dam sliding upon its base; this condition is fulfilled if we have the following inequality:

Weight of dam $\times$ coefficient of friction $>$ Total water load.

The coefficient of friction of masonry on masonry, or masonry or concrete on rock, is 0·7, so that we have

$$31{,}200 \times 0\cdot7 > 17{,}500.$$

The resultant R of the forces P and W can be found graphically or algebraically by the following equation:

$$R = \sqrt{W^2+P^2} = \sqrt{31{,}200^2+17{,}500^2} = 35{,}770.$$

Its direction can be found by the equation

$$\tan a^\circ = \frac{P}{W} = \frac{17{,}500}{31{,}200} = 0\cdot56;$$

and the point of application H found thus

$$EH = AF \tan a^\circ = 0\cdot56 \times 7\cdot6 = 4\cdot26.$$

This point of application H must fall within the middle third of the base; this condition must be fulfilled whether the reservoir created by the dam is full or empty.

It is advisable to round up the dam at the upstream corner O to prevent chipping by logs or floating ice. It is also advisable to give to the downstream face a form resembling as much as possible the probable curve of the overfalling water. Corner D should therefore be rounded up, and a concave curve built at B. This usually curves upward for some feet in height (see previous observations, Vyrnwy Dam), and terminates in a drop wall. The aim being to create a standing wave.

When the proposed dam is over 30 or 40 ft. it is better to subdivide it into a certain number of sections and to test each block separately according to the method described; the resultant in each case must fall within the limits of the middle third. In the example given it has been assumed that there was no upward water pressure at the base; when infiltration and under pressure must be provided for, this new force is subtracted from the weight and combined with it before calculating the resisting moment.

Features of Practice.

MARGINAL GROUTING OF A RESERVOIR.

Madden Reservoir, United States.—The cavernous nature of the limestone forming the impounding ridges made grouting desirable, and linings of loamy sand in caves and crevices made cement grouting impracticable. Clay grouting has been carried out, the best results being with grout containing 55 per cent. of water. Grout containing less than 48 per cent. of water could not be forced through a 1-in. pipe under a pressure of 100 lb. per sq. in. The distance to which the grout reached underground ranged from 13 to 50 ft., and seams only ⅛ in. wide were penetrated. (F. H. Kellog, *Eng. News-Record*, vol. 109, pp. 395–399; *Eng. Abst.*, vol. 54, No. 151.)

Exploration of Dam Site.—A valuable recent contribution to exploration methods is the large-diameter core drill, which permits the geologist or engineer to enter the hole and examine the exposed section of the rock. . . . Water may be controlled by preliminary grouting, through holes drilled either outside of, or within the circumference of the proposed large-diameter hole. (W. J. Mead, *Civil Engineering* (U.S.A.), May 1937. Extracts, *The Surveyor*, June 11, 1937.)

The Anchoring of Gravity Dams.—A recent development of considerable interest. Based on an article in *Annales des Ponts et Chaussées*, August 1935, a description and illustrations in *The Engineer*, February 21, 1936, show how Cheurfas Dam, in Algeria, was anchored—in order to add to the downward pressure—by vertical cables carried through sandstone, limestone and, again, sandstone, to argillaceous marl, at a depth about 30 m. below river-bed and some 20 m. below the bottom of a concrete base on which the dam—which is 22 m. high—stands. The cables have caps bearing on the crest of the dam and were sealed at and near their lower ends. Each is composed of 630 steel galvanised wires, laid parallel and sheathed with a 'sandwich' of grease and bitumen between wrappings of sail-cloth.

Heightening a Dam.—To keep the line of resultant pressure within the middle third, while raising the water level by 3 ft., the spillway of the Mahadeo Nala Dam, Bombay Presidency, was raised by forming on the existing crest, a reinforced concrete block, 6 ft. wide and 5 ft. in upstream-side depth, with an upstream overhang of about 18 ins. (G. O. Minnitt. Paper No. 5023, Inst. C.E.)

ARCH DAM: MEASURED STRAINS UNDER WATER LOAD.

Stevenson's Creek Dam, California.—A concrete dam built for the purpose of measuring strains under water load and comparing deduced stresses with calculated stresses. Founded on solid rock, in a gorge. Height 60 ft., crest length 140 ft., upstream radius 100 ft., thickness at base 7½ ft., at crest 2 ft. Tests were made at night to minimise temperature effects. At 30 ft. below crest the maximum deflection under full load, 60 ft. water depth, was 0·378 in. At the crest the deflection was downstream at the crown and upstream between the quarter points and the ends. Computed for a modulus of elasticity of 3,600,000, the stresses were: at the crown, 260 lbs. per sq. in. (theoretical 207); at the abutments, 900 to 1,000 lbs. per sq. in. (theoretical 879). Considered as assigned in part to arch resistance and in part to vertical elements resistance, the share of the latter in sustaining the water load seems to have been less than that assigned by theory. This may have been due to the formation of a horizontal crack at the base of the upstream face between the dam and its foundation, which appeared when the water had risen to 30 ft. depth. The formation of this crack is in accordance with the theory regarding the design of the foundations at and near the upstream toe of a dam, according to which the dam should not there be keyed into its foundation. [See 'Mass Dams and Dams of Arches' (R. Ryves): Madras Government Paper, 1910.]

MEASURED SHRINKAGE OF AN HYDRAULIC FILL DAM.

(*C. H. Eiffert, 'Engineering News-Record, Jan.* 27, 1927.)

Huffman Dam.—Height of dam above ground level, about 62 ft.; width of crest, 32 ft.; symmetrical to central vertical plane; two berms; both slopes, from toe, 3 to 1; between berms, 2½ to 1; to crest, 1·825 to 1. Shrinkages, including settlement of foundation which is believed to have been very small: at 40 ft. below crest level, 0·40 to 0·68 per cent.; at 30 ft. below crest level, 0·31 to 0·54 per cent.; at 20 ft. below crest level, 0·44 to 0·84; at 10 ft. below crest level, 0·24 on east side and 0·73 on west side; at the crest, 0·60 on the east side and 0·49 on the west side; average, 0·51 per cent.

DIMENSIONS OF SLUICES.

Lake Mentz Dam, Sundays River, South Africa.—Five free roller sluice gates, each 30 ft. wide by 25 ft. high; about 25½ tons weight; counterpoise, 53 tons; designed for water head of 26 ft. Each gate is built up of latticed bowstring girders as the span elements.

Sennar Dam, Blue Nile.—The portion called the 'sluice dam' is 606 m. (1,988 ft.) long, with 80 sluices 2·0 m. wide by 8·40 m. high to the springers (6·56 ft. × 27·55 ft.). The head regulator has 14 sluices, each 3·0 m. wide by 5·0 m. high to the springers (9·84 ft. × 16·4 ft.); seven to be provided with gates and seven (held in reserve) blocked.

Automatic Syphon Spillways: Early and Recent Examples.

Champlain Canal, U.S.A. (G. F. Stickney, 1910).—At lock No. 9 the syphon spillway was 57 ft. long as against 200 ft. if an ordinary weir had been built for the small permissible depth on sill. Each syphon had an area of 7¾ sq. ft.; head, 10½ ft.; discharge, 160 cusecs; flared inlets. At Whitehall, on the same canal, a similar weir, 6 syphons; 16 ft. head. These syphons were formed in the body of the weir. (See *Engineering News*, October 13, 1910.)

Maramsilli Reservoir.—A syphon spillway to pass 40,000 cusecs. (See Paper No. 4602, *Proceedings Inst. C.E.*, by Powys Davies.)

Bridgeport Dam, Nevada, 1926.—Cross-section, 3 ft. by 6 ft.; air vents in syphon hoods at different elevations, so that the syphons will come into action one at a time.

Tests of Syphon Weirs (J. T. Corwin and A. W. Kidder).—The tests showed that the lower end of an inclined syphon barrel should be sealed, by a pool or by a vertical bend, the former being preferable, as it does not give rise to pulsation. When a syphon has to work on a very small rise in the water level, the provision of an auxiliary priming weir seems to be justified. In respect of the vacuum needed for priming, a low throat section is advantageous. Tests indicated that the ends of down-turned vacuum breaker pipes, with an area 6 per cent. of the throat area, should be about 2 ins. above the water level at which it is desired to stop the syphon. The coefficient of discharge was, by a single test, 0·69 (*Eng. News-Record*, vol. 108, pp. 649–652; *Eng. Abs.*, vol. 53, No. 255).

Tummel Development.—Perthshire. The Hydro-Electric Power Works of the Grampian Electricity Supply Company. In a concrete spillway 70 ft. long and about 60 ft. high; four syphons, two having throat dimensions 6 ft. wide by 3 ft. high, the suction heads being 16 ft. 9 ins. and 22 ft. 3 ins.; two having throat dimensions 8 ft. wide by 4 ft. high and a suction head of 23 ft. The priming is automatic and occurs when the water level is, above the throat, 5½ ins. in the smaller and 7½ ins. in the larger syphons. Capacities, the smaller syphons, together, 850 cusecs; the larger, together 1,700 cusecs. (Fully described and illustrated, a dimensioned drawing included, *The Engineer*, July 6 and 13, 1934.)

Shing Mun Dam.—To supplement the flow over the bell-mouthed weir: six 4-ft. dia. syphons have been constructed; of the type which has a separate air pipe to the throat for the control of priming; these pipes being so set that three of the syphons prime for 6 ins. depth on the weir cill and three when the depth is 12 in. Each syphon has a discharge capacity of 500 cusecs.

Brent Reservoir.—Area draining into the reservoir, 19,000 acres, or nearly 30 square miles. Syphons provided as an additional means of discharge in case the sluices do not suffice to carry the whole flow. The five syphons are of reinforced concrete. The lip of the hood (of each) reaches to a depth below crest varying from 7 ft. 6 in. to 8 ft. The area of the opening into the hood is about 82 sq. ft., reducing to 21 sq. ft. (7 ft. wide by 3 ft. high) over the crest and downward to 12 ft. 9 in. below it, whence it reduces, in a length of 6 ft. 6 in., to a circular section of 4 ft. 6 in. diameter, or 15·9 sq. ft. cross sectional area, maintained to the outlet. (Description and drawings in *Civil Engineering* (*London*), August 1937.)

Laggan Dam.—Six syphons, each having a discharge capacity of 600 cusecs, are embodied in the concrete of the dam. The hood and throat is of steel, the remainder being formed in the concrete, except the outlet, which is a cast steel bend, throwing the issuing jet slightly upwards so as to clear the toe of the dam. The toe is at, about, elevation 690 ft., four of the jets at 755, and two at about 765, the crests of the syphon throats being at 822, maximum flood level. (See 'Syphon Spillways,' by A. H. Naylor; a book. The syphons are shown, in a section of the dam, *The Engineer*, May 29, 1936.)

Bellmouth Spillways.

Ladybower Reservoir.—Derwent Valley Water Board. There are two identical weirs, The rim is 80 ft. in diameter, and the shaft curves to a horizontal tunnel 15 ft. in diameter. To prevent whirlpool formation, 12 cutwaters are disposed radially in the lip and guide walls beyond them.

Burnhope Reservoir.—Durham County Water Board; opened in September 1937. The provisions for overflow comprise a weir about 200 ft. long with its crest 7 ft. below the top of the embankment, and a bell-mouthed swallow-hole, 50 ft. in diameter at the top, reducing in a depth of 25 ft. to a vertical shaft 12 ft. in diameter, which connects, by a bend constructed of cast-iron segments, to a horizontal tunnel.

Dunwan Reservoir.—Eastwood and Mearns Water District of the county of Renfrew. A combined bellmouth weir and upstand tower. Overall diameter of bellmouth 33 ft., internal shaft diameter, 21 ft. Maximum depth over sill in an exceptional flood, 1 ft. 8 in. Internal diameter of upstand tower, 11 ft. The whole structure is of reinforced concrete, founded on rock. Described, with dimensional drawings and photographs, by J. Semple, *Civil Engineering* (*London*), January 1939.

Other Bell-mouth Weirs.—In a paper (*Inst.W.E.*) on Bell-mouthed Weirs and Tunnel Outlets for Disposal of Flood Water (Extracts, *Civil Engineering* (*London*), June 1937), W. J. E. Binnie furnished data of dimensions and velocities of flow relating to such weirs, as provided for the reservoirs: Taf Fechan, Silent Valley, Pontian (Singapore), Davis Bridge (United States), Burnhope, Jubilee (Hong Kong), and Falls Dam (New Zealand).

Laboratory Experiments.—In Paper No. 5236, *Journal Inst. C.E.*, January 1941, A. M. Binnie and R. K. Wright describe laboratory experiments on bellmouth spillways, one being a simple, or cone, funnel and the other a splayed funnel. Tests were made with orifices of different sizes, and both with and without a tailpiece.

The Danel Weir.—Described, with illustrations, *The Engineer*, May 5, 1939. The system consists of constructing a semi-circular or semi-elliptic drainage basin, the floor of which ends in a half funnel or half trumpet shape, so calculated that the water flowing out at the bottom can run directly into a canal or tunnel without the provision of any special transitional or connective works.

Dams Recently Built or Under Construction.

Earthen Dams.

Sardis Dam.—Under construction, 1939. In the Mississippi flood prevention scheme. Situated on the Little Tallahatchie River. Volume, 16,760,000 cu. yd., including 13,815,000 cu. yd. of hydraulic fill. Maximum height above river bed, 117 ft. Crest length about 2¾ miles. Reservoir: area, 92 sq. mi., capacity, 1,570,000 acre-ft.

Tal-y-Bont Dam.—Newport water supply. The reservoir has a catchment of 6,000 acres, and the average rainfall of the three driest consecutive years is (or was in 1933) 68½ ins. The expected yield is 15 million gals. per day. The capacity of the reservoir is 2,500 million gals. and the water spread 320 acres. The dam of the earthen type with a clay puddle core above ground and a 6-ft. concrete core from ground level, carried well into the solid rock. Dimensions are: length, 1,400 ft.; height above river bed, 97 ft.; the crest carries a 10-ft. carriageway; maximum width at base, 550 ft.; slopes, upstream, 2½ to 1 with a berm of 15 ft. at half height; downstream, varying from 2½ to 1 to 4 to 1. Both slopes have toe walls. The water slope is pitched with roughly squared stones on shingle. The overflow weir is 270 ft. long. Volume of dam, 560,000 cu. yd., including 40,000 cu. yd. of puddle clay. Described *The Engineer*, July 21, 1939.

Rio Salado Dam.—Don Martin Project, Mexico. An earthen and gravel embankment 3,231 ft. long with a concrete overflow portion of 768 ft. The embankment portion has the dimensions: greatest height 114 ft.; upstream slope 1¾ to 1; downstream slope 2 to 1, crown width 19·7 ft., freeboard 13·1 ft. It is revetted with a concrete slab, 11 to 8 ins. thick, reinforced with ⅜-in. rods, 12-in. square mesh.

Fort Peck Dam.—Completed in 1939; on the headwaters of the River Missouri. For power, the irrigation of 180,000 acres, regulation of flow for navigation, and mitigation of floods. Maximum height, 287 ft.; water surface at flood level to be raised to 220 ft. above river bed. Maximum width at base nearly 3,000 ft.; crest width 100 ft.; length 9,000 ft., or including a dyke portion, 20,500 ft. Volume of main dam, 92 million cu. yds.; of the dyke portion, 6·5 million cu. yds. The reservoir will be nearly 185 mi. long, with a maximum width of 17 mi. and a capacity of 19·5 million acre-ft. (See *Engineering News-Record*, August 29, 1935; *Civil Engineering* (U.S.A.), July 1936; *The Engineer*, June 4, 1935, April 6 and 23, 1937.)

Ladybower Dam.—Derwent Valley Water Board; completed September 1945. Length (l), 1,250 ft.; max. height (h), above river bed, 141 ft.; volume (v), about 900,000 cub. yds. Upstream slope, about 1 in 3, the lower part protected by a stone 'beaching' and the upper part by stone pitching. Downstream slopes (benched), upper part 2 in 3; lower part, 2 in 5.
The completed works were opened by His Majesty. The greatest width at base is 665 ft., crest width, 17 ft. The capacity of the reservoir is 6,300 million gals.

Hollowell Dam.—Northampton water supply. An earthen dam 1,407 ft. long, 43 ft. max. height and 368 ft. max. width, an increase of 30 ft. on the design width. The capacity of the reservoir is 460 million gals.; the area of the catchment, 2,500 acres, the average rainfall 26·5 ins.

Afton Dam.—Ayrshire; supply inaugurated, September 1935—a part of the Loch Bradan water district scheme. Catchment reaching to alt. 2,000 ft. Length, 600 ft.; h, 80 ft.; capacity of reservoir, 102·4 million cub. ft. Cost, including subsidiary works, about £100,000. (Ref. Paper by J. A. Banks; *Inst. W.E.*)

Schwammenaul Dam.—In Ruhr Valley, Germany. Completed June 1938. Height, 18 ft. 6 in.; base width, 984 ft.; crest width, 34 ft. 6 in.; crest length, 984 ft.

Deer Creek Dam.—A work included in the Provo River Project, Utah. Height above streambed, 155 ft.; base width about 840 ft., not including heavily pitched toe of 200 ft. in the cross dimension. Crest width, 35 ft.; length, 1,400 ft., between flanking canyon walls. Slopes: upstream, 1 in 3, downstream, 1 in 2½. The dam rests on sand, gravel and boulders, through which a trench about 80 ft. deep, 205 ft. wide at the original ground level, reducing to 30 ft., was excavated to rock, in which is founded a reinforced concrete cut-off 20 ft. in maximum height. The trench is filled integrally with the body of the dam, which is formed of clay, sand and gravel, rolled in 6-in layers: (*a*) in the upstream prism, graduated in coarseness to the upstream face; (*b*) in the middle prism—the base of which extends to some distance downstream of the crest—impervious material; (*c*) in the downstream prism, semi-impervious material, graduated in coarseness to the downstream face; (*d*) in the extended toe, sand, gravel and cobbles. The useful storage capacity of the reservoir will be 147,000 acre-ft. (Information and illustrations in a paper by E. A. Jacob and E. O. Larsen; condensed in *Civil Engineering* (*U.S.A.*), January 1939.)

Rock-Fill Dams.

Bakhadda Dam, Algeria.—Height, 148 ft.; crest width, 16 ft.; length 726 ft. Maximum size of rock, 3 tons. The impermeable shell on the upstream face is of r.-c., in two layers, the lower 12 ins. thick and the upper—laid 7 weeks later to allow for settlement under water load—15½ ins. (For reference and abstract, see *Engineering Abstracts*, vol. 73, No. 36.)

Oued-Kébir Dam, Tunisia.—Height, 115 ft., crest lenght. 1,095 ft., with a central impermeable wall of r.-c. The construction presents unusual features. (For reference and abstract, see *Engineering Abstracts*, vol. 73, No. 38.)

San Gabriel Dam.—Under construction, in San Gabriel Canyon, California. Stated to be the largest rock-fill flood control dam in the world, necessitating the blasting, quarrying, removal and placing of 10,478,000 cub. yds. of material. Estimated cost, £2,400,000. Average height above bed rock, 375 ft.; crest length, 1,540 ft.; downstream slope, 3 to 1, benched; capacity of diversion tunnel, 35,000 cusecs.

Long Valley Reservoir.—Mono Lake Project. Waterspread, 5,000 acres; capacity, 163,000 acre-ft. The rock-fill dam will have a welded steel face. (The project is described in *The Engineer*, July 3, 1936.)

A Composite Dam.—Loch Treig Dam, Lochaber water power development. Base width at maximum section, 255 ft. including a spill-receiving trough at the downstream toe, structural width, 238 ft. 10 in.; height from ground level to spillway crest, 37 ft. Upstream and downstream slopes 1 in 3. A middle concrete core-wall about 7 ft. thick at the top, widening to 10 ft. at the base, and carried down to 21 ft. below ground level, into rock. On the downstream side, next the concrete wall, a prism of hand-packed rubble about 18 ft. wide at the base, backed by tipped rubble, rolled in layers about 12 in. thick; facing 1 : 5 concrete on 18 in. of large gravel, quarry rid and spalls. On the upstream side of the concrete wall, a prism of selected clay, 3 ft. wide at the top and 18 ft. at the base, the remainder being earth fill, and the facing, 18 in. granite random rubble pitching, on the same base as on the downstream face. Sheet piles have been driven to rock at the downstream toe. Dimensioned Cross-section, *Civil Engineering* (*London*), February 1937.

Masonry and Concrete Dams.

GRAVITY TYPE.

MASONRY AND CONCRETE DAMS.

The Chambon Dam.—The largest dam in Europe. Gravity type, one flank curved in plan. On the river Romanche, near Grenoble. Catchment. 254 sq. kilom.; mean annual flow intercepted, 280 million cu. m. Capacity of reservoir (useful), 50 million cu. m.; height of dam above river-bed, upstream, 89·74 m. (294·4 ft.). downstream, 91·26 m. (299·4 ft.). Maximum height above foundations, 136·53 m. (447·9 ft.). Crest width, 5 m. (16·4 ft.). Stresses, at the downstream toe, reservoir full, 15 kg. per sq. cm. (213 lb. per sq. in.; 13·7 tons per sq. ft.). At the upstream toe, reservoir empty, 17 kg. per sq. cm. (242 lb. per sq. in.; 15·54 tons per sq. ft.). Full description, *The Engineer*, April 24, and May 1 and 29, 1936.

Grand Coulee Dam.—Under construction; the principal work in the Columbia Basin project in the State of Washington. Crest length, 4,200 ft.; height above lowest contact with the supporting rocks, 557 ft.; water surface to be raised 357 ft. above low water in the river; max. width at base, 500 ft.; crest width, 30 ft. Volume of concrete in the dam, the two associate power houses, the pumping stations and appurtenant works, 11,250,000 cub. yd., or 2½ times the volume placed in the Boulder Dam. The dam and power houses together will contain 32,500 tons of reinforcing steel. The capacity of the reservoir will be 1,050,000 acre-ft. (The scheme and works were described in *The Engineer*, February 4, 11 and 18, 1938.)

The Shasta Dam.—In the irrigation and flood control project for the Central Valley, California. Stores 4,500,000 acre-ft. Maximum height above foundations, 560 ft. Volume, concrete, 5,400.000 cub. yd. Slightly curved in plan on the wings. Crest length 3,100 ft., including, in straight section, a 375-ft. spillway. Initial water power plant, 375,000 h.p.

The Friant Dam.—In the same project. A concrete gravity dam; central spillway; 286 ft. maximum height (320 ft. above lowest foundation); crest length, 3,430 ft.; maximum base width, 268 ft.; crest width 20 ft.; volume 2,150,000 cub. yds. Supplies a canal 160 miles long, flow 3,500 cusecs, and another, 40 miles long, 1,000 cusecs, also one 50 miles long. Reservoir 520,500 acre-ft., the top 70,000 acre-ft. reserved for flood control, the remainder also functioning in flood control. Lake (reservoir) 15 miles in length, reduced to 10 miles with 90,000 acre-ft. dead storage.

Fontana Project.—Begun 1942, dam on the Little Tennessee River. Catchment, 1,571 sq. mi.; average flow, 3,970 cusecs. Design flood based on an assumed run-off of 10 in. in four days giving peak inflow 239,000 cusecs, corresponding to $6{,}000\sqrt{A}$; or a coefficient of 1,770 in the Ryves' formula. Usable capacity of reservoir 1,200,000 acre-ft., storing 42 per cent. of the annual flow, from the catchment. Fall from reservoir to river mouth (60 miles), 1,400 ft. The works will increase the outputs of existing plants and plants in hand, 1943, by 2,000 million kilowatt-hrs. in a year of average flow. Height of dam (or height above foundations) 470 ft. Straight, gravity, concrete type; volume, 2,600,000 cub. yds. Transverse joints spaced 50 ft.

The Génissiat Dam.—Under construction, 1936–39. In a gorge of the river Rhône. Length, at base, 147 ft., at a height of 312 ft., 656 ft. It will create a lake 14·3 mi. long with a storage capacity of 52 million cu. m., or nearly 418,000 acre-ft. Height, above river bed 80 m., or 262·35 ft.; above lowest foundation, 110 m., or 360·9 ft. The body of the dam will be divided up into monolithic concrete masses, 15 to 20 m. in width, separated by transverse vertical joints, having sealing spaces which will be filled by means of injection pipes when the mass concrete has sufficiently cooled. (The works and natural conditions are described, with plans and illustrations, in *The Engineer*, December 30, 1938.)

Hume Dam.—On the Murray River, Australia.—Catchment, 6,000 sq. mi. A main concrete dam, 1,042 ft. long, flanked by earthen embankments of 430 ft. and 3,913 ft. Capacity of the reservoir, 155,000 acre-ft., when first used, in 1931. Recently increased to 255,000 acre-ft. Ultimate capacity, as provided for in the plans, 1,250,000 acre-ft.

Vaalbank Dam.—The concrete dam is of the overspill gravity type, the section allowing of 8 ft. heightening. Volume 220,000 cub. yds. The length of the earthen embankment is 4,500 ft.; crest width, 20 ft.; upstream slope, 3 to 1; downstream slope, 3½ to 1, with berms; volume, about 300,000 cub. yds. See Part IV.

Tygart Dam.—On the Tygart River, West Virginia. Catchment, 1,183 sq. mi. Reservoir capacity, 290,000 acre-feet to the spillway crest, equivalent to 4·5 in. of run-off. Capacity above spillway crest, 88,000 acre-feet. Provided to control floods and augment low-water flows in the Monongahela River, 128 mi. long. The dam is 1,920 ft. long and 230 ft. high, above bedrock. Volume, 1,250,000 cu. yd. (See 'Operation Experiences, Tygart Reservoir,' by R. M. Morris and T. L. Reilly, Paper, *Proc. Am. Soc. C.E.*, April 1941.)

Single-Arch Dams

Mount Bold Dam.—On the Onkaparinga River, South Australia, 25 mi. from Adelaide, for which city it will create a storage of 6,500 million gallons. Catchment, 150 sq. mi. Data (30 years records), max. year's rainfall, 50·41 in. with run-off 24·17 in.; min., 16·8 in. with run-off 0·537 in. Greatest flood, 10,000 cusecs, in 1917. (Greatest recorded, 24,000 cusecs.) The dam has a constant upstream radius of 350 ft.; height over foundations 170 ft., thickness tapering from 92 ft. at the base to 12 ft. 6 in. at the crest; length along crest, 732 ft. The spillway section, 200 ft., providing for a discharge of 36,000 cusecs. The downstream face is battered 1 in 2·2; the upstream face is vertical for 120 ft. and is thence overhung with a batter of 1 in 2·3. (Described, with illustrations, in *Civil Engineering (London)*, July 1938.)

Airel Dam, Washington.—Of the thin arch type with one gravity abutment. Height, 313 ft.; crest length, 1,300 ft. The catchment, Lewis River basin, is 750 sq. miles; the maximum recorded flood 63,000 cusecs, the minimum flow 800 cusecs.

Marèges Dam.—Stores water for a power station of the Paris-Orleans Railway Company at Marèges, on the Upper Dordogne. Catchment, 2,500 sq. km. (965 sq. mi.); average flow of the river 65 mecusecs (2,295 cusecs). The dam is arched and of arched type section, extrados radius, at crest, 100 m. (328 ft.); at base, 80 m. (262·5 ft.); maximum height, 90 m. (295 ft.); capacity of the reservoir, 47 million cub. m. (1,660 million cub. ft.); available capacity, 35 million cub. m. (1,236 million cub. ft.). The hydro-electric power plant has a capacity of 150,000 kv.-a.

Parker Dam.—Colorado River, Southern California. Height above lowest foundation, 320 ft.; above river bed, 85 ft., the depth below river bed being 235 ft. Cracking of the concrete is assigned to the use of a high-alkali cement with certain types of aggregate. (See *Proc. Am. Soc. C.E.*, December 1940, and April 1941.)

O'Shaughnessy Dam.—Water supply of San Francisco. Has been increased in height to 430 ft. Curved to a radius of 700 ft., to the upstream face. Crest length, 600 ft. Might be classed as a gravity dam; for the radius is a large one, and the average thickness is about 50 ft. Heightening described, *The Engineer*, July 28, 1939.

SINGLE ARCH AND TANGENTIAL GRAVITY WINGS.

Galloway Water Power System.—Dams, as named; height (h) in feet of spillway from original ground level; radius (r) of curvature; downstream batter (b) of arch, of wings, 0·75 in all cases.

Deugh: h, 70; r, 230; b, 0·33. Ken: h, 70; r, 165; b, 0·25. Blackwater: h, 40; r, 125; b. 0·25. Carsfad: h. 65; r, 190; b, 0·30. Earlstoun: h, 70; r, 145; b, 0·22. Tongland: h, 70; r, 145; b, 0·25. (See *The Engineer*, September 25, 1936.)

Multiple-arch Dams.

Beni-Bahdel.—On the Oued Tafna, Oran. A multiple-arch dam, in the central portion, 726 ft. long, the flanking portions being of the gravity type. Total length, 1,056 ft. The counterforts are built of plain concrete, are triangular in form, and spaced 66 ft. apart between centres. They are 10 ft. thick at the top and about 16 ft. at the base and their maximum height above foundations is 188 ft. (Ref. *Bull. Tech. Suisse Rom.* 63, 141. See also *Civil Engineering* (*London*), September 1937.)

Hohenwarte Dam.—Under construction, 1940. With the completed Bleiloch Dam will control the flow of the River Elbe. Height, 243 ft.; length, 1,350 ft.; radius 1,300 ft. Will impound 6,050 million cu. ft.

Bartlett Dam.—On the Verde River, Arizona. Catchment, 5,600 sq. mi. Reservoir capacity, 200,000 acre-ft., for irrigation in the valley of the Salt River. Ten arches of 48 ft. clear span and 60 ft. between centres of hollow buttresses. Inclination of the arches about 45 deg. Described with illustrations and drawings in *The Engineer*, August 2, 1940.

Basic Economy of Dams.

VOLUME OF WATER STORED PER UNIT VOLUME OF MASONRY IN THE DAM.

Poona, India, 62; Beetaloo, Australia, 79; Booton, New Jersey, 200; Cross River, New York, 610; Granite Springs, Wyoming, 716; Wachusett, Massachusetts, 1,369; Sodom, New York, 1,569; Sweetwater, California, 1,679; Aswan, Egypt, 3,900; Periar, Madras, 2,620; Roosevelt, Arizona, 7,426; Calitzdorp, S. Africa, 113; Woronorah, 2,402; Boulder 14,470; Cauvery Metur, 1,731; Shasta, California, 1,344.

Earthen Dams: Bon Accord, Transvaal, 36·4; Blijde River, 20·2; Grassridge, S. Africa, 202; Vaalbank (300,000 cu. yd., earthen, 220,000 cu. yd. concrete), 5,866; Fort Peck, 319.

DAM FAILURES.

St. Francis Dam.—Los Angeles water supply. An arch dam, curved to a radius of 492 ft. and 205 ft. in maximum height. Failure was due to defective foundation materials, for which the design was not suited. The dam was built without State control and without the services of a consulting engineer or a geologist. The dam collapsed when the reservoir was nearly full, in March 1928. Deaths, about 400. Damage, about £2,000,000.

Spencer Dam.—The failure of the Spencer Dam in Nebraska was caused by the dam being built on a shale so weak that it was hardly more than mud rock. The dam, which failed on September 24, 1936, was built in 1926 and was a 1,800 ft. long earth dyke, about 18 ft. high, with a 400 ft. concrete spillway section. It was the spillway section which failed. (*Engineering News-Record.*)

Marshall Dam, Kansas.—An earthen dam, 1,480 ft. long and 90 ft. high. When within 10 ft. of full height a slump occurred, 170 ft. long and extending from 20 ft. upstream of the centre line to 110 ft. downstream of it. Samples taken from the disturbed ground showed high proportions of silt and soft clay.

Fort Peck Dam.—The dam was built by the hydraulic-fill method. In September 1938, when it was nearly completed, a slide carried away about 8,000 cu. yd. of the upstream slope, releasing the water from the core pool and burying eight workmen. The slide occurred suddenly, though distortions were noted, it has been stated, more than three hours before it took place.

Test bores, drilled after the accident, showed that there was very high hydrostatic pressure in the shale downstream, one hole developing a pressure equivalent to 60 ft. head above the top of the dam. It was deduced that trapped water was squeezed out of the shale. Though it was concluded that there was a good factor of safety against the bursting of the shell by the core, the upstream face, the average slope of which had been 1 in 4, was very much flattened when the dam was completed.

River Control and Diversion Weirs.

Egypt.—The Nag Hammadi Barrage (fully described in articles in *The Engineer*, September 2, 1927, January 30 and February 6, 1931). Contract price £1,976,555. Founded entirely on sand. Length between centres of abutment piers, 822·25 m., or nearly 2,700 ft. Total of sluiceways, 1,968 ft. provided by 100 sluices of 6 m. widths and 6·1, 7·1, or 8·1 m. deep. Opened December 1930.

Gebel Aulia, River Nile.—Including the portions in embankment. The length of the masonry dam is 5,554 ft.; of embankment, 10,850 ft. (1,693 m. and 3,307 m.). The masonry portion includes a solid dam, 60 ft. high, a length of 1,488 ft. with 60 sluices, and a large lock. The water will pool up to 300 ml. upstream on a width of about 4 ml. The first year's programme of work was completed before the rise of the Nile in 1934 and the river diverted to a channel, 525 ft. wide, between embankments. The 1935 programme included the closing of this channel and the diversion of the river to the eastern bank, where a temporary bridge with an opening span had been provided. Estimated cost about £E2,100,000. The work was completed in May 1937. See 'Literature.'

Mohammad Aly Barrages.—New barrages and locks in the Damietta and Rosetta branches of the Nile. Estimates: weirs, £1,168,000; sluice gates and mechanisms, £230,000; lock gates and swing bridges, £102,000. See 'Literature.'

Iraq.—The Kut Barrage. Opened March 1939. On the River Tigris. Has the effect of semi-canalising a tributary, the Shatt-el-Gharraf, for a considerable distance. Rise of the Tigris, from low water to high flood (140,000 cusecs) level, is 20 ft. There are 56 sluices, each 19 ft. 8 in. wide and controlled by a single gate. The piers are of precast concrete blocks and mass concrete hearting. (*The Engineer*, April 1939.)

Literature.

'Arch Deflections and Temperature Stresses in Curved Dams.' Article, by Ivan E. Houk, *The Engineer*, April 2 and 9, 1937.

'A Buttressed Dam Built in 1747.' *Revista de Obras Publicas*, June 1, 1936 (*Engineering Abstracts*, vol. 70, No. 66).

'Stresses Around Galleries in Concrete Dams.' Article, by Ivan E. Houk, *The Engineer*.

'Stress Conditions in a High Multiple-arch Dam.' Article, by Ivan E. Houk, *The Engineer*, July 7, 1939.

'Geology of Dam Sites.' By Prof. W. J. Mead. (1) 'In Hard Rock'; (2) 'In Shale and Earth.' Articles, *Civil Engineering* (U.S.A.), (1) May, (2) June 1937. (Extracts, *The Surveyor*, (1) June 11, (2) August 20, 1937.)

'Dams on Shale.' By Irving B. Crosby. Paper, *Proc. Am. Soc. C.E.*, Vol. 66, No. 5; May 1940. (Extracts and Abstract, *The Surveyor*, June 14, 1940.)

'British Practice in the Founding of Dams.' By W. L. Lowe-Browne. Paper, Inst. of Water Engineers.

'Fort Peck Dam: Investigation of the Cause of the Slide.' By T. A. Middlebrooks. Paper, *Proc. Am. Soc. C.E.*, December 1940.

'The Gebel Aulia Dam.' By A. G. Vaughan-Lee. Paper No. 5238. *Journal Inst. C.E.*, June 1941.

'The Gorge Dam.' By W. J. E. Binnie and H. J. F. Gourley. Paper, *Journal Inst. C.E.*, March 1939.

'The Design of Dams.' By F. W. Hanna and R. C. Kennedy. Book: reviewed, *Civil Engineering (London)*, March 1939.

'Assiut Barrage Remodelling.' *The Engineer*, September 2, 1938.

'The Emerson Barrage.' By F. F. Haigh. Paper, *Journal Inst. C.E.*, December, 1941.

'Testing the Materials of an Earthen Dam.' By F. M. Van Auken. Paper reproduced in *Civil Engineering* (U.S.A.), August 1941. Relates to systematic and comprehensive tests.

Paper No. 5324, *Journal Inst. C.E.*, November 1942. By L. F. Cooling and H. Q. Golder. Relates to an analysis of the failure of a dam during construction.

'The Waldershelf Slip: Broomhead Reservoir.' Paper, by L. Bendelow, describing the extensive and elaborate remedial works carried out. *Journal Inst. C.E.*, February 1944.

ADDENDA 1949.

DAM FAILURES.

The design of earthen dams and the causes of some failures was the subject of a paper (Inst. E. Australia, 1944), by Mr. M. G. Speedie, acting chief designing engineer, State Rivers and Water Supply Commission of Australia. The following notes relate to passages in that paper.

Source of Information.—In a table analysing the causes of failure of 253 dams (H. C. Hinderlider, *Trans. Am. Soc. C.E.*, Vol. 98, 1933), the earthen dams number 159. Of those earthen dams which showed some form of failure, 29 per cent. were less than 25 ft. high, 82 per cent. less than 75 ft. Lack of spillway capacity caused 28 per cent. of the failures; inadequate cut-off provision, 22 per cent.; excessive proportion of fine material, 8 per cent.; instability of embankment or foundation, 5 per cent.

Piping Through Foundations.—This was the cause of the failure of the Corpus Christi Dam, on the Nueces River, Texas. The northern wing wall was undermined and overturned, and some 200 ft. of embankment, 60,000 cu. yds. of filling, washed away. The sheet piling had not been driven into the clay of the foundation—it was officially concluded; but one engineer thought that because sheet piling is not perfectly watertight, resulting leakage may cause piping. Differential settlement was another suggested cause. The dam is 4,080 ft. long, including 1,250 ft. of concrete spillway. Maximum height, 61 ft.

Horizontal Shear.—Tappan Dam, when under construction, was 28 ft. high when some hundreds of feet of the upstream toe moved about 8 ft. into the reservoir. A horizontal shear plane was found 6 to 12 ft. below the top of the foundation clay layer, extending for more than 50 ft., under the filling, from the upstream toe.

Inadequate Drainage.—To this defect was assigned a failure of the Alexander Dam, Kauai, Hawaiian Islands. Some 250,000 cu. yds. of material slid downstream in 30 seconds.

High Pore Pressures.—Belle Fourch Dam is 6,500 ft. long and 150 ft. high. The upstream slope was originally 1⅓ to 1 above and 2 to 1 below maximum water level. In 1932, about 25 years after construction, a slide of 20,000 cu. yds. of earth, 600 ft. in length, occurred on the upstream face following a water-level drawdown of 25 ft. in 60 days. The rate of drawdown was, obviously, too rapid to allow of release of pore water at a corresponding rate.

RECENT LITERATURE.

'The Waldershelf Slip: Broomhead Reservoir.' By L. Bendelow. Paper (*Journal Inst. C.E.*, April 1941).

Foundations: Design.—A paper by T. B. Crosby (*Proc. Am. Soc. C.E.*, May 1940), related to Dams founded on Shale. In the same issue are papers on the subjects: basic design assumptions, the design of arch dams, the preparation of foundations, construction joints, and concrete control.

Geology.—In his book, 'Geology for Engineers,' Mr. F. G. H. Blyth, deals very ably with the subjects: rainfall, run-off, dispersal of rainwater, ground water, and some of the most important formations from which water is derived. The chapter on the geology of reservoir and dam sites includes descriptions of conditions at the Vyrnwy Reservoir; Caban Goch Dam, Rhader; the Dolgarrag Dam, Laggan; Boulder and Grand Coulée Dams. [London: Edward Arnold & Co., second edition, 1945. Price 21*s*. net.]

SECTION XVIII

PART IV

IRRIGATION — AVAILABLE SUPPLY — EVAPORATION — ABSORPTION—EVAPORATION FROM SOILS—LOSSES FROM IRRIGATION CANALS — CRITICAL VELOCITY — DUTY OF WATER.

Contributed by Reginald Ryves, M.Cons.E.
(Author of 'A Note on Mass Dams and Dams of Arches,' etc.)

IRRIGATION.

Available Supply of Water.

Irrigation from Large Rivers.—The discharges of the river during the seasons when the water will be required are calculated from the observed depths over weirs, from differences of water level above and below bridge openings, and by the methods described in Part II under 'Capacities of Canals.' Whether these supplies will be fully available or not may depend upon the water levels which can be maintained above the weir, and upon what cultivable areas are commanded by the water at these levels. In this respect the scheme must be considered as a whole, and by making suitable regulations as to the crops which may be grown on certain areas and at certain seasons the availability of the supply may be increased.

Supplies from Defined Catchments.—In the case of a supply from a defined catchment the estimates of run-off may be made by the methods described under 'Rainfall,' the necessary deductions being made when the whole of the water stored cannot be directly utilised. Out of the whole recorded range of annual or seasonal rainfalls, and corresponding runs-off, one will be chosen as that suitable as a basis for the project, and the capacity of the reservoir designed accordingly. A table drawn up for catchment areas in Nagpur, Central Provinces, India (*Strange*), gives estimated yields for monsoon rainfalls ranging from 1 in. to 60 ins., and from this table the following examples are taken.

MONSOON RAINFALLS AND ESTIMATED RUNS-OFF FOR NAGPUR.

Monsoon Rainfall in Inches.		1	5	10	20	30	40	50	60
Runs off in millions of cubic feet per square mile	Good catchment	0·002	0·116	0·999	6·970	18·33	34·85	56·69	83·63
	Aver. catchment	0·001	0·087	0·749	5·227	13·75	26·13	42·51	62·73
	Bad catchment .	0·001	0·058	0·499	3·485	9·17	17·42	28·34	41·87

Evaporation from a Water Surface.

The actual rate of evaporation from the surface of a pool of water depends mainly on four factors: the temperature of the water itself; the velocity of the wind; the degree of humidity of the air; and the temperature of the air. As regards the first, this is more or less proportional to the air temperatures if we are measuring evaporation over considerable periods of time in different places. As regards the second, the velocity of the wind, this factor does not as a rule vary so much as others if we take a period such as a year; but it sometimes varies a great deal within areas of moderate extent, and it should be taken into consideration in making estimates of evaporation. The degree of humidity varies very greatly, not only in different countries or different districts, but also in proportion to the distance from the sea and, in some cases, with elevation. There may be a marked decrease in the humidity of the air in a distance of a few miles from a long, straight coast line, the decrease being more gradual as we pass farther inland. There are also sharp differences between places just within and those just without the path of a monsoon current. The relation between water temperature and the rate of evaporation was studied at four stations in California (*Fortier*) with the following results, the evaporation being in sixty-fourths of an inch per day. The tanks used were of galvanised iron, 22 ins. in diameter and 28 ins. deep sunk in the ground.

Water Temperature and Evaporation.—53°F., 6 ; 62°F.. 12 ; 73°F., 23 ; 80°F., 31 ; 88°F., 39.

The following were the results of experiments to test the rates of evaporation in air currents of different velocities, that in still air being taken as unity. (*Russell.*)

Wind Velocity and Evaporation.—Velocity 0, evaporation 1·00; velocity 5 miles an hour, evaporation 2·2; velocity 15, evaporation 4·9; velocity 20, evaporation 5·7; velocity 25, evaporation 6·1; velocity 30, evaporation 6·3. Observations taken at the Aswan reservoir show the relative importance of average month's temperatures and average month's wind velocities. The results do not, of course, aid in the establishment of a relation between actual wind velocities, actual temperatures, and rates of evaporation, but they are of practical use for purposes of comparison.

Average Velocities of Wind, Average Temperatures, and Rates of Evaporation, Aswan, Egypt. Latitude, 24°. Year 1912.

Month	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.
Wind velocity, miles per hour	3·6	4·2	6·6	5·3	4·2	3·0	3·3	3·5	1·2	1·7	3·4	3·7
Temperature °F.	60·7°	66·7°	72·3°	80·7°	87·2°	91·4°	91·2°	92·1°	89·3°	83·3°	73·6°	62·2°
Evaporation, ins. per day	0·12	0·17	0·24	0·31	0·35	0·45	0·43	0·44	0·32	0·29	0·21	0·15

Humidity and Evaporation.

By taking a sufficient number of observations of rates of evaporation, it is possible to establish at the same time the relation between evaporation and air temperature and that between evaporation and humidity. Experiments of this kind with water at a constant temperature seem to be lacking, nor have observers, as a rule, noted the actual temperatures of the water. Results are, however, comparable in groups, such as those with small experimental tanks, and those with floating tanks in deep and shallow waters, respectively.

The following table gives the results of observations at Vehar reservoir, near Bombay, latitude 19° (*Conybeare*), and in Nagpur, 21° (*Binnie*). The values seem to be low, but this hardly affects their application in practice, as the engineer would draw the curves and fix his own zero line on the result of monthly or annual evaporations for the locality.

Approximate Monthly Evaporations in Feet with given Temperatures and Relative Humidities.

Percentage of Saturation.	Mean Temperatures, Fahr. 65°	70°	75°	80°	85°	90°	93°
88	0·14	0·17	0·19	0·21	0·24	0·27	0·30
85	0·19	0·22	0·22	0·23	0·27	0·30	0·33
80	0·22	0·27	0·29	0·30	0·32	0·36	0·38
75	0·24	0·29	0·33	0·35	0·37	0·40	0·42
70	0·26	0·30	0·34	0·35	0·38	0·42	0·44
65	0·28	0·31	0·34	0·36	0·40	0·44	0·46
60	0·30	0·35	0·38	0·40	0·42	0·47	0·49
55	0·32	0·37	0·42	0·46	0·50	0·52	0·54
50	0·35	0·40	0·45	0·50	0·55	0·60	0·62
45	0·45	0·50	0·57	0·67	0·71	0·75	0·78
40	0·61	0·66	0·70	0·72	0·77	0·82	0·84

The figures in the following table are taken from results of observations by the Egyptian Public Works Survey Department (*Cay*). It is clear that the rate of evaporation depends considerably upon the mean altitude of the sun, since, for air temperatures which are nearly the same, the rate of evaporation is affected by the latitude and the month. This is shown by the small tables, and the evaporation at latitude 24° in October with air temperature 83° and humidity 41 may be compared with that at latitude 27° in August with nearly the same humidity.

OBSERVATIONS BY EGYPTIAN PUBLIC WORKS DEPARTMENT.

Locality	Abassia, Cairo. Lat., 30° 1′.					
Month	May.	June.	July.	Aug.	Sept.	Oct.
Mean temperature of month, Fahr.	75°	78°	81°	80°	77°	73°
Relative humidity	47	50	55	62	70	71
Evaporation, inches	·32	·25	·20	·22	·17	·18
Locality	Asyut. Lat., 27° 11′.					
Month	May.	June.	July.	Aug.	Sept.	Oct.
Mean temperature of month, Fahr.	81°	84°	—	85°	81°	76°
Relative humidity	26	26	—	40	46	56
Evaporation, inches	·55	·61	—	·51	·37	·27
Locality	Assuan. Lat., 24° 2′.					
Month	May.	June.	July.	Aug.	Sept.	Oct.
Mean temperature of month, Fahr.	90°	90°	90°	90°	89°	83°
Relative humidity	16	20	20	20	32	41
Evaporation, inches	·62	·60	·52	·52	·47	·35

Temperature 80°–81°F.				Temperature 89°–90°F.			
Humidity.	Evap.	Month.	Lat.	Humidity.	Evap.	Month.	Lat.
62	·22	Aug.	30°	32	·47	Sept.	24°
55	·20	July	30°	20	·60	June	24°
46	·37	Sept.	27°	20	·52	July & Aug.	24°
26	·55	May	27°	16	·62	May	24°

The effect of altitude upon evaporation was tested at four stations on the same mountain in California with the following results :—

Evaporation at Different Altitudes.—4,500 ft., 2·69 ins. ; 7,100 ft., 2·04 ins. ; 10,000 ft., 1·63 in. ; 12,000 ft., 1·61 in.

Annual Evaporation.

At stations selected as representing the chief irrigation districts of California the following were evaporations for a year (*Fortier*). At Berkeley, the cooler coast area with relatively high rainfall, 41 ins. ; at Chico, the bottom of the Sacramento Valley, 53 ins. ; at Pomona, representing Southern California, 66 ins. ; at Tulare, the San Joachin Valley, 68 ins. ; at Calexico, the Imperial Valley, 71 ins.

The evaporation at a number of representative stations is given in the following table, but the methods adopted in measuring evaporation varied considerably. At Croydon, the evaporation from a 5-in. vessel in air was more than 50 per cent. greater than that from the 12-in. vessel floating in water.

Evaporation at Representative Stations.

Locality.	Approximate Latitude.	Evaporation in Inches.	Remarks.
Whitehaven, England	54½°	29·91	Elevation 90 ft. near coast.
Bolton-le-Moors, England	53½°	25·65	Average of 10 years. Elevation 320 ft.
Croydon, England	51½°	19·95	13 years, max. 22·85, min. 15·01. A 12-in. vessel floating in a tank 4 ft. in diameter.
Kennich, Devonshire, England	50½°	20·88	Elevation 836 ft. Average of 4 years.
Dijon, France	47½°	26·11	Inland station.
Boston, U.S.A.	42½°	39·11	
Croton River, U.S.A.	41½°	24·15	
New York, U.S.A.	41°	39·21	
Tharsis Mines, Spain	37½°	60	Average of 7 years, max. nearly 65 ins.
Chicco, California	39½°	53	Galvanised iron tanks, 2 to 3 ft. in diameter, and 30 to 36 ins. deep; sunk in the earth.
Berkeley, „	38°	41	
Tulare, „	36°	68	
Pomona, „	34°	66	
Calexico, „	—	71	
A station in California	—	30	Elevation 6,225 ft.
Perth, Western Australia	32°	65·8	
Rajputana, India	25° to 28°	73·8	General average.
The Rand, South Africa	26°	63	
Tulsi Lake, Bombay Presidency	19°	36·4	Floating boxes. Average of 8 years.
Red Hills, Madras, India	13°	66·9	Near the coast.

Seasonal and Monthly Losses by Evaporation.

It is often necessary to be able to estimate the losses by evaporation during particular seasons, and monthly rates of evaporation, or daily rates in individual months, are frequently recorded. For considerable areas in India the seasonal losses may roughly be reckoned as: In the cold weather 12 ins., in the hot weather 32 ins., during the monsoon 16 ins.; total, 60 ins.

In the following table, some of the rates of evaporation are comparable, but the table has been drawn up chiefly in order to show the relative rates of evaporation in different months at the same station.

Monthly Evaporations from Water Surfaces. (Inches.)

Locality.	Approx. Lat.	Month. Jan.	Feb.	Mar.	April.	May.	June.
Emdrup, Denmark (av. of 11 years)	56°	0·7	0·5	0·9	2·0	3·9	5·3
Bolton-le-Moors, England, 320 ft. (av. of 10 years)	53½°	0·64	0·95	1·59	2·59	4·38	3·84
Whitehaven, England, 90 ft.	54½°	0·95	1·01	1·77	2·71	4·11	4·25
Tulare, California	36°	1·5	2·7	3·3	3·8	7·2	10·5
Perth, Australia	32°	2·06 (July)	1·75 (Aug.)	3·20 (Sept.)	5·51 (Oct.)	7·69 (Nov.)	9·47 (Dec.)
Cairo, Egypt	30°	—	—	—	—	9·88	7·56
Aswan, Egypt	24°	—	—	—	—	19·16	17·96
Pashan, Poona, India	18°	5·27	3·92	5·27	8·10	11·78	*
Red Hills, Madras, India	13°	4·37	4·93	6·57	7·51	8·67	8·16

* Monsoon month.

MONTHLY EVAPORATIONS FROM WATER SURFACES. (Inches.)—*Cont.*

Locality.	Approx. Lat.	Month.						Year.
		July.	Aug.	Sept.	Oct.	Nov.	Dec.	
Emdrup, Denmark (av. of 11 years) . . .	56°	5·2	4·4	2·6	1·3	0·7	0·5	27·9
Bolton-le-Moors, England, 320 ft. (av. of 10 years)	53½°	4·02	3·06	2·02	1·28	0·81	0·47	25·65
Whitehaven, England, 90 ft.	54½°	4·13	3·29	2·96	1·76	1·25	1·02	29·91
Tulare, California . .	36°	11·6	9·8	7·4	5·5	3·8	2·0	68·5
Perth, Australia . .	32°	10·22 (Jan.)	8·35 (Feb.)	8·26 (Mar.)	5·21 (April)	2·27 (May)	1·80 (June)	65·79
Cairo, Egypt . . .	30°	6·35	6·95	5·09	5·49	—	—	—
Assuan, Egypt . . .	24°	16·11	16·11	14·17	10·74	—	—	—
Pashan, Poona, India .	18°	Monsoon	months.		7·75	5·7	4·34	—
Red Hills, Madras, India.	13°	5·91	3·92	4·38	4·80	3·29	4·01	66·88

Losses by Evaporation and Absorption taken together.

In a great many cases the available data of losses from reservoirs are those for evaporation and absorption, or percolation, taken together. The more nearly the engineer can estimate what proportion of the total loss is due to evaporation alone, the more easily will he be able to decide as to the necessity or expediency of taking measures to reduce percolation. Fairly high losses may be tolerated from reservoirs where a considerable proportion of the percolating water reaches wells from which it can be lifted and used; and this may in some cases be an advantage when the level of the water in the reservoir is very low, or, at least, a mitigation of the disadvantage. Actual losses by percolation can be estimated when the rate of evaporation is obviously very small. Conversely, the proportion lost by evaporation may be estimated when the losses by percolation are known to be small. The losses may be measured as depths on the maximum waterspread, or as actual depths on the mean of the waterspreads at the beginning and end of each short period of observation. Thus, there are often apparent discrepancies between really similar cases, especially when the reservoirs are very shallow in parts, and when, as often happens, the maximum rate of evaporation occurs when the tank is very low.

Losses by percolation are sometimes very large. The losses from four irrigation reservoirs in the United States were as follows, measured in acre-inches or acre-feet referred to the *maximum* waterspread in acres :—(1) one year, 50·4 ins., all attributed to evaporation ; (2) maximum of 2 years, 53·8 ins.; (3) maximum of 4 years, 132 ins.; (4) maximum of 2 years, 17·3 ft., ⅞ of which was attributed to percolation.

At Red Hills tank, Madras, a total loss of about 7·5 ft. seems to have been indicated by one set of tests, and 8·39 by another series. An allowance of 9 ft. is now made, and is considered to be ample, a fairly liberal estimate being desirable as the water is used for the supply of the city. Tanks in Rajputana were found to lose 9·77 ft. per annum, of which 3·66 ft. was attributed to absorption.

The reservoirs of the Sheffield waterworks are estimated to have lost only about 15 to 15½ inches by evaporation and absorption, the loss at Rotherham in a neighbouring area being 22 ins.

The following were the results of observations at Red Hills Tank, Madras, the year being roughly divided into seasons which do not, however, begin and end at even approximately the same dates every year.

SEASONAL LOSSES (INCHES PER DAY) BY EVAPORATION AND ABSORPTION, RED HILLS TANK MADRAS.

Humid Season under the modified influence of the South-West Monsoon.		North-East Monsoon and Cool Dry Season.		Hot Dry Season.	
July	0·33	Nov.	0·27	Mar.	0·26
Aug.	0·32	Dec.	0·13	April	0·30
Sept.	0·38	Jan.	0·24	May	0·37
Oct.	0·27	Feb.	0·24	June	0·36
Average	0·33	Average	0·22	Average	0·31

EVAPORATION AND ABSORPTION (ALMOST ENTIRELY EVAPORATION) RECKONED ON THE TOP AREAS OF RESERVOIRS NEAR DHARWAR, BOMBAY PRESIDENCY (*Strange*).

Month.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May.	June.	July.	Aug.	Sept.	Oct.	Total.
Evaporation, ins. per day	0·16	0·16	0·16	0·33	0·67	0·67	0·51	0·31	0·16	0·16	0·16	0·17	3·65 ft.

Evaporation from the Soil.

A study of rates of evaporation from the soil is of practical importance as bearing upon estimates of the actual amounts of water required by crops under varying conditions, and the probable losses from the ground when the water is applied some time before cultivation. The importance of the subject is the greater in proportion to the extent to which the cultivation has the character of fruit growing or gardening, as distinct from the growth of large areas of grain crops. In orchards especially, it is possible so to apply the water that the loss by evaporation is considerably reduced. It is often an advantage to keep the water in distinct channels as long as possible.

CALIFORNIAN EXPERIMENTS ON EVAPORATION FROM THE SOIL.

Important series of tests have been carried out during recent years in California, and have been reported by Mr. S. Fortier, Chief of Irrigation Investigations, United States Department of Agriculture.

The experiments were carried out with tanks about 47 ins. deep and 23½ ins. diameter, sunk into the soil and filled nearly to the brim. These tanks were weighed in order to measure losses. The following tabular statement gives the results of experiments made in July and August on the evaporation from soils containing different proportions of water, measured as inches depth over the area of the surface. In the case of the initial free moisture this represented about 4½ to 6 per cent. The amounts of irrigation water added represented, respectively, very light, light, and ordinary irrigation.

WATER CONTENTS IN SOIL AND EVAPORATION (IN INCHES).

Initial moisture	2·8	2·7	2·7	4·1
Irrigation water	0	2·6	4·8	6·3
10 days' evaporation . . .	0·4	0·9	1·2	1·5

Evaporation from the soil may be much reduced by beating or crumbling the dry surface to a powder, by spreading a layer of finely divided soil over the surface, or by irrigating by means of furrows, afterwards spreading the dry soil from between the furrows by some means which will tend to crumble or powder it. In small-scale irrigation and where the cost of the water is high these considerations may be of importance. The following results were obtained in California with experimental tanks containing sandy loam, the evaporation from which when it was left uncovered after the water had been applied was compared with the evaporation after coverings of different depths of finely divided sandy loam had been added. The result of the experiment with the thinnest layer is, of course, the most important generally. During the fourteen days of the experiment the shade temperature of the air at noon was often more than 100°, while the soil temperature in the sun, at noon, varied from 120° to 140°. The figures were :—

Evaporation in hundredths of an inch per day: from uncovered soil, 72·5; from soil protected by a 4-inch layer, 20·6; 8-inch layer, 10·0; 10-inch layer, 2·0.

The following were the results of a twenty-one days' test.

SOIL EVAPORATION IN PERCENTAGE OF WATER APPLIED.

Days	2½	6½	10	14	17	21
Bare soil	11	18	21	25	28	32
3-inch cover . .	2	5	7	9½	11½	14

A further series of experiments was carried out with tanks 47 inches deep and 23½ inches diameter.*

* See *Engineering News*, Sept. 5, 1912, but note that in figs. 2 and 3 the diagrams of evaporation in inches have been accidentally transposed.

In one set of tests the tanks were nearly filled with soil, leaving room for the covering of loose soil in each case, spread after the application of the water, while in another set of tests the tanks were nearly filled and the water applied at the bottoms of furrows of corresponding depths, the soil being afterwards spread. The following table gives the results of both experiments :

EVAPORATION AND SOIL COVERINGS.

Period after water was applied: days . .		7	14	21	28	21	28
		Evaporation in percentage of water applied.				Evaporation in inches.	
Surface irrigated and left		17·5	25	31·5 (29)		1·75	
Surfaces irrigated and covered with loose soil (inches)	3 ins.	4·8	9·5	13·7 (12·5)		0·75	
	6 ins.	1·5	3·2	6·0 (5·7)		0·34	
	9 ins.	1·5	1·3	4·1 (3·3)		0·22	
Surface irrigated and left		13·2	16·4	18·6	20·8		1·25
Water applied at the bottoms of furrows, depths in inches	3 ins.	9·2	12·0	14·9	16·5		0·99
	6 ins.	7·0	9·6	12·3	14·3		0·86
	9 ins.	6·1	7·5	9·9	12·0		0·72

In the column for the 21st day, the figures in brackets are calculated from the evaporation in inches, and possibly correspond to earlier readings. The other figures in the column are taken from the curve.

Relation between Evaporation from Soil and Evaporation from Water.

The experiments, on the results of which the table below is based, showed what were the evaporations from soils containing different proportions of water, and from a water surface.

Percentage of Free Water.	Temperatures, Fahrenheit.				Weekly Evaporation in Inches.	
	Air in Shade.	Soil in Shade.	Soil in Sun.	Surface of Water.	Soil.	Water.
Saturated.	71°	76°	95°	77°	4·75	1·88
17·5%	76°	78°	106°	80°	1·33	1·94
11·9%	76°	78°	106°	80°	1·13	1·94
8·9%	76°	78°	108°	80°	0·88	1·94
4·8%	76°	78°	108°	80°	0·25	1·94

Ratio of Pan Evaporation to Open Water Evaporation.

(*Meeker: Trans. Am.Soc.C.E., vol.* 90, 1927.)

Denver, Colorado, field laboratory ; on open prairie land at elevation 5,346 ft. Mean annual temperature, at Denver, 53 years record, 50·1° F. ; mean annual precipitation 14·27 ins. ; relative humidity 53 per cent.

Ground Pans.—Pans 3 ft. deep, water in pans 2·75 ft. deep ; pans set 2·75 in the ground. Coefficients of reduction to open water surfaces : diameter of circular pans ; 2 ft., coeff. 0·77 ; 4 ft., coeff. 0·84 ; 6 ft., coeff. 0·90 ; 9 ft., coeff. 0·98. Also 6 ft., tank, 2 ft. deep, 1·75 ft. in ground, coeff. 0·88 ; and 4 ft. dia. 10 ins. deep pan, set on timbers, depth of water 0·62 ft., coeff. 0·66.

Floating Pans.—Three feet square by 1·5 ft. deep, depth of water in pan 1·25 ft. ; coeff. 0·91; circular, 4 ft. dia., 10 ins. deep, depth of water 0·58 ft., coeff. 0·92 ; 3·5 dia., 1·83 ft. deep, depth of water 1·5 ft., 0·91 (approximately, by interpolation).

Conversion.—(R. Follansbee, *Proc. Am. Soc. C.E.*, vol. 59, p. 223.) Conversions of pan evaporations to equivalent (*i.e.*, corresponding) evaporation from a reservoir: (1) Land pan, 0·70, with a reasonable range from 0·60 to 0·82 ; (2) sunken pan, 0·78, ranging from 0·75 to 0·86 ; (3) floating pan, 0·80, ranging from 0·78 to 0·82.

Evaporation and Grass.—California Bureau of Agricultural Engineering, 1933. Tests of evaporation from bare soil and from soil covered with vegetation, including Salt Grass, Bermuda Grass and Tule. With the water table ranging from 1 ft. to 5 ft. below the surface, the grasses evaporated water at the rate of 22 to 43 in. per year.

ANNUAL EVAPORATION IN INCHES DURING TWELVE YEARS, LEA BRIDGE, ENGLAND.

Water: max., 26·93 ; min., 17·33 ; mean, 22·2.
Soil: max., 25·14 ; min., 12·07 ; mean, 19·53.
Sand: max., 9·10 ; min., 1·42 ; mean, 4·65. (*Greaves.*)

Rice Irrigation.—(T. T. Cheng and C. L. Pien.) See *The Engineer*, July 26, 1940. Typical figures are :—

	China.				United States (South-West).	
	Ins.	%	Ins.	%	Ins.	%
Evaporation .	8·54	23·7	9·28	47·5	10·10	36·0
Transpiration .	15·48	43·2	6·85	35·0	16·38	58·5
Seepage . .	11·88	33·1	3·42	17·5	1·56	5·5
Total . .	35·90	100·0	19·55	100·0	28·04	100·0

Factors of variation: variety of rice ; method of irrigation ; length of the irrigation season, and climate.

Water Losses from Irrigation Canals.

No general rules can be given for estimating the rate at which water is lost from irrigation canals. Some are practically watertight, and only lose water by evaporation, while others lose large proportions of their supply by leakage.

On the Ganges Canal, at one period the water sent down was accounted for as follows :— Irrigation, 56 per cent. ; loss in canal, 15 per cent. ; in distributaries, 7 per cent. ; in village water-courses, 22 per cent.

In the Punjab, a loss of 8 cusecs per million square feet of wetted surface has been allowed, and larger amounts in some cases. Certain canals with discharges from 400 to 450 cusecs were found to lose about one cusec per mile. The Bari Doab Canal, when eighteen years old, lost about 12 to 14 per cent. of its discharge in a length of fifty miles.

Recently, instead of allowing 8 cusecs per million square feet of wetted area, it has been usual to allow, within the limits 4 ft. to 12 ft. depths, one cusec per foot depth per million wetted square feet.

Many canals lose a great deal of water for some time after their construction, and afterwards become reasonably watertight. The deposit of silt at a reasonable rate for some time after the canal is in service may be facilitated by suitable designing of the cross-sections and temporary adjustments of levels at the falls ; but no general rules can be given as to this difficult problem.

When L is the loss from a canal, measured in the first mile, the loss up to a further distance of D miles is L Dx, where x varies from $\frac{1}{3}$ to $\frac{2}{3}$.

Reservoir Losses.—Cauvery-Metur reservoir ; estimated, evaporation and percolation, inches : June, 7·0 ; July, 5·4 ; August, 5·8 ; September, 5·8 ; October, 5·0 ; November, 4·0 ; December, 4·0 ; February, 4·0 ; March, 6·0 ; April, 9·0 ; May, 10·0 ; Total, 70 ins.

River Losses.—Percentage losses, estimated, during the flow from the Cauvery-Metur reservoir to the Grand Anicut—for the double crop : June, 50 ; July–October 15, 20 ; October 16 to December (incl.), 15 ; December 16 to January 15, 40. Single crop : July to November 15, 20 ; November 15 to December 15 ,30.

Critical Velocity.

The 'critical velocity' is that mean velocity at which, for a canal of given depth, the current will neither deposit much silt nor unduly erode its bed. A determination of the critical velocity, the theory of which was evolved by R. G. Kennedy (see Min. Proc. Civil Engineers, vol. cxix.), is of great importance in the design of irrigation canals.

The equation for determining the critical velocity for a given depth of water in the canal is

$$V_o = cd^m.$$

The values of c and m vary somewhat according to the nature of the silt. For fine sand-silt brought down from the hills usual values are, $c = 0·84$ and $m = 0·64$. The following table gives the critical velocities calculated on this basis, for different depths, and the observed critical velocities on the Shwebo and Mandalay canals, Burma. With the latter velocities there was, on the whole, a slight tendency to scour.

CRITICAL VELOCITIES.

Depths in feet . . .	1	2	3	4	5	6	7	8	9	10	12	15	20
$V = 0·84\ d^m$, suitable for the Punjab . . .	0·84	1·30	1·70	2·04	2·35	2·64	2·92	3·18	3·43	3·67	4·12	4·75	5·71
Observed in Shwebo and Mandalay Canals . .	—	—	1·72	1·98	2·24	2·48	2·75	—	3·09	—	—	—	—

A canal which would be non-silting on a slope of 1 in 7,600, if drawn from the Indus in Sind, would require to have a slope of 1 in 4,200 to be non-silting if carrying the silt usually found in the Punjab (*Buckley*).

Argentina.—Adopted for canals of the Upper Rio Negro Irrigation (*R. E. Ballester*):

$$V_o = 0·52d^{0·64} \text{ metric,}$$

the relation being, British $V_o = 0·84d^{0·64}$ corresponds to metric, $V_o = 0·546d^{0·64}$.

United States.—Working data for irrigation engineers (*E. A. Moritz*): proposed values; for fine silt or sandy mud, $V_o = 0·464d^{0·5}$; for somewhat coarser silt or sandy mud, $V_o = 0·508d^{0·5}$ and, for the same index, the coefficients: sandy, loamy silt, 0·55; coarse silt and other hard earths, 0·602.

Lacey's Formulae.—First presented in Paper No. 4736, *Inst.C.E.*, by G. Lacey.

(1) $V_o = 1·17\ (fR)^{0·5}$; (2) $Af^2 = 3·8\ V_o^5$; (3) $Qf^2 = 3·8\ V_o^6$; (4) $P_w = 2·668\ Q^{0·5}$; (5) $S = f^{5/3}/1788\ Q^{1/6}$. In which, P_w is wetted perimeter and f is a silt factor, equal to the square of the ratio of (*a*) the critical velocity required for the depth of channel and character of silt considered, to (*b*) the critical velocity for the same depth and the character of silt in the Punjab canals, on which Kennedy's formula is based. This definition is limited to channels of 2,000 cusecs or less; for rivers, $f = 8\sqrt{d}$, where d is the diameter, in ins., of the predominant type of silt transported.

CRITICAL VELOCITY RATIO.

The application of a formula cannot be usefully made except by engineers of special qualifications and experience. The subject was presented by such engineers in a discussion on a paper, 'A Theory of Silt and Scour,' by W. M. Griffiths (*Min. Proc. Inst. C.E.*, vol. 223), in the course of which Sir Thomas R. J. Ward observed, in reference to the application of Kennedy's formula to Punjab canals: It was soon discovered that, although the rule worked well for the head reach of the distributary, farther down the distributary Nature threw up very broad berms which reduced the width and increased the depth of the cross section to what had been the practice before Mr. Kennedy published his diagrams and formula. That difficulty had been surmounted by the introduction of the 'critical velocity ratio'—by reducing the coefficient as one went down the channel. Knowledge of that ratio was a matter of practice, and it could only be learned by long experience in that particular class of designing.

HYDRAULIC MEAN DEPTH: MEAN VELOCITY: SCOUR.

(*P. A. M. Parker.*)

For:	Fine Silt.			Heavy Silt and Fine Sand.			Coarse Sand		
There is no scour in a channel of H.M.D.	1·0	2·5	5·0	1·0	2·5	5·0	1·0	2·5	5·0
Until a mean velocity is reached of	0·4	0·7	0·9	0·9	1·5	1·75	1·75	2·25	3·0

	Small Pebbles (Size of Peas) and Gravel.			Large Pebbles (Hen's Egg) and Coarse Gravel.				Large Stones.	
There is no scour in a channel of H.M.D.	1·0	2·5	5·0	1·0	2·5	5·0	10·0	1·0	10·0
Until a mean velocity is reached of	2·25	3·0	3·5	5·0	6·0	7 0	9·0	15·0	23·0

VALUE OF c IN $V = c\sqrt{RS}$. $n = 0·0250$. (By Kütter's formula, page 676.)

R in Feet.	S: 1 in 1,000. 1 in 1,000.	0·5 in 1,000. 1 in 2,000.	0·3 in 1,000. 1 in 3,333.	0·15 in 1,000. 1 in 6,667.	0·10 in 1,000. 1 in 10,000.	0·05 in 1,000. 1 in 20,000.
1·0	55·4	54·9	54·3	52·9	51·8	49·4
1·5	61·3	60·9	60·5	59·5	58·7	56·6
2·0	65·5	65·2	64·9	64·3	63·7	62·4
2·5	68·7	68·5	68·3	68·0	67·6	66·9
3·0	71·2	71·2	71·1	71·0	70·9	70·6
4·0	75·1	75·3	75·4	75·7	76·0	76·6
5·0	78·1	78·3	78·6	79·3	79·9	81·3
6·0	80·4	80·8	81·2	82·2	83·1	85·2
7·0	82·3	82·7	83·3	84·6	85·7	88·5
8·0	83·9	84·4	85·1	86·6	88·0	91·3
9·0	85·3	85·9	86·6	88·4	89·9	93·8
10·0	86·5	87·1	88·0	89·9	91·6	96·0
11·0	87·6	88·3	89·2	91·3	93·2	98·0
12·0	88·5	89·3	90·2	92·5	94·6	99·8

VALUES OF c IN $V = c\sqrt{RS}$. $n = 0·0225$. (By Kütter's formula, page 676.)

R in Feet.	S: 1 in 1,000. 1 in 1,000.	0·5 in 1,000. 1 in 2,000.	0·3 in 1,000. 1 in 3,333.	0·15 in 1000. 1 in 6,667.	0·1 in 1,000. 1 in 10,000.	0·05 in 1 000. 1 in 20,000.
1·0	62·5	61·9	61·2	59·7	58·5	55·7
1·5	68·8	68·4	67·9	66·8	65·9	63·8
2·0	73·2	72·9	72·6	71·9	71·2	69·8
2·5	76·5	76·4	76·2	75·8	75·4	74·6
3·0	79·2	79·2	79·1	79·0	78·8	78·5
4·0	83·3	83·4	83·6	83·9	84·2	84·9
5·0	86·3	86·6	86·9	87·6	88·3	89·9
6·0	88·7	89·1	89·5	90·6	91·6	93·9
7·0	90·7	91·1	91·7	93·1	94·3	97·3
8·0	92·3	92·8	93·5	95·2	96·6	100·3
9·0	93·7	94·3	95·1	97·0	98·6	102·9
10·0	94·9	95·6	96·5	98·5	100·4	105·1
11·0	96·0	96·7	97·7	99·9	102·0	107·2
12·0	96·9	97·7	98·8	101·2	103·4	109·0

(For the same *actual* R (such as 3·28 ft., or 1 metre) the coefficient for metric units is the coefficient for British units $\times \frac{100}{181}$, or $\frac{5}{9}$, nearly.)

Duty of Irrigation Water.

The duty of irrigation water, or the area irrigated by a given rate of flow during the season, depends upon the nature of the crop, the initial proportion of moisture in the soil, the rate of evaporation, and the rainfall during the irrigation season. It may also depend upon custom and acquired rights and is often affected by the degree of care or of carelessness exercised by the

cultivators. All figures relating to duty are, therefore, to be accepted with due reserve for purposes of estimates. The duty reckoned on the amount passing the headworks is, of course, less than that reckoned on amounts measured out to the cultivators, and the former figures will be affected by the length of the main canal and principal branches, and the rate of loss therefrom. The duty of irrigation water may be measured in cusecs continuously for the crop; in feet; or in acres per million cub. ft. stored.

On the Jamrao Canal system, India, the duty of one cusec per 50 acres has been allowed for during the inundation season, and 100 acres during the cool season. The duty has been as high as 83 acres in the inundation season, for 'dry crops' such as millet and cotton. In the Punjab 100 acres has been allowed for in the rains, and 200 acres in the cool season.

The following duties, in acres per cusec, may be noted :—

Palked canal, Bengal Deccan, 63; Ekrak tank project, Bombay, 123; Pahakarpur canal, Bombay, 165; Kabul river canal, Bombay, 114; Lower Swat river canal, Bombay, 175; Letabo canal project, Transvaal, 145; Mooi river, Transvaal, 145.

Duties of Certain Important Irrigations.

Lower Chenab: at heads of distributaries, kharif crops, 89 to 111, usually about 95; rabi crops, 217 to 284, usually about 250. At heads of canals kharif, 73 to 95, usually about 75 rabi, 190 to 237, usually about 220.

Jamrao: kharif, 85; some branches give 90 at their heads, and one gives 100; average at heads of branches, 85.

Sind: proposed under conditions to be brought about by the Sukkur project; kharif crops, 87; rice, 43½; rabi crops about 180.

Kharif Crops: Triple Canal Project, 100; Lower Bari Canal, 109; other cases, 65 to 75, and 85.

Rabi Crops: Lower Bari Canal, 200; Lower Chenab, 208; Upper Bari Doab, 234; Lower Bari Doab, 198.

Cauvery Delta: Requirements in respect of the Cauvery-Metur project were based on the yearly depths of irrigation—Single crop: existing area (*e*) 4·2 ft.; new area (*n*), 3·5 ft. Double crop, (*e*) 5·6 ft.; (*n*) 5·0 ft.

By amounts: in Kathiawar, jowari, a grain crop, requires 100,000 cub. ft. per acre, and in Jeypore State the same crop requires about 120,000 cub. ft., including losses and contingencies. In the Madras Presidency, where large areas are irrigated from numerous shallow reservoirs, the quantity, including all losses, which is stored per acre is, roughly, about 216,000 cubic ft., or nearly 5 ft. depth, for the 5 months' 'monsoon' rice crop, and 175,000 cubic ft., nearly 4 ft. depth, for a cool weather rice crop. In Central India, the storage is about 200,000 to 250,000 cubic ft. per acre.

Total Depths in Inches.

Punjab: wheat, barley, and poppy, 10·6; senji, 10·5; gram, 3·4; sugar cane, 25·3; Indian corn, 10·6; cotton, 10·6; charki, 6·3.

Punjab: the Colony Canals (H. W. Nicholson); not including preliminary watering of the ground: kharif crops; sugar-cane, 30 to 45; rice, 35 to 40; cotton, 12 to 20; maize, 12 to 18; great millet, 6 to 9; rabi crops: wheat and barley, 9 to 10; sometimes 15 for wheat; toria (*Brassica campestris*), 6 to 10; senji (*Mendicago sativa*), 15 to 20. *Bari Doab:* kharif crops, 32; rabi crops, 20.

Duty of Irrigation Water: Averages or Estimated Averages: Acres per Cusec.

(Duty × 14·3 = hectares per cu. metre per sec., or, $\frac{6,040}{\text{duty}}$ = cu. metres per day per hectare.)

Southern California, surface irrigation, 150 to 300; S. California, sub-irrigation, 300 to 500; Southern Arizona, 100 to 150; New Mexico, 60 to 80; Utah, 60 to 120; Colorado, 80 to 120; Northern Italy, 60 to 150; Madras Presidency, rice 66, 'dry crops,' 132.

The average duty of five canal irrigation systems in the Bombay Deccan, from 1894–1895 to 1900–1901, excluding two famine years, was, successively:

For *Kharif*, 70, 52, 74, 111, and 138, with a mean of 89; and for *rabi*, 60, 68, 90, 75, 85, with a mean of 76.

Costs of Irrigation Projects.

The costs of irrigation projects in India do not admit of comparison unless the nature and purpose of the project in each case be described in considerable detail. Some projects are highly productive, others are protective, and many are of an intermediate character. The conditions are very different in the United States, and, on the assumption that the works have been carried out on the narrower economic basis, the following costs may be worth noting.

COSTS OF IRRIGATION PROJECTS OF THE UNITED STATES RECLAMATION SERVICE.

Area in Thousands of Acres	216	206	164	131	129	118	103	100	72	60
Cost of water right in dollars per acre	30 to 36	22 to 30	45 to 50	55 to 66	45 to 55	22 to 30	52	30 to 35	30	45

In Nebraska, 1920, 1 per cent. of farm lands irrigated; average per farm, 147 acres; capital invested, *5l.* *4s.* per acre; annual cost of operation and maintenance, *6s.* *3d.* per acre; average depth of irrigation water on the land, 2·4 ft.

COSTS OF IRRIGATION: S. AFRICA.

With Stored Water.

Blijde River: per acre of irrigable area, *31l.* *18s.*

Calitzdorp: per acre of irrigable area, on total cost including land and roads, *120l.* *12s.*; on engineering cost, *110l.* *18s.*

Van Ryneveld's Pass: per acre of normal irrigation, *23l.* *11s.*; per acre of total area (irrigable in good seasons), *17l.* *18s.*

With Pumped Water.

On the total cost of 63 pumping projects, the cost per acre was *6l.* *17s.* Near the coast, the cost of putting 2 ft. per acre on to the land may be taken as between *1l.* and *2l.* for a 50-ft. lift, and between *1l.* *10s.* and *2l.* *10s.* for a 150-ft. lift. Up country, the costs would be about *2l.* *10s.* and *3l.* respectively. (*Legg.*)

PUMPED WATER SYSTEMS: AUSTRALIA.

Victoria.—About 11,000 sq. miles of almost streamless sandy country, supplied by more than 6,100 miles of earthen channels, supporting a population of 100,000. Minimum charge to farmers, *4d.* per acre per annum.

South Australia.—Pumping from the Tod River supplies 10,000 sq. miles of farm lands. The charge is *4d.* to *7d.* per acre.

Murray River.—In the county of Millewa, 1,000 sq. miles are supplied by pumping from this river. There are high-level and low-level channel systems, with respective minimum rates of *10d.* and *8d.* per acre.

Waste Weir Capacities.

For rates of rainfall and calculations for run-off, see Part I. For discharges of weirs, see Part II. The following formula is used in India for the discharges of the overflow weirs of storage tanks and reservoirs.

Discharge of clear overfall weirs:—

$$D = c\, b \sqrt{H^3}.$$

D = discharge in cusecs; b = width of the crest in feet; H = total height above the weir crest in feet, allowing for afflux; c is a coefficient usually about 3·333, or as low as 3·25.

In ordinary practice in the Bombay Presidency, India, it is usual to assume a general rainfall over the whole of a small catchment, and to calculate the run-off accordingly, thus:—

Catchment in Sq. Miles . . .	Up to 5.	5 to 10.	10 to 25.	25 to 75.	75 to 150.	Over 150.
Run-off in ins. of rain per hour .	2·00	1·50	1·25	1·00	0·75	0·50

In using this table special circumstances have to be considered; mountain catchments would be given larger runs-off, and for catchments wholly in the plains smaller runs-off would be assigned. (*Strange.*)

METRIC VALUES OF c IN $V = c\sqrt{RS}$. $n = 0.0250$.

(See page 710.)

R in Metres.	S: 1/40,000 0·000025	1/20,000 0·00005	1/10,000 0·0001	1/5,000 0·0002	1/2,500 0·0004	1/1,000 0·001	1/100 0·01
	c						
0·025	9	10	11	12	13	13	14
0·05	12	13	15	16	17	18	18
0·10	17	18	19	20	21	22	22
0·20	22	23	24	25	26	27	27
0·30	26	28	29	30	30	31	31
0·50	31	32	33	34	34	35	35
1·00	40	40	40	40	40	40	40
2·00	50	48	47	46	45	45	45
3·00	56	53	51	49	48	48	47
5·00	64	59	54	53	52	51	50
10·00	75	66	60	57	55	54	53
15·00	81	71	63	59	57	56	55
20·00	85	72	64	60	58	57	56
30·00	90	76	67	62	60	58	57

Note.—A curve drawn for one slope will give intermediate values for that slope and for all values of R.

AREAS—DEPTHS AND FLOW.

For the full table see Buckley's 'Irrigation Pocket Book' the figures given below suffice for the calculation of any acre-feet-time quantity, from cusecs.

Cusecs.	Acre-feet per Hour.	12 hours.	24 hours.	Week.	30 days.
1	0·08	0·99	1·98	13·88	59·5
2	0·17	1·98	3·96	27·76	119·0
3	0·25	2·97	5·95	41·65	178·5
4	0·33	3·97	7·93	55·53	238·0
5	0·41	4·96	9·92	69·42	297·5
6	0·49	5·95	11·90	83·30	357·0
7	0·58	6·94	13·88	97·18	416·5
8	0·66	7·93	15·87	111·10	476·0
9	0·74	8·92	17·85	124·95	535·5

SILTING OF RESERVOIRS.

Muchkundi: capacity, 704 million cub. ft.; catchment, 26 sq. mls.; accumulated silt, in 22 years, 58 million, and in 26 years, 64 million cub. ft.; respective rates, 101,000 and 95,000 cub. ft. per sq. ml. of catchment per annum. Capacity reduced 8·2 per cent. in 22 years; 9·9 per cent. in 26 years.

Gokak: capacity, 908 million cub. ft.; catchment, 1,080 sq. mls.; accumulated silt, in 18 years, 155 million, in 19 years 174 million, and in 20 years, 189 million, or 8,750 cub. ft. per sq. ml. of catchment per annum. Capacity reduced 28 per cent. in 20 years.

Lake Fife: not having sluices for silt, in the dam; at 14 ft. above sill, area of water surface: in 1883, 88·76 million sq. ft.; in 1905, 72·34 million sq. ft.; loss of capacity in 22 years, 16·33 million cub.ft., or 18 per cent.

Lake Whiteing: having sluices for silt, in the dam; no appreciable change in capacity in 5 years.

The O'Shaughnessy reservoir, on the Scioto river, Colombus, Ohio. Waterspread, 829 acres; length, 8 mls.; greatest width, 1,900 ft.; average depth, 20 ft. Original capacity, 16,673 acre-ft.; volume of silt deposited in 9 years, 1,016 acre-ft., or 6·1 of the capacity and equivalent to 0·02 in. off the catchment of 988 sq. mls. Specific gravity of the silt, 2·66. Including 2·21 ins. in 1934, the average annual rainfall of the 9 years was 10·5 ins. On the same river, the average annual rate of silting in the Griggs reservoir, during 30 years, was 0·8 of 1 per cent.

Mahadeo Nala Reservoir, Bombay Presidency. Catchment, 13·5 sq. mls.; water impounded, 1911, 47·47 million gals.; reduced by silting to 33·25 million gals. in 1927; a reduction of 38·9 per cent. or 2·43 per cent. per annum.

Proportions of Silt Borne by Rivers and Canals.

Reputed maximum proportions of silt to water, by weight: Rivers—Mississippi, 1/572; Nile, in August, 1/666; Indus, at Sukkur, 1/163 to 1/198, velocity 6·4 to 8·3 ft. per second; a tidal river in East Bengal, during six days, ranged from 1/455 to 1/25. The Vaal, rising, and 3 ft. 6 ins. over weir, 1/268; falling, and 1 ft. 7 ins. over weir, 1/630; Rhône, velocity 8 ft. per second 1/45; Vistula, velocity 10 ft. per second, 1/48; Kistna, June, 1/42; Vaal River, 1/400.

Cub. ft. of damp silt per million cub. ft. of water: Sirhind canal (1897, 32,600 million cub. ft. intake), 5,464, (1895, 43,000 million cub. ft. intake) 1,650, (1896, 54,600 cub. ft. intake) 2,343. Sone canals, June 1,785; July, 3,250; Aug., 2,750; Sept. 625; Oct., 62. Guadalquivir, on successive dates, March 30 to May 2, 2,321, 775, 521, 695, 2,048, 596 .273, 222, 198.

Irrigation Systems.

Typical South African Irrigations.

Van Rynveld's Pass: catchment, 1,477 sq. mls.; average rainfall, 14½ ins.; area irrigated, max., 22,314 acres; average, 17,000 acres. Dam, length, 1,170 ft.; crest width, 10 ft.; clear height (max.), 109 ft.; height plus deepest foundation, 152 ft.; capacity of reservoir, 64,000 acre-ft. Cost 400,000*l.*

The Hartebeestpoort River: irrigation controlled by the Hartebeestpoort Dam; altitude of river bed, 3,827 ft.; catchment, 1,560 sq. miles; average rainfall near the dam, about 23 ins. Dam: height above river bed, nearly 190 ft.; thickness at base, 73 ft.; radii at base, downstream face, 75 ft., upstream face, 148 ft.; increasing radii to vertical top section where the upstream face radius is 240 ft.; one tangential abutment; crest and top section width, 15 ft.; both batters 1 in 5·714; freeboard above full supply level, about 22·75 ft., capacity of reservoir, at waste weir crest level, 136,241 acre-ft.; capacity above outlet level, 123, 200 acre-ft.; waterspread at F.S.L. (waste weir crest level), 6·7 sq. miles; 9·2 sq. miles available.

Orange River.—A scheme for the control of floods and the supply of irrigation water involves the construction of a dam, at an estimated cost of £1,000,000.

Vaal River Scheme.—Also called the 'Vaal-Hartz' scheme. It has cost round about £7,000,000, and includes the following works.

The Vaalbank Dam.—See Part III, page 697.

Loskop.—The dam, averaging 100 ft. in height and creating a lake 15 ml. long, is situated 140 ml. from Johannesburg and 32 ml. from Middleburgh. The area to be irrigated, about 18,000 morgen or 38,000 ac., is in the Pretoria district, 100 ml. from Loskop. The works included driving a tunnel 3,000 ft. long.

Riet River.—The dam has created a storage of 298,000 acre-ft., for the irrigation of 23,000 ac. The works included a tunnel 11,000 ft. long.

Lindleyspoort Dam.—Storage, 9,840 acre-ft.

Egmont Dam.—Storage, 10,400 acre-ft., for the irrigation of about 1,700 ac.

Irrigation in New Zealand.

Systems in Operation: Totals:—*number:* twelve or thirteen. *River discharges*, minima, 295 cusecs. *Main Canal Discharges:* maxima, 795 cusecs, designed discharges; attained in 1929–1930, 483 cusecs. *Rainfalls:* mean of averages as estimated from records available, 18·55 in. *Gross Area Commanded:* 71,829 acres. *Area Irrigated at Present:* 42,695, in 1929–1930. *Main Canals:* 300 miles; average 23 miles. *Distributaries:* 343 miles; average 26·4 miles.

The above totals include the systems:—*Steward Settlement:* supply from Waitaki River; average rainfall 20·94 in.; main canal, maximum designed discharge, 110 cusecs; gross area commanded, 18,000 ac.; main canal 15 miles; distributaries, 50 miles.

***Ida Valley*: supply from streams, and storage by Manorburn Dam; average rainfall, 16·07 in.; main canal discharge (max.), 110 cusecs; gross area commanded, 14,000 ac.; area at present irrigated, 11,440 acres; main canal, 73 miles; distributaries, 54 miles.**

The Thal Irrigation Project.—The most recent of the large irrigation systems in India. Fully described in *Civil Engineering* (London), August 1942.

***Upper Manuherikia Project*: sources of supply, Dunstan River and a storage dam at the falls on Manuherikia River; average rainfall 20·81 in.; mimimum rivers' flow, 77 cusecs; maximum discharge of main canal, 500 cusecs; gross area to be commanded, 96,000 acres; main canal, 130 miles.**

Indian Irrigation Systems.

Sind.—Dominated by the recently completed Sukkur Barrage and canals served by it.

Barrage 4,926 ft. long, with 66 openings of 60 ft.; gates, 63 ft. 3 ins. by 18 ft. 6 ins. in height; The Rice Canal, capacity, 12,846 cusecs; length, 87 miles; North-West Perennial Canal, 4,313 cusecs, 97 miles; South-East Perennial Canal, 2,767 cusecs, 140 miles; Rohri Perennial Canal, 10,250 cusecs, 205 miles; Eastern Nara Supply Channel, 12,200 cusecs, 15 miles; 242 miles of River Nara canalised. Totals of areas to be irrigated, acres: rice, 823,000; cotton, jowari, etc., 1,739,000; rabi crops, 3 338,000. Cultivation acreage, total, 5,900,000 annually in a total commanded area of 8,132,000 acres. Estimated annual tonnages of crops after 30 years' development: rice, 515,000: cotton, 190,000; jowari, etc., 575,000; wheat 935,000.

Estimated cost about 20 crores, or £15 millions.

Total quantity of earthwork in canal construction. and major drainage works, 5,690 million cu. ft. Relating to the barrage works and regulators (millions of cu. ft.): excavation, 87·92; concrete work, in cement and in lime, 1·24; rubble masonry in foundations, 8·20; superstructure masonry, 3·40; cut stone work, 0·17; reinforced concrete work, 0·64; concrete block work, 0·7; stone pitching, 15·51; also pile driving, permanent piling, 0·71 million sq. ft.

The River Indus, the flow of which is thus regulated, at a point virtually the head of a vast delta, has an average flow of 110,000 cusecs and a minimum flow of 17,567 cusecs. The greatest recorded flood was about 1,100,000 cusecs.

Bombay Presidency.—Lloyd Dam, at Bhatgar: completed October, 1928; catchment, 128 sq. miles; capacity of reservoir, 555,555 acre-feet; height above lowest foundations, 190 ft.; crest length, 5,333 ft.; 81 sluices, of which 45 are automatic; volume of masonry in the dam, 21,500,000 cu. ft.; area commanded, 834,000 acres, of which 202,000 will be irrigated annually; cost of dam, 1,260,000*l.* of the whole project, including the two main Nira canals, 106 and 100 miles long, cost 4,140,000*l.*; annual value of crops grown on the irrigated area, 2,380,000*l.*

Cauvery-Metur System.—Effects improvements, regulation and extension of irrigation in the river basin, chiefly in the delta. Catchment of reservoirs 15,700 sq. mls. fed by the South-West Monsoon on about 10,000 sq. mls., receiving also the North-East Monsoon rains. At the head of the delta the average yearly discharge of the Cauvery, June to January, is 425,000 million cub. ft., of which the South-West Monsoon, June to mid-October, supplies 335,000 and the North-East Monsoon, mid-October to January, 90,000 million. The Upper Anicut is about 64 mls. from the coast, the Grand Anicut about 20 mls. farther downstream. The Cauvery delta irrigation is about 895,000 acres in extent. The Lower Coleroon (delta branch) Anicut controls the irrigation of about 108,000 acres. Non-deltaic irrigation, Bhavani, 39,000 acres; the Noyel, 14,500 acres; Amravati, 44,000 acres; Cauvery, 62,500 acres.

United Provinces: Tube Wells.—The Ganges Canal Hydro-electric Grid has made available 15,000 kW. for pumping for irrigation. Seven hydro-electric generating stations on the falls of the Upper Ganges Canal, the total installed capacity of which is 18,900 kW.; a steam station at Chandaussi, 9,000 kW.; transmission system 4,525 mi. in length. The pumping load includes power for 1,500 tube wells, the average yield of which is 1·2 cusecs each; for the Ramganga pumped canal, carrying 150 cusecs; for a distributary fed from tube wells carrying about 80 cusecs, and for about 300 private tube wells. All those pumping installations command about 1·6 million acres. The allotment of power for irrigation pumping is 25,000 kW., out of a total installed capacity of 35,000 kW. In 1938–39 about 700,000 acres were irrigated.

The Emerson Barrage.—Controls the flow (900 to 650,000 cusecs) of the Chenab. Provides up to 7,750 cusecs for Haveli project canals. Length, 3,026 ft.; the weir section has 37 spans of 60 ft. with single gates 15½ ft. high. Sluice sections; left 8 spans, right 6 spans each of 30 ft., with double gates of total height 21 ft. All piers except those, 25 ft. wide, between weir section and sluice sections, are 7 ft. thick. The difference in level between gate tops and cistern floor is 29 ft. Areas affected by the project: *Old Area*, to be irrigated—(*a*) perennially, 547,000 acres; (*b*) non-perennially, 369,000 acres; *New Perennial Area*: (*c*) Crown-waste, 155,000 acres; (*d*) proprietary, 25,000 acres; *New Non-Perennial Area*: (*e*) Crown waste, 51,000 acres; (*f*) proprietary, 325,000 acres; total, 1,472,000 acres.

IRRIGATION IN AUSTRALIA.

The following notes are from an article by J. M. Antill, in *Civil Engineering (London)*, January 1938.

Rainfall.—More than 36 per cent. of the continent has an annual rainfall of less than 10 ins.

[More than two-thirds of the area of 3,000,000 sq. ml. has an annual rainfall of less than 20 ins. Away from the coastal districts evaporation is estimated to equal the rainfall when that is about 36 ins. (T. R. East.)]

Victoria.—One-fourth of the State is now watered by artificial means, at a capital expenditure of £25,000,000.

New South Wales.—An area of over 8,000,000 acres is under the control of Water Trusts.

South Australia.—Irrigation systems cover 20,000 sq. mls. There is a large system of pipe reticulation, totalling about 6,000 miles of water supply mains. On Eyre's Peninsular a large tract of wheat land is reticulated, the trunk main extending 240 miles from the reservoir.

Bore-Wells.—There are five large and three smaller artesian basins on the continent, including the Great Basin 570,000 sq. mls. in area. Both artesian and pumped supplies are derived from government bore-wells, of which there are 100 in South Australia, 40 to 50 in north-west Victoria; 200 in New South Wales, yielding 63 million gallons per day, also 250 private bore-wells yielding 150 million gallons per day; 800 artesian and 300 pumped bore-wells in Queensland. The cost of boring is usually 14*s.* to 42*s.* down to 2,000 ft.; at 4,000 ft. the range is from 28*s.* to 65*s.*

IRRIGATION IN NORTH AMERICA.

Irrigation System, San Diego County, California.—Average annual rainfall, 9·75 in. with an increase of about 6 in. per 1,000 ft. of altitude on the Pacific slope. Longest drought 1897–1904, practically no run-off. A low run-off at intervals of 11 years. Area of district served, 16,772 acres, of which 12,390 acres are entitled to water; assessment values, watered 156·8 dollars, unwatered, 6·04 dollars, per acre. Cost of water for irrigation, from 5 dollars per acre-ft. for lands with shallow wells, to 30 dollars for 'the larger agencies.' *Duty.*—Average 1·13 acre-ft. per acre. Assessment charges, inclusive, 7 to 8 dollars per acre.

A Rio Grande Project.—A Mexican Government project, to create an irrigated area, or an area mostly irrigated, of 150,000 acres in the lower Rio Grande valley, opposite the irrigated land on the United States side of the river. Estimated cost of the dam and canals, $8,600,000.

Don Martin Project, Mexico. (See Rio Salado Dam, p. 695.) The irrigable area of 160,000 acres is supplied from a reservoir of 48,412 acres area and 1,123,600 acre-ft. capacity. Discharge capacity of the overflow weir, with 21·1 ft. depth over sill, 210,000 cusecs. Main canal, bottom width 60 ft., depth of water 10·6 ft.; computed velocity 2·78 ft. per sec.; discharge 2,239 cusecs. Length to first branch 26 mls. beyond that 75 mls. See 'The Don Martin Project.' By A Weiss. *Proc. Am Soc. C.E.*, December 1930.

Central Valley Project.—California: Works are in hand. The scheme is for bringing the surplus waters of Northern California southward to the delta and Suisun Bay regions near San Francisco and to parts of the San Joachin Valley. Estimated cost, 170,000,000 dollars. The works include the formation of Kennett Reservoir, capacity nearly 2,950,000 acre-feet, by the construction of Shasta Dam, 420 ft. in height; also San Joachin Reservoir, capacity 450,000 acre-ft., formed by a dam 250 ft. in 'height.'

The project may be described (1939) as intended to check retrogression, control floods and provide permanent irrigation (see Shasta and Friant dams, p. 697) by storage of water in the San Joachin and Sacramento rivers. There will be 350 miles of main canals and a series of pumping stations and canals for irrigating high lands. Estimated cost, £35,000,000. (Description and map, *The Engineer*, December 16, 1938.)

Columbia Basin Project.—State of Washington. Designed hydro-electric plant capacity, 2·7 million h.p. Area to be irrigated, 1·2 million acres of semi-arid lands, on which the normal annual rainfall is about 10 ins. (See *The Engineer*, February 4, 11 and 18, 1938.)

River Flows.

Fluctuations of the Vaal River.—(W. Van Warmelo). A table gives for eight gauging stations on the River Vaal and its tributaries the average flow for each month of the year, in one case based on 2 years' observations, in two cases 8 years, in other cases 9 to 17 years. A rough averaging of the 8 cases gives the following months' flows, as percentages of the average 12 months' flow: Oct., 4; Nov., 10; Dec., 12; Jan., 17; Feb., 21; March, 20; April, 7; May, 2; June, 1½; July, 1½, Aug., 1½; Sept., 3. For the River Vaal, at Standerton, the figures, based on 16 years' observations, were: Oct., 5·56; Nov., 14·99; Dec., 14·75; Jan., 27·15; Feb., 20·12; March, 11·49; April, 2·39; May, 1·05; June, 0·63; July, 0·53; Aug., 0·72; Sept., 0·62.

Run-off of the Thames Basin, in 1932 : 37 per cent. on 10·43 ins. from 28·2 ins. of rain. The rainfall distribution was unusual. Over the basins of the Thames and Lea the summer months' rainfall was 8·97 ins. more than that of the winter months, whereas, normally, the winter months' rainfall exceeds that of the summer months by 2·47 ins.

Thames and Lea Basins.—Rainfall July-December, 1933, 8·9 ins., the lowest in 51 years; in December, 0·48 ins., comparing with the average, 2·99 ins. Deficiency in the 6 months, 413,000 million gallons (66,080 cub. ft.). Flow of the Thames in February, 500 million gallons per day, or 926 cusecs, comparing with the average 2,399 million gallons per day, or 4,443 cusecs.

Flow of the River Severn.—From a paper (Inst. C.E., April, 1937), by S. M. Dixon, G. Fitzgibbon and M. A. Hogan. Relates to flows measured at Bewdley, from the catchment of 1,632 sq. mls., during the 15 years, 1921–1936, the water-year being October 1 to September 30.

Years' Discharges, expressed as inches of depth on the catchment: average, 18·90; maximum, 26·63, or 141 per cent. of the average; minimum, 8·49, or 54 per cent. of the average; average of the wettest three consecutive years (1929–1932) 24·19 ins.; of the driest three consecutive years (1932–1935) 12 69 ins.

Run-offs; consecutive percentages of the 15 years average: 85, 99. 117, 108, 83, 110, 122, 72, 141, 139, 104, 81, 45, 75, 119.

Corresponding Rainfalls, expressed as percentages of the 15 years average: 96, 94, 114, 105, 95, 118, 100, 78, 133, 123, 100, 76, 68, 89, 110. The average for the 15 years was 38·14 ins.; for the period 1881–1915, it was 34·6 ins.

Canal Excavation: The Dragline.

Outputs and Costs.

Sukkur Barrage Canals System.—Costs per cu. yd., in pence: (1) big steam draglines, 3·5; (2) medium-size, steam, 2·8; (3) Diesel-electric, 1·9; (4) small Diesel, 1·4. Fuel costs per cu. yd.: Coal, (1) 1·2; (2) 1·0 Diesel oil, (3) 0·3; (4) 0·14. (W. Barnes.)

Cauvery-Metur System Canals.—Diesel-electric, crawler-mounted draglines, about 1·2*d.* per cu. yd. The conditions were unusually difficult. (W. Barnes.)

Salonika Plain Reclamation Works.—Small draglines (*s*) and large ones (*l*); horse-power per cu. yd., *s*, 84 to 88; *l*, 63; h.p. per ton weight: *s*, 2·4 to 3·0; *l*, 0·83; weight, tons per cu. yd. capacity: *s*, 28 to 36; *l*, 76; ground pressure, lb. per sq. in.: *s*, 11 to 13; *l*, 22½; average output, cu. yd. per hour per cu. yd. of bucket capacity: *s*, 46 to 72; *l*, 31. (B. W. Huntsman.)

Operating costs in cents (U.S.A.) per cu. yd. are given with reserve, owing to fluctuating exchange rates, but the relative costs of items are of interest. Mr. Huntsman gives these for (*a*) Diesel, and (*b*) steam draglines on land, and for (*c*) Diesel-electric, and (*d*) steam, afloat; costs in cents (U.S.A.) per cu. yd.: (*a*) 2·29; (*b*), 4·48; (*c*), 3·25; (*d*), 5·00. The items for (*a*) were (cents): fuel, 0·20, or 8·73 p.c.; lubricants, waste, etc.: 0·25 or 10·91 p.c.; spares and repairs, 0·80 or 39·94 p.c.; labour on machine, 0·82 or 36·25 p.c.; labour on repairs, 0·21 or 9·17 p.c.

Recent Literature.

'The Salonika Plain Reclamation Works.' By B. J. Huntsman. Paper, *Journal Inst. C.E.*, March 1937.

'The Dragline Excavator.' By W. Barnes. Paper No. 5217, *Journal Inst. C.E.*, March 1940; discussion on the paper, *ibid.*, oral, March; written, October 1940.

'Methods of Excavation Work at Home and Abroad.' Dugald Clerk Lecture, 1940, by W. Barnes. *Journal Inst. C.E.*, January 1941.

'Underground Supplies of Water in the Trap-Rock Zone in the Bombay Deccan and Other Allied Tracts.' By N. S. Joshi. *Journal Inst. E. (India)*, April 1941.

'The Development of Irrigation.' By E. Bruce Ball. *The Engineer*, February 23, 1940. A part of a presidential address entitled 'The Influence of the Mechanical Mind on the Development of Irrigation throughout the Ages.'

'Staining Natural Rivers Sands for Studies of Sediment Movement.' By R. G. Grassy. *Civil Engineering* (U.S.A.), November 1941.

'Indian Irrigation at the Outbreak of War.' By R. A. Ryves. *Civil Engineering (London)*, March 1944.

The Flood Problem in Iraq.' By E. V. Richards, *Journal Inst. C.E.*, April 1945.

Addendum 1949.

Costs of Irrigation. The costs given on pp. 712, 714 and 715 are, of course, much lower than present costs. They are, however, of full significance with regard to relative costs of different forms of irrigation, and in different countries.

SECTION XVIII

PART V

THE DEVELOPMENT OF WATER-POWER.

(Revised by R. A. Ryves, M.Cons.E.)

WATER-POWER.

The theoretical power or effective energy of a fall of water, in ft.-lbs. per min., is equal to the weight in lbs. of the volume of water flowing per min. over the fall, multiplied by the vertical height of the fall in ft.

$$\text{H.P.} = \frac{Q \times h \times W}{550} = \frac{Q \times h \times 62\cdot5}{550} = \frac{Q \times h}{8\cdot8},$$

where,

Q = discharge in cub. ft. per sec.; W = weight of 1 cub. ft. of water = 62·5 lbs.; h = net head or net fall of water in ft.; H.P. = horse-power (33,000 ft.-lbs. per min. or 550 ft.-lbs. per sec.).

Thus 1 cu. ft. of water per sec. falling through a head of 10 ft. will develop theoretically 1·134 h.p.

For practical purposes, and in order to take into account the losses in the water-wheels, it is better to adopt the following formula, which is based on a water-wheel efficiency of 80 per cent.,

$$\text{Practical H.P.} = Q \times h \div 11.$$

An efficiency of 80 per cent. is now attained even in the medium-priced wheels, and 90 per cent. is frequently reached in the high-priced wheels. Assuming an efficiency of 88 per cent., then

$$\text{Practical H.P.} = Q \times h \div 10.$$

In metric units one cubic metre of water falling through one metre develops 13·15 British horse-power.

***Comparison with Coal Power.*—10 tons of water (about 358 cu. ft.) falling 100 ft. develops the energy equivalent to that which can be produced through the burning of one pound of coal. That is, 10 tons, or 20,000 lbs., falling 100 ft., will develop 2,000,000 ft.-lbs. Dividing this by 778 gives 2,570 B.Th.U. of energy. Assuming a hydraulic turbine efficiency of 92 per cent., which is a fair average for a modern hydraulic turbine, the energy actually developed from this water is about 2,370 B.Th.U. This corresponds to the energy made available by the combustion of coal having a heating value of 13,000 B.Th.U. per lb. with an over-all plant thermal efficiency of approximately 18 per cent., which is perhaps fairly representative of good central-station practice although not equal to the best, which approaches 25 per cent.**

***Available Power from Rainfall (W. T. Taylor).*—In terms of *average* of rainfall, catchment area and head, the available power can be expressed:—**

$$\text{H.P.} = 0\cdot00828\, f\, m\, h\,;\quad \text{B.H.P.} = 0\cdot0073\, f\, m\, h\,;\quad \text{kW} = 0\cdot0062\, f\, m\, h\,;$$

where

f = average rainfall in inches; m = catchment area in sq. miles; h = average head in ft.

***Example.*—The catchment of the Lochaber water-power scheme is 303 sq. miles; the average annual rainfall is 73 in.; and the average head of water available near Fort William is 742 ft. What will be the theoretical power?**

$$\text{H.P.} = 0\cdot00828 \times 73 \times 742 = 136{,}000 \text{ H.P.},$$

and, as 1·0 H.P.-year = 1 × 8,760 × 0·746 = 6,535 kW-hours, the output (theoretical) would be

$$6{,}535 \times 136{,}000 = 888\cdot76 \textit{ million kW-hours.}$$

Tide-Power.

Depending upon the tidal and topographical conditions, tide-power may be developed: (a) directly, with the small and varying falls and the natural flows; (b) by utilising that power to pump water to embanked reservoirs on adjoining land; (c) similarly, but pumping a proportionate volume of water to a high-level reservoir.

The possibilities of tidal power, if it can be developed commercially, are very great.

In Great Britain the highest tides are found in the estuary of the Severn, the mean range of the spring tides at Chepstow being 42 ft., and of the neap tides 21 ft. In France the maximum

range occurs at St. Malo, where it amounts to 42·5 ft. at spring tides, and about 18 ft. at neap tides. The tidal range in the Dee is 26 ft. at springs, and 12 ft. at neaps, while the mean range of spring tides around the coast of Great Britain is 16·4 ft., and of neap tides 8·6 ft.

Assuming a mean tidal range of only 20 ft. at springs, and 10 ft. at neaps, and adopting the single-basin method of development with operation on both rising and falling tides, each square mile of basin area would be capable, without storage, of giving an average daily output of approximately 110,000 h.p.-hours. In such an estuary as the Severn, where an average of 20 square miles could readily be utilised with a spring tidal range of 42 ft., the average daily output without storage would be approximately 10,000,000 h.p.-hours.

(*Nature*, June 3, 1920.)

Severn Estuary.—Near the site of the proposed barrage and power station, the mean water levels are: high water, springs 23·6 ft. and neaps 12·7 ft. above O.D.; low water, neaps 9·7 ft. and springs 17·1 ft. below O.D.; mean tide, 2·4 ft. above O.D. (See *The Engineer*, March 31, 1933.)

Wind Power (Ryves).—Inland, as well as on the coasts, wind power can be utilised by conversion to water power (as (*c*) at the bottom of the proceding page). Inter-dependent factors are: (1) reservoir capacity; (2) the significant period of least wind power—as a shorter period with little or no wind, or a longer period of low average wind power; also water heads, that pumped and that serving the hydro-electric plant, being included as numerical factors; (3) the capacity of the wind-engine plant; (4) the capacity of the hydro-electric plant. For a period long enough to give, substantially, the average of wind power, or wind duration-force, curves of reservoir inflow and outflow can be drawn, the former for a number of different wind-engine plant capacities and the latter for a number of different power-station capacities; and the required size of reservoir can be deduced therefrom. Should that be greater than the available size, the diagram, studied with inflow and outflow curves for the maximum available size, will give the wind-engine plant capacity. The costs of impounding the water and those of providing wind-engines should then be compared and—further curves being drawn as required—the most economical provision determined. If alternative, higher and lower, reservoir sites be available, the cost of installing power units has also to be considered, and, as a minor item, the cost of the pressure pipe. Final calculations can be precise, and the only serious obstacle to such projects is with regard to amenities.

A wind-power electric plant on the top of a 2,000 ft. mountain in Vermont is built on the top of a tower 110 ft. high. The wind turbine has a 24 in. shaft and stainless steel blades 65 ft. long. The turbine drives, through double helical gearing, a 1,000 kilowatt, 60-cycle, 3-phase generator running at 600 r.p.m.

Practical Horse-power corresponding to Heads in Feet and Cubic Feet of Water used per Minute.

(1 cu. ft. per min. = 1·7 cu. metres per hour. 1 cu. metre per hour = 0·59 cu. ft. per min

Writing the equation on the preceding page,

$$\text{Practical H.P.} = \frac{Q \times h}{8 \cdot 8} \times f,$$

where f is the percentage efficiency; and making $f/8 \cdot 8 = 1/K$, the equation becomes,

$$\text{Practical H.P.} = (Q \times h)/K,$$

and the values of K corresponding to efficiencies from 80 per cent. to 90 per cent. are:—

80 p.c., 11·0; 81 p.c., 10·86; 82 p.c., 10·73; 83 p.c. 10·6; 84 p.c., 10·48; 85 p.c., 10·35; 86 p.c., 10·23; 87 p.c., 10·11; 88 p.c., 10·0; 89 p.c., 9·89; 90 p.c., 9·78; 91 p.c., 9·67; 92 p.c., 9·57.

Instead of the table formerly given (pp. 720 and 721 in the 1941 and earlier editions), the engineer can draw a graph for the full ranges of heads and flows considered in any particular investigation, and for any adopted value of f. The graph can be to a relatively open scale if zero on the diagram be taken as the minimum head and minimum flow. Three calculations will give the straight line graph.

General Features of a Water-Power Development.

Although the lay-out of hydraulic plants changes with every water-power site, there are certain features which are generally found in all developments, and which can be described as follows:

1. The dam, which impounds the water, regulates the level of the upper pool, and directs the water towards the water wheels. 2. The head gates, which control the supply of water, and by means of racks permit the removal from the water supply of injurious floating and suspended matter. 3. The head race, flume or penstock, which carries the water to the wheels. 4. The power house, which contains the hydraulic machinery. 5. The water wheels, which transform the hydraulic energy. 6. The draft tube, and the tail race, which discharge the water used in the power house into the river.

In some plants the power house forms a part of the dam, and the water is delivered to the wheels without passing through a canal or penstock and discharged back into the river directly through the draft tubes.

Fig. 1 shows a rather complicated lay-out, with a regulating canal for the head race. The regulation of the level of the upper pool is obtained either by a regulating weir or sluices in the dam, or shutters (sluices) on the crest of the dam.

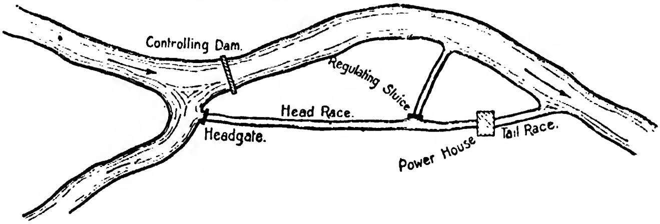


FIG. 1.—Typical General Arrangement of Hydraulic Development.

In approaching the study of any water-power site, the two first things to be determined are the head of water available at all times and the characteristic flow of the river. It is not economical to instal machinery which can utilise only the minimum flow of the river, and it is considered good practice to build a plant capable of using the *characteristic flow* of the river, which can be defined as the available dependable flow during at least nine months of a low year.

The available head is determined by levelling and by the daily reading of gauges located in the upper and lower pools of the proposed development.

The *characteristic flow* is found by a series of discharge measurements combined with the daily reading of gauges extending over as long a period as economically possible.

AVAILABLE DISCHARGE.

(*Contributed by R. A. Ryves.*)

Graphical Records and Computations.—The records of gaugings supply the basis of computation. If they do not extend far enough back to cover periods of nearly maximum and minimum flow, including both short and long periods, the period examined may be extended on the basis of rainfall data. (See Part I.)

A base is drawn representing elapse of time to scale. On ordinates to this base, flow records and anticipated drafts on the supply are marked and the curves of flow and draft thus drawn. The area of each curve, that is, to scale, the volumes of water, can be shown—as increasing amounts from zero time and as amounts between any two ordinates marking points in time—by means of sum curves. The areas enclosed between any two ordinates and between the flow and draft curves represent volumes available for storage or to be drawn from storage in the period between the two points in time.

Arithmetical Tabulation.—It is convenient to accompany the graphical record and computations by arithmetical computations, which are tabulated, with columns for progressive totals. No other mathematical work is needed. Thus, for any period of time, the average and total discharge or draft can be found by drawing a balancing straight line through the intercept on the curve.

Reservoir Capacity.—The curves will show when in the whole period examined there would have been flow in excess of that which could be stored in the reservoir. Or the curves (or the tabulated figures) will show what storage capacity is necessary in order that no water may be wasted. The costs of construction of reservoirs of this and a series of diminishing capacities are considered in relation to the money advantage in each case of the respective output of the power-station. Usually, a minimum and a maximum capacity, the latter not intended to provide for the storage of all the water in the wettest rainy season or year on record, will be decided upon by judgment, and that range of capacities examined as described above.

Provision for Floods.—The curves will show the effect of the reservoir in abating floods.

Opening Sluices in the Dam.—If the provision of sluices at levels considerably below top-water evel in the reservoir is contemplated, or to be considered, as a means of lowering the water in the reservoir in order further to abate dangerous floods, a curve is drawn on the diagram representing the varying flow through these sluices—with diminishing head between the time of receiving the flood warning up to the time when the discharge from the reservoir only equals the inflow, and with increasing head as the inflow exceeds that outflow. Similar computations to those already described will indicate, precisely, how far the sluices for which the curve of their flow is drawn are too many or too few. Or the number and depth of the sluices may be directly computed from a curve showing the difference between inflow on the one hand and, on the other hand, the draft to the power-station, the flow over the waste weir and the discharge of such sluices as would in any case be provided.

The procedure applies to all cases in which determinations are required of the effect of a reservoir in abating floods.

Head Gates.

Head gates are placed at the entrance of the head race, in order to permit the emptying of the canal and pipes in case of repairs, and in order to be able to stop quickly the admission of water in case of accident to the canal, flumes, pipes, or water wheels.

The head gates are, of course, built differently, according to the size of the opening they have to close.

Head gates for a canal consist of wooden or steel gates moving between concrete, masonry, or crib piers. The weight of the heavy gates is generally partly balanced by a counter weight, and they are raised and lowered by electric or hydraulic motor. The smaller gates are moved by hand with the help of a mechanical hoist.

The piers must be designed to withstand load equal to the total water load between the centres of two adjoining openings. (See Part III, 'Parallel Sluce Dams,' p. 683.)

When pipes are used to carry the water to the turbines they are generally protected by a concrete head wall designed to resist the full hydrostatic pressure. The gates in this case are much smaller, and are operated by hand, or mechanically (as electrically) operated valves are provided.

The force necessary to raise the gate can be calculated as follows :—The total weight of gate in water plus the total water load on the gate must be multiplied by a coefficient of friction which varies according to the materials used. To this weight should be added the weights represented by the mechanical losses in the transmission of power, and the grand total increased by about 18 per cent. in order to provide for the extra force necessary to start the gate.

Racks.

Racks are usually made of flat bars having sections from $\frac{5}{16}$ in. by 1 in., to $\frac{3}{8}$ in. by $3\frac{1}{2}$ ins. held together by bolts and separators. These racks may be either of large or small spacing, the former being those in which the bars are from $1\frac{1}{2}$ ins. to 4 ins. apart, and the latter where the bars are from $\frac{3}{8}$ in. to $1\frac{3}{8}$ in. apart. The spacing is determined by the cross section of the openings of the distributor and runner buckets of the water wheels.

Sometimes two racks are provided, to facilitate the removal, without loss of time, of the larger floating objects brought down by the stream.

The racks are generally placed at an angle varying from 45° to 60° with the vertical; there should always be an operating platform at least 3 ft. wide, so placed that the attendants can work to best advantage either when cleaning or raising the racks.

The racks should be designed on the assumption that they act as a solid dam, and that all water behind may be drawn off. In such a case the racks and their supports would have to take care of the entire water pressure.

For the purpose of keeping floating *débris* from choking the gratings at the circulating water intakes at an American power station, a pipe has been installed just above water level in front of the grid. The pipe is drilled with $\frac{1}{4}$-in. holes, and is supplied with water at 120 lbs. per square inch. The jets of water issuing from these holes wash away any leaves, sticks and snow that might clog the grating.

Design of Head Races.

The cross-section area of a head race should be such that the water will rise to about three-quarters to seven-eighths of the height.

Wooden flumes are ordinarily rectangular and designed to carry a depth of water equal to about half the width.

In other respects head races may be designed as are canals (see Part II for basic hydraulics and Part IV for other information).

Design of Pipes.

Pressure in Pipes.

The interior pressure in a pipe due to the presence of water, and tending to burst this pipe, is larger transversely than longitudinally.

The pressure in pounds per sq. in. p, determined by a head of water H expressed in feet, is :

$$p = \text{pressure in lbs. per sq. in.} = \text{head in. ft.} \times 0 \cdot 44.$$

and the diametral pressure P, tending to burst the pipe, is represented by the equation,

$$P = p \times d/2$$

where d = diameter of pipe in ins.

The longitudinal tension P in the pipe is given by,

$$P' = p \times d/4$$

This explains why steel riveted pipes have generally a single row of rivets at the circumferential joints and two rows of rivets along the longitudinal joints.

Design of Steel Pipes.

The thickness of metal t (expressed in ins.) required to carry a certain pressure can be determined by the following formula:

$$t = \frac{(p+p')d}{2f.e} + c$$

where, t = thickness of steel shell in ins.; d = inside diameter of pipe in ins.; p = pressure of water in lbs. per sq. in.; p' = allowance for water hammer in lbs. per sq. in.; f = allowable tensile stress of steel in lbs. per sq. in.; e = efficiency of riveted joint; c = thickness to be added to plate for corrosion.

The permissible unit tensile stress with ordinary open-hearth steel is assumed at 16,000 lbs. The efficiency of the joint may be taken at 55 per cent. for single riveted pipe, and at 70 per cent. for double riveted pipe. The thickness to be added to compensate for the weakening of the pipe due to corrosion is about $\frac{1}{16}$ in. In large pipes under small pressure, the thickness of metal computed with the above assumptions must still be increased to give the pipe the necessary stiffness.

Design of Reinforced Concrete Pipes.

Reinforced concrete pipes are designed under the assumption that the steel takes care of all longitudinal or transversal stresses in the pipe whilst the concrete acts as a waterproof shell. It is evident that, in this case, the steel can be reduced to the minimum thickness required.

The permissible stress per sq. in. of metal can also be taken at 16,000 lbs. per sq. in.

The area of the circles forming the transversal reinforcement can be determined as follows:

Let p = total internal pressure in lbs. per sq. in.; d = diameter of pipe in ins.; A_s = area of steel reinforcement per lineal foot, expressed in sq. ins.; f = permissible stress in steel in lbs. per sq. in.; then assuming that the concrete is to take no part of the tensile stresses, we have,

$$A_s = \frac{pd \times 12}{2 \times f}$$

The longitudinal reinforcement is taken with an area per lineal foot of circumference equal to half the area per lineal foot of pipe as determined above.

Spacing of Pipe Supports.

Steel and concrete pipes are not always built resting on the ground, and very often they have to be carried on concrete or masonry piers resting preferably on rock, or having a bearing surface in proportion to the load carried and to the resistance of the foundations.

The spacing of the piers is determined by considering the pipe as a circular beam resting freely on supports and carrying a uniformly distributed load consisting of the weight of the pipe plus the weight of the water. This load determines a longitudinal tension in the lower side of the pipe, and this stress added to the longitudinal tension of the pressure of water caused by the static head, plus the extra head due to water hammer, should not be allowed to exceed f, the safe tensile strength of the material.

The stress, f_b, in lbs. per sq. in., due to bending is, $f_b = \frac{3}{2} \cdot \frac{WL^2}{S}$,

where, L = distance between supports in ft.; W = total load in lbs. per ft. of length; S = section modulus of the pipe in terms of (inches)^3.

The longitudinal fibre stress, f_t, due to water pressure is, $f_t = \frac{pd}{4t}$.

where, p = pressure in lbs. per sq. in.; d = nominal inside diameter of pipe in ins.; t = thickness of pipe in ins.

This formula applies to steel pipe, but can be utilised in the calculation of a reinforced concrete pipe, by reducing the total area of steel reinforcement per foot length to a continuous pipe of uniform thickness. This method, however, would not take into account the resistance in compression of the concrete on the upper side of the pipe.

Adding together the stress due to bending f_b and the stress due to water pressure f_t we have a total stress,

$$= \frac{3}{2} \frac{WL^2}{S} + \frac{pd}{4t};$$

from this we can get the distance between the supports

$$L = \sqrt{\frac{2}{3} \frac{S}{W} \left(f - \frac{pd}{4t}\right)}.$$

The permissible value of f, in a steel riveted pipe, can be taken from 8,000 to 11,000 lbs. per sq. in. according to the efficiency of the joint; in a reinforced concrete pipe 16,000 lbs. per sq. in. could safely be assumed.

DESIGN OF WOOD STAVE PIPES.

In Canada and in the United States extensive use is made of pipes composed of wooden staves held in place by steel wires wound continuously around the pipe, or by separate steel bands or rods connected by malleable iron shoes.

The steel is designed to take care of all the stresses and the wood forms only the shell. These pipes are very cheap, and have a greater carrying capacity than either steel or reinforced concrete pipes; they have been used for heads as high as 200 ft.

The following information has been compiled from the writings of Robert E. Horton and of Andrew Swickard.

The staves are made of California redwood, Douglas fir, and Canadian red pine. Practice varies somewhat as to the thickness of staves to be used under given pressure for a pipe of given diameter. The following considerations should govern the thickness of staves to be adopted in a given case.

1. The thicker the staves the greater stability of pipe against deformation by its own weight, and accordingly the staves must be thicker for large than for small diameter of pipes.

2. The staves should be thick enough to prevent material loss by percolation, and accordingly should be thicker for heavy than for lighter pressures.

3. The staves should not be so thick as to prevent their being thoroughly saturated. This and considerations of economy require the use of as thin a stave as is compatible with the first two conditions named.

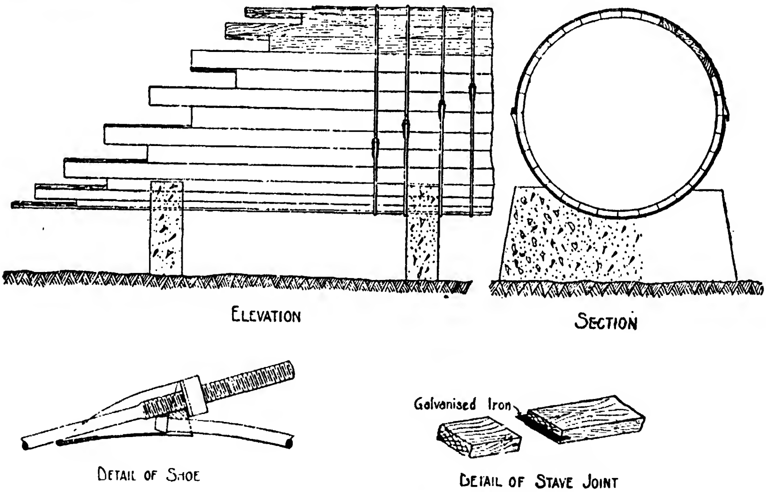


FIG. 2.—Details of Wood Stave Pipe.

The following rule serves as a general guide for determining the approximate thickness of staves to be used in any case; the actual thickness of staves may be made equal to the nearest ⅛ in. or ¼ in. greater thickness than shown by the rule

$$t = 1 \text{ in.} + (\text{Head in ft.} + \text{Diameter of pipe in ins.}) \div 100$$

The minimum radius of bends which may be made with continuous pipe increases with the diameter of the pipe and with the thickness of the staves. An approximate rule is

$$\text{Minimum radius, in ft.} = 4 \text{ to } 5 \times (d + 4t^2)$$

where, d = inside diameter of pipe, in ins.; t = thickness of staves, in ins.

Wood stave pipes should be supported through an arc extending about 55° each side of the vertical axis, in order to prevent deformation of the pipe by its weight and that of the water it contains.

Stave pipes (fig. 2) are generally carried on wooden or on concrete cradles, concrete being preferable for large pipes. Joseph P. Frizell says that wooden cradles are best made of 6 in. × 8 in. or 8 in. × 8 in. timbers, spiked or drift-bolted together, of a length somewhat greater than the diameter of the pipe, and at distance apart about equal to their length. Robert E. Horton is of the opinion that the spacing of the supports is about the same for wood stave and steel pipes.

The abutting end joints of the staves are connected together by iron plates inserted in a slot sawn in for that purpose. This iron tongue should be about $\frac{3}{32}$ in. wider than twice the depth of the slot and about $\frac{1}{8}$ in. longer than the slot, allowing a projection of $\frac{1}{16}$ in. into the adjoining staves.

Creosoted wood-stave pipes put in place twenty-eight years ago were said, at the recent convention of the American Wood Preservers' Association, to be still in perfect condition. The best method of treatment for the staves was described as being the empty-cell process with 50 lbs. initial air pressure, following the creosote pressure period with several minutes' reheating at about 25° F. in excess of any previous temperature, then a vacuum of not less than 20 ins. maintained for from 30 mins. to 1 hour. By this process a satisfactory creosote penetration is obtained, and the staves come out thoroughly clean, and so dry as scarcely to soil the hand.

Area of Bands and their Spacing.

The two principal considerations governing the size and spacing of bands are as follows :—

(1) The area of section of the band per ft. length of pipe must be such as to safely withstand the water pressure plus the stress in the band due to swelling of wood when saturated.

(2) The size and spacing of bands must be such as not to produce crushing of the wood.

Taking into account both the stress due to water pressure and that due to swelling of the staves, the following formula has been derived for determining the necessary area of cross-section of metal in the bands per in. length of pipe.

$$a = \frac{pd + 200t}{2f}$$

where, a = required area of band section per in. length of pipe ; p = pressure in lbs. per sq. in.; d = inside diameter of pipe in ins.; t = thickness of stave in ins.; f = permissible unit stress in lbs. per sq. in., taken generally as 16,000 lbs.

In order to avoid dangerous crushing of the wood, the bearing pressure between bands and staves should not exceed about 800 lbs. per sq. in. of contact between the band and stave. For round bands the width of contact between the band and stave should be about equal to the radius of the band.

From these data the following formula is obtained for determining the maximum permissible spacing S in ins. between bands of given size under a given pressure.

$$S = \frac{400 \times \text{diameter of round bands in ins.}}{\text{Pressure in lbs. per sq. in.}}$$

Shoes or Band Couplings.

The band couplings should be made of malleable iron. The weight of these shoes will vary with the size of the band and the diameter of the pipe, but the following table gives average weights for various sizes of bands, which are accurate enough for estimating purposes :

Diameter of Band, ins.	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
Weight of Shoe, lbs.	0·35	0·40	0·65	1·00	1·75	2·75	4·00	6·00

Length of Bands.

Before the length of a band can be determined it is necessary to know how much lap-over of the rod there is in the shoe. This differs in shoes for different sizes of pipe, increasing with the increase in size of rod. However, this quantity will average about as follows for various sizes of bands :

Diameter of Band, ins.	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
Lap-over of Band in Shoe, ins.	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$

The end of the rods should be upset and threaded over a length of not less than $4\frac{1}{2}$ ins. for $\frac{3}{8}$ in., 5 ins. for $\frac{7}{16}$ in. and 6 ins. for other sizes. There are three types of head in use—button. square, and tee heads. The tee head has the advantage of keeping both ends of the rod closer to the staves, and shortens the space over which the rod is raised from the staves ; it also permits the use of a shorter shoe. The length of rods under the head should be such that, after the pipe is finally tightened, at least $1\frac{1}{2}$ in. of thread will remain ahead of the nut. The length of the rod, excluding the head, is determined in the following manner, where the rod is one part and 6 ins. of thread is being used:

$$\text{Total length} = d + 2t + r)\pi + k + 4\tfrac{1}{2} \text{ ins.}$$

where, d = nominal inside diameter of pipe in ins.; t = thickness of staves in ins.; r = diameter of rod in ins.; k = lap of rod in shoe in ins.

If the rod is in two parts, the following formula applies:

$$\text{Total length} = (d + 2t + r)\,\pi + 2k + 9 \text{ ins.}$$

From 68 tests on wood-stave pipe lines with diameters from 4 to 162 ins., and velocities ranging from 1 ft. to 8 ft. per sec., it was found that with diameters from 4 to 24 ins. the average value of n in Kutter's formula was 0·0109, and with diameters from 31 to 162 ins. the average value of n was 0·0124.

(*Mallett: Flow of Water in Pipes, Vol. CCVIII (April* 1921), *Proceedings Inst. C.E.*)

Unbalanced Pressures on Bends, etc.

Where any considerable pressure has to be sustained by pipes of large size the unbalanced pressure on bends, etc., must be resisted, horizontally by strutting or tying, vertically by strutting or anchoring. The unbalanced pressure on any bend may be found thus:—

Unbalanced pressure on back of bend = the sum of the areas of the ellipses shown in the elevation × the pressure per unit of area, or A × B × 0·7854 × 2P = unbalanced pressure, fig. 3.

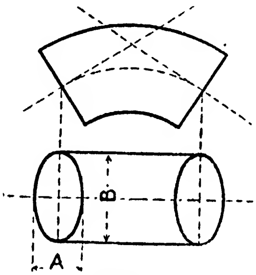


Fig. 3.

Outardes Falls Power Development, Ontario.—The works include a wood-stave pipe line with an inside diameter of 17 ft. 6 in. The staves are of Douglas Fir, $4\frac{11}{16}$ in. thick, with double-tenon butt joints. The bands are 1 in. in diameter, of medium-grade structural steel, with malleable cast iron shoes. (See *The Engineer*, March 25, 1938, for description and illustrations.)

Pressure Pipe Lines.

The Lochaber Plant.—In the second stage, for the utilisation of waters brought into the system from Loch Laggan and the upper Spey, a three-pipe line was constructed. The fall is about 800 ft. The pipes are 2,900 ft. long, and make three changes of inclination, requiring, at the change points, massive concrete thrust blocks. The internal diameter of 78 in., in the upper sections, is reduced towards the bottom end to allow of the increased wall thickness, to a maximum of $1\frac{3}{16}$ in. The pipes, of the spigot and socket type, were shop-fabricated in lengths of about 30 ft., formed of a single plate with the longitudinal seam forge lap welded. They were joined by electric fillet welding of the inner and outer joints.

The Etzel Plant.—The power station utilises the fall of the River Lihl, which discharges into the Lake of Zurich. It has an output of about 160 million kW.h. per annum. The fall is about 910 ft. The pipe line comprises two pipes of diameters ranging from 82·7 ins. at the surge tank to 70·9 ins. at the power station. Each pipe was formed from a steel sheet bent to form a helical joint, which was electrically welded. After welding, the pipes were heated to a temperature of 600° C.

Outardes Falls, Ontario.—In continuation of a wood stave pipe line of 17 ft. 6 in. inside diameter there is a steel pipe line, 18 ft. in diameter, stiffened at $22\frac{1}{2}$ ft. intervals by ring-shaped steel plate girders. The head is 184 ft.

Brooklyn, New Zealand.—The pipe line is 72 chains in length. The pipes are welded and are of copper-bearing steel with flexible joints. They are 14 in. in diameter from chain 0 to chains 51, and 12 in. in diameter thence to the power house. The thickness increases from $\frac{1}{4}$ in. to $\frac{5}{16}$ in. The fall is 920 ft. in 42 chains.

The Roaring Meg Station, New Zealand.—Head, 987 ft. The all-welded concrete-lined steel pipe, $1\frac{1}{2}$ ml. long, tapers from 30 in. to 26 in. and then to 22 in. diameter.

Low-Head Installations.

Pisançon, R. Isère.—Range of head, 41·6 ft. to 8·5 ft., the least head at which the turbines will work. Four 14,000 h.p. turbines; two Francis type, 16 ft. diameter; two Kaplan, 15·5 ft. diameter. Efficiency of Kaplan turbines, 87 per cent. with *h*, 41·6 ft.; 82 per cent. with *h*, 21·3 ft.; 62·5 per cent. with *h*, 8·5 ft. Alternators each 12,500 kVA, at 10,000 volts; rotors 24 ft. 6 in. in diameter; weight of a unit, turbine and alternator rotor, 45 tons.

Upper Gangès Canal.—Supplying current to the grid. In a length of 158 ml. of the canal, the stations: Ranipore, minimum head in feet (*h*), 12; cold weather discharge in cusecs (*q*), 5,000; capacity in kilowatts (*k*), 3,934. Bahadurabad and Salempore, *h*, 15·5; *q*, 5,000; *k*, 5,082. Pathri, *h*, 10; *q*, 5,000; *k*, 3,280. Asafnagar: *h*, 10; *q*, 4,000; *k*, 2,623. Mahmudpur: *h*, 15·5; *q*, 2,000; *k*, 4,066. Nirgajni, *h*, 15·5; *q*, 4,000; *k*, 4,066. Chitaura: *h*, 15·5; *q*, 3,000; *k*, 3,049. Salawa: *h*, 15·5; *q*, 3,000; *k*, 3,049; Bhola: *h*, 12·0; *q*, 3,000; *k*, 2,700. Palra: *h*, 9·0; *q*, 1,200; *k*, 708. Sumera: *h*, 15·5; *q*, 1,200; *h*, 1,219. Total, *k*, 33,776.

Chats Falls, Ottawa River.—There are eight vertical propellor-type turbines rated at 28,000 h.p. under 35 ft. head and at 125 r.p.m. Each turbine is controlled by a Morris-Pelton governor. The eight generators are rated at 23,500 kVA., 13·2 kV., 85 per cent. power factor.

Features of Recent Plants.

Malakand.—The fall below the Malakand Tunnel provides the power for three generating sets each with an output of 3,200 kW. at 11 kV. at 600 r.p.m. The single spiral reaction turbines are rated each at 4,560 b.h.p. under a head of 240 ft. The average span of the transmission lines is 750 ft. Opened in April 1938, the plant will serve a wide area in the North-West Province. (Described in *Indian and Eastern Engineer*, July 1938.)

Galloway Power System.—At one station in this system: head, 365 ft.; two 13,000 kW. vertical Francis turbines. At another there are three 11,000 kW. turbines of the same type.

Grampian Power System.—The Tummel power station, in this system, is equipped with two horizontal Francis type turbines, each of 2,400 b.h.p. Current is generated at 11,000 volts, 50 cycles, 3-phase.

High Falls, Lièvre River, Quebec.—Three steel-plate penstocks 14 ft. in dia. The first three turbines to be installed are of the vertical axis type, rated at 30,000 h.p. under a head of 180 ft. at 180 r.p.m. Three vertical 3-phase, 60-cycle alternators, direct-coupled, are rated at 25,000 kVA. at 85 per cent. power factor, generating current at 13,200 volts.

Roaring Meg Stream, New Zealand.—Head, 987 ft. Two double-jet Boving-Pelton wheels with deflector control drive two 800 kW. Metrovick alternators, generating current at 6,600 volts.

Pangani Falls, Tanganyika.—Head, 620 ft. Two G.E.C. alternators, direct-coupled to the turbines, and generating 3,125 kVA. at 6,600 volts, 750 r.p.m. Power factor 80 per cent.

Pykara River.—Out of a total fall of 4,000 ft., a head of 3,080 ft. is utilised at one station, to develop a maximum of 22,000 h.p. The penstock consists of two pipes, each in three sections of 27 in., 24 in., and 21 in. diameter, and is 9,100 ft. long. The power-house plant consists of three single-jet horizontal impulse wheels rated at 10,900 h.p. each, coupled to 7,810 kVA. 3-phase, 50-cycle alternators, generating at 11,000 volts. (See paper No. 4976, *Inst. C.E.*, by M. G. Platts.) A major extension, 1939, provides two Pelton-wheel units served by a penstock pipe ranging from 42 in. to 37½ in. diameter. The wheels are of 16,000 h.p. each, at 600 r.p.m., driving 12,500 kVA. alternators generating at 11,000 volts.

Papanasam.—For the development of about 27,000 h.p. in the Tinnevelly district, to tie in with the Pykara system, south of Madura. Begun in 1938. The works provided for include: a masonry gravity dam 170 ft. high, impounding 5,500 million cub. ft. of water; a diversion weir, 6 miles downstream, 35 ft. high and 1,350 ft. long, designed to pass a flood of 120,000 cusecs; two 9 ft. low-pressure pipes, 3,500 ft. long; four 69 in. pipes (head 330 ft.); one vertical Francis turbine of 8,150 h.p., driving a 7,250 kVA, 11,000-volt alternator.

Boulder Dam, Colorado River.—There are fifteen 82,500 kVA. water power and generating units at this dam; where the largest turbine is one of 115,000 h.p. of the Francis type; head 590 ft.

River Isarco, Italy.—Carlo Cigogna station. Each of the five large units has its own pipe line of welded steel plate with solid, forged reinforcing rings in the lower sections. Diameter diminishing from 9·2 ft. to 6·2 ft. A sixth pipe serves three Pelton wheel units. The five 45,000 h.p. vertical-shaft Francis turbines drive alternators generating each 36,000 kVA. at 10,000 volts. The Pelton wheels generate current at 16⅔ periods for traction on the Brenner railways. They discharge 335 cusecs at 250 r.p.m.

Recent Plant: United States.—Impulse turbines up to 70,000 h.p. for heads up to 3,000 ft. Reaction Francis turbines for heads up to 850 ft. The largest propeller type turbine (about 1941), 66,000 h.p. For turbines under heads of 100 to 200 ft. the two-bearings unit with umbrella type generator, with an overall efficiency from headwater to tailwater such as 92·6 per cent., is favoured

Hyderabad State.—Available water power, kilowatts, of the rivers: Godaveri, 65,000; Lower Kistna, 50,000; Tungabadra, 40,000; Upper Kistna, 30,000; Lower Manjora, 3,000; Kaddam, 4,000; Parna, 3,500; Painganga, 3,500. Heads from 100 to 300 ft.

British Installations.—Fully described in readily available literature; journals, 'Proceedings,' books, etc.

Recent Literature.

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'Recent Practice in Hydro-Electric Power Development.' By B. Hellstrom. Lecture delivered at the University of London, November 1935, and published in *The Engineer*, issues of February, March and April 1936.

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'The Galloway Water Power Scheme.' *The Engineer*, September 11, 18 and 25, and October 2, 1936. Also *Civil Engineering (London)*, March 1936.

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The Engineer:—

Lièvre River, Quebec: August 12, 1932. *Chats Falls:* September 23, 1932. *Outardes Falls:* Paper by H. G. Acres; March 25, 1938, *et seq.* *Waitaki, New Zealand:* August 24, 1934. *Roaring Meg, New Zealand:* July 17, 1936. *Mandi, India:* March 3, 1932. *Cardano, Italy:* March 29 and April 5, 1935. *Galloway:* September 11, 18, 25 and October 2, 1936. *Lochaber:* May 8, 15, 22, 29 and June 5, 1936. *Grampians:* January 4, 1935. *Malakand:* June 17, 1938.

Civil Engineering (London):—

Galloway: March 1936. *Lochaber:* March 1934, January and February 1937 and (pipe lines) November 1938. *Rio Negro, Uruguay:* article by W. Borkenstein; December 1937. *Pangani, Tanganyika:* July 1936. *Etzel, Switzerland:* February 1938. *Mar ges, R. Dordogne:* September 1934. *River Vatcha, Bulgaria:* October 1934.

Waikato River: New Zealand.—A power station at Karapiro has an output of 90,000 kW. Completed in March 1947, the works include a dam 1,400 ft. long, the building of which took six years. It was necessary to carry the foundations to 43 ft. below sea level. The dam is arched in plan and is of a complex design which gives it somewhat the character of a multiple arch dam. First functioning in 1943, a large horse shoe tunnel was constructed, to take the flow of the river during the building of the dam above the river bed. The closure of the tunnel is effected by lowering an iron gate weighing 50 tons. The flow of the river was such that the reservoir filled in four days, although the gate had been raised for a time in order to supply water to Hamilton and other towns.

SECTION XVIII

PART VI

(See also Parts I, II, and III.)

WATER SUPPLY.

STORAGE RESERVOIRS.

The capacity of storage reservoirs is arrived at from the formula

$$C = \frac{500}{\sqrt[3]{R}},$$

where,

C = number of days' storage, R = available rain in inches, as explained under heading 'Gathering Grounds.'

Messrs. W. J. E. Binnie and H. Lapworth suggested the following formula for calculating storage:—

$$s = 15\frac{(n - d)^2}{r^{1 \cdot 85}},$$

where,

s = inches of storage required to supply (constantly from a stream) n inches per annum; d = the extreme minimum dry-weather flow reduced to inches per annum; r = mean run-off, also in inches per annum.

The storage required in gallons is equal to $s \times$ area of gathering-ground in acres $\times$ 22,610.

The formula is empirical, and was arrived at by a study of the data of seven areas of supply.

GATHERING GROUNDS

In arriving at the water available from an area the average rainfall on the area must be found from rainfall records.

For a study of run-off under various climatic conditions, see 'Rainfall' (Part I).

On the moorland gathering-grounds supplying Sheffield, rainfall averages 40 ins. a year, two-thirds of which is reckoned on as available; one-third goes as compensation to millowners, etc., and one-third to inhabitants. Every 1,000 acres is said to give an average of 1,250,000 gallons per diem. On the Leeds moorland gathering-grounds $\frac{1}{4}$ cubic feet per second, or 134,600 gallons per diem per 1,000 acres, is the usual minimum dry-weather discharge; the formation is millstone grit, rainfall 26 ins. to 45 ins., of which 14 ins. is allowed for evaporation and 6 ins. as compensation to millowners.

Water-Bearing Formations.

The best water-bearing formation in England is the Lincolnshire (Oolitic) limestone, which contains many spouting copious wells; notably one at Sleaford, only 6 inches diameter, which overflows at 30,000 gallons per hour 3 feet above surface. The chalk is very free bearing in many parts, but capricious and occasionally nearly dry. The Keuper marl yields good brewing water at Newark, Burton, Leicester, etc., but not very copiously, and apt to be very small in amount and capricious in character. The Keuper sandstone is a free yielding stratum, and gives good soft water. The Permian sandstone is a fair and at times a copious water-bearing rock and so also is the magnesian limestone.

The Lower Greensand sometimes yields copiously, and, as a rule, the more copious the supply, the better the quality. It is at times ferruginous. The Lias clay is apt to produce saline water; so also is the London clay. The Upper Tertiary beds of the London basin below the London clay often yield fair but slightly alkaline supplies.

The carboniferous rocks vary greatly, and often give copious supplies, and but few rocks are quite dry. As a rule, the more copious yields are all within 200 feet of the surface. Deeper bores yield less and less. Water has been obtained from the Lower Greensand in Windsor Forest at 1,243 feet below surface, the quantity being small, the quality good, and the water rising to about 230 feet above sea level, or overflowing the surface nearly 8 feet. *(W. H. Booth.)*

FILTRATION.

(Based on a contribution by H. F. Rutter, M.I.C.E.)

Sedimentation.—For most waters, except those from deep bore-wells, sedimentation precedes filtration. Trouble arises if water of turbidity exceeding 100 parts per million be taken directly to filters. Coagulants are often used, to promote sedimentation of colloids, the choice of the coagulant depending upon the chemical constitution of the colloid.

Filters.

The area of sand surface needed per million gallons per diem varies considerably, according to the character of the water and the fineness of the filtering medium. Yields of from one million to two million gallons per acre are common values. Modern filters have four feet of water over the sand, two feet nine inches of sand supported by eight inches of graduated gravel, or a greater depth, including a deep layer of shingle at the bottom. Care should be taken that the uppermost layer of shingle should consist of stones less than half the size of a pea, so as to avoid the danger of the sand being drawn through the shingle. The means of collecting the water should cover the whole floor of the filter and should not consist of transverse pipes placed at intervals. A satisfactory and inexpensive method is to cover the floor with transverse rows of bricks laid dry end to end with a space of 4½ inches between each two rows. These are then completely covered with bricks laid dry at right angles to the rows. A series of 4½ in. by 3 in. channels are thus formed, which lead the filtrate to the central channel made below the floor.

A Deacon meter on the outlet pipe forms a useful means of checking and recording the rate of flow of the filter. Float gauges should be provided to show the relative levels of the water over the sand and in the outlet pipe. The difference between these levels indicates the head under which the filter is working. This head may vary from one inch when the filter is started slowly after cleaning, up to 2 or 3 feet.

Means should be provided for drawing the raw water from above the sand to waste, in case it be required to clean a filter immediately which has become clogged. A quarter to half an inch of sand with the matter arrested thereon is removed by flat shovels. This sand can be washed and spread again either at once, or after the depth of the sand has been reduced by five or six successive cleanings. The cleanings may be at intervals of from one week to four months, according to the character of the water, and the amount of algal growth in the water, which latter condition may materially affect the life of the filter. The removal and cleaning of the main body of the sand is needed at very rare intervals. Filters are in use yielding water of perfectly satisfactory bacteriological character in which the main body of sand has not been changed for forty years.

When a filter is to be cleaned the water should not be drawn down in the sand to a level more than 12 inches below the surface of the sand. This leaves the surface quite hard enough to support the weight of the cleaners, and prevents the sand becoming unduly compacted.

All sensible suspended matter is arrested on the surface of the sand of a filter, and the largest elimination of bacterial life takes place at or near the surface. The bacterial contents are, however, still further reduced progressively in the passage of the water through from five to ten inches of the depth of the sand.

Rapid Filters.

Rapid filtration is effected by passing the water under pressure through similar filtering media, the washing of which is effected by reversing the flow. (See Descriptive Section.)

Removal of Humus Colouring Matter.—(F. Sartorius). Employed in Germany. The water is saturated with oxygen and filtered, successively through: (1) 10 cm. of iron shavings; (2) 25 cm. of 'magno-mineral,' a burned dolomite, in grains of 2 mm.; (3) 60 cm. of a mixture of (1) and (2) in proportion 1 : 10; (4) 50 cm. of 'magno-mineral.' Reduction of humus, 60 to 80 per cent.

Standard of Purity of Water.

Total hardness in practice varies from 0 to 30 degrees; 5 degrees is generally considered best. A degree is one grain per gallon.

Water containing more than the following quantities of impurities should be looked upon with suspicion: Chlorine, 1 part per 100,000; Nitrites, 0·2 part per 100,000; Nitrates, 0·3 part per 100,000; Ammonia, 0·1 part per 100,000; Organic Carbon, 2 parts per 100,000; Organic Nitrogen, 3 parts per 100,000; Albumenoid Ammonia, 0·01 part per 100,000.

Bacteriological Examination.—Examination of water in order to ascertain the number of, if any, bacilli of various species in a given volume of water is conducted by bacteriologists. In almost all but the purest spring or well water, *bacillus coli communis* is found. It is not a producer of disease, but is usually evidence of some accompanying danger, roughly proportional to the density of *B. coli*.

Albuminoid Ammonia.—The albuminoid ammonia content of water is, on the whole, the most important index with respect to the fitness of a water for human consumption. Water is regarded as pure if the content is less than 0·002 parts per 100,000, even if free ammonia is present in large proportion. If the content exceed 0·005 parts per 100,000, and the proportions of free ammonia and chlorine are large, a bacteriological examination is necessary.

Free Ammonia.—A water is seldom safe if the quantity of free ammonia per 100,000 parts of water exceeds: in spring waters, 0·01; in upland surface water, 0·008; in water derived from cultivated land, 0·025.

Chlorides.—Except in so far as common salt is naturally present in the water, the significance of chlorides is with regard to the probability of pollution by sewage or excreta.

SIMPLE TESTS OF WATER.

For *Sewage.*—Four drops of a solution of potash (Condy's Fluid) to a small glass of water. The colour turns pale or yellow if decomposed organic matter is present.

Lead.—Six drops of sulphuric acid in small glass of water. White precipitate will be found if lead is present.

Zinc.—Six drops of ferro-cyanide of potassium to small glass of water. Green colour if zinc is present.

Copper.—Eight drops of ammonia in small glass of water. Water turns blue if copper is present.

Hardness of Water.

A water is said to be *hard* when it is difficult to obtain a lather with soap; in washing, soap dissolves in pure (soft) water, but with a hard water it combines with the calcium and magnesium present, and forms an insoluble precipitate. Before a lather can be obtained it is necessary to add sufficient soap to precipitate all the soluble calcium and magnesium salts present in the water. About ¼ oz. of soap is required to remove 1° of hardness from 10 gallons of water.

At the present time the hardness is preferably expressed in terms of the salts actually causing it, and the proportion of each of these in the water.

Temporary hardness is that which is removed by boiling the water. It is due to calcium bicarbonate and/or magnesium bicarbonate, which are decomposed when the water is heated, calcium carbonate and/or magnesium carbonate being precipitated and so rendered inert to soap.

Permanent hardness is that which remains after boiling. It is chiefly due to calcium and/or magnesium sulphate and/or chloride, but may also be caused by any other soluble salts in the water. The calcium and magnesium salts combine with soap and form insoluble compounds. Soluble salts act differently and precipitate the soap as soap, because it is insoluble in a solution of salt. In most permanently hard waters the proportion of soluble salts other than those of calcium and magnesium is so small that it may be ignored, but in softening sea-water or other saline water it must be taken into consideration.

Acid water also decomposes soap and so may be regarded as hard water.

Zero hardness is a term applied to water which is devoid of hardness. Most softened waters retain two or three degrees of hardness and are not softened to zero.

SOFTENING WATER.

Hard water may be softened by (i) rendering the hardness-causing substances insoluble, or (ii) by removing them. The substances causing *temporary hardness* may be rendered insoluble by (*a*) heating the water—usually to about 150° F., but in all cases by boiling it (212° F.). The heating decomposes the bicarbonates present, carbon-dioxide passing off as gas, and the mono-carbonates being left suspended in an insoluble and inert form in the water.

(*b*) Adding some substance which will decompose the bicarbonates without forming a product which can effect soap. Any alkali soluble in water can be used; the commonest being lime, caustic soda, and baryta. Lime is the cheapest of these substances. *Lime* combines with half the carbon dioxide in the bicarbonate and with it forms calcium carbonate which is insoluble and inert to soap.

$$\underset{\text{Soluble bicarbonate.}}{XCO_3.H_2CO_3} + \underset{\text{Lime.}}{CaO} = \underbrace{XCO_3 + CaCO_3}_{\text{Insoluble carbonates.}} + \underset{\text{Water.}}{H_2O}$$

Baryta acts precisely like lime and forms insoluble barium carbonate.

Caustic soda decomposes the bicarbonate, precipitating the insoluble mono-carbonate of lime and/or magnesia, and forming soluble sodium carbonate. The latter is useful if the water is permanently hard, but may be objectionable in water used for industrial or scientific purposes.

Substances causing temporary hardness may be removed from hard water by passing it through a porous mass of zeolite such as 'Permutit.' This is a sodium alumino-silicate which gives up its sodium in exchange for an equivalent amount of lime or magnesia in the water.

$$\underset{\text{Sodium zeolite.}}{Na_2Z} + \underset{\text{Soluble bicarbonate.}}{XCO_3H_2CO_3} = \underset{\text{Altered zeolite.}}{XZ} + \underset{\text{Soluble bicarbonate.}}{2NaHCO_3}$$

The soluble sodium carbonate is useful in removing an equivalent of permanent hardness but may be objectionable in water used for some purposes.

Permanent hardness can be removed from water by:

(1) Distillation, *i.e.*, boiling the water and condensing the steam. This is the only method for fully purifying water rich in soluble sodium or potassium salts (*e.g.* sea water).

(2) Precipitating any soluble calcium or magnesium saltsby the addition of a soluble carbonate, sodium carbonate (soda) being generally used.

$$XSO_4 + Na_2CO_3 = XCO_3 + Na_2SO_4$$
$$XCl_2 + Na_2CO_3 = XCO_3 + 2NaCl$$

Hardness-forming salt. Soda. Insoluble carbonate. Soluble salt.

This method leaves a soluble sodium salt in solution; in most instances it does no harm, but it is objectionable in water used for some purposes. This method softens acid water by neutralising the acid and so rendering it inert to soap.

If caustic soda is used for softening a water having both temporary and permanent hardness, sodium carbonate will be formed first from the action of the 'temporary hardness' and this will be available for removing an equivalent amount of permanent hardness.

$$XCO_3H_2CO_3 + 2NaOH = XCO_3 + Na_2CO_3 + 2H_2O$$

Temporary hardness. Caustic soda. Insoluble carbonate. Sodium carbonate. Water.

$$YSO_4 + Na_2CO_3 = YCO_3 + Na_2SO_4$$

Permanent hardness. Sodium carbonate. Insoluble carbonate. Sodium sulphate.

Hence, for every 100 parts of calcium carbonate (present as bicarbonate) causing temporary hardness, sufficient sodium carbonate will be formed to remove 136 parts of the calcium sulphate causing permanent hardness. For this reason, when a water has both temporary and permanent hardness it is usually cheaper to use caustic soda than lime for removing the temporary hardness. Incidentally caustic soda is much less troublesome to use as it is readily soluble in water. An excess of caustic soda must be avoided as it may cause embrittlement and leakage at the fittings.

(3) Removing the soluble calcium or magnesium salts by passing the water through a porous mass of zeolite such as 'Permutit.'

$$Na_2Z + XSO_4 = XZ + Na_2SO_4$$

Sodium zeolite. Hardness-forming salt. Altered zeolite. Sodium sulphate.

The advantage of using a zeolite for softening water is that it acts automatically so long as any unaltered zeolite remains; and there can be no 'mistake' due to adding too much or too little reagent. On the other hand, when all the zeolite is altered, it has to be re-activated by treatment with a solution of common salt.

The periodic renewal of zeolite is necessary with most waters containing permanent hardness, as the sodium bicarbonate liberated by the zeolite combines with an equivalent amount of calcium and/or magnesium sulphate or chloride, and precipitates the corresponding carbonate in the pores of the zeolite which are gradually filled with it.

Softening by a zeolite also has the advantage of producing a water of zero hardness in many instances where the addition of 'chemicals' leaves about 3° of permanent hardness.

For further information on the Treatment of Feed-water for boilers and the softening of water see Descriptive Section XXVI, Part II.

COST OF WATER SOFTENING.

From an article, *The Engineer*, October 28, 1938, fully describing the water-softening plant at the Newnham pumping station of the Mid-Kent Water Company. The water is pumped from three bore-holes 350 feet deep.

	Parts per 100,000.
Chemical Analysis.—Saline ammonia	0·0005
Albuminoid ammonia	0·0007
Oxygen absorbed from $KMnO_4$ at 26·7° C. in 4 hours	0·004
Total solid residue, dried at 180° C.	32·5
Nitrogen as nitrates	0·35
Nitrogen as nitrites	Nil
Calcium	10·3
Magnesium	0·25
Silica	0·85
Iron and aluminium	Traces
Sodium and potassium	1·6
Sulphates as SO_4	0·5
Chloride	1·65
CO_2, free	2·8
HCO_3 combined	29·9
Hardness: Temporary	24·2
Permanent	3·2
Total	27·4
*p*H value	7·1
Bacteriological Examination.—Number of micro-organisms per cubic centimetre growing on gelatine at 22° C.	14
Number of micro-organisms per cubic centimetre growing on agar at 37° C.	2
Bacillus Coli. Absent from 110 cubic centimetres.	
Streptococci " " " " "	

Cost of the Treatment.—Per 1,000 gallons. Reagents: lime, 0·330*d.*; ferric chloride, 0·09*d.*; chemicals for test, 0·018*d.*; total reagents, 0·438. Labour: unskilled attention 2 hr. per day, 0·08*d.*; skilled attention 1 hr. per day, 0·05*d.*; total labour, 0·130*d.* Total cost of labour and chemicals, 0·568*d.* per 1,000 gallons.

Lime v. Soap.—To remove 1 lb. of calcium hardness requires about 12 lb. of soap, but only 0·6 lb. of 93 per cent. lime if equivalent bicarbonates are present in the water, or 1·08 lb. of 98 per cent. soda-ash if bicarbonates are not present. The cheaper soaps cost at least fifteen times as much per lb. as equivalent lime, and ten times as much as soda.

Therefore, to remove calcium hardness from water with soap costs from 100 to 3,000 times what it costs to remove it with lime or soda, depending upon the relative amounts of the chemicals required. (C. H. Spaulding, Supervising Chemist, Water, Light and Power Department, Springfield, Illinois.)

CHEMICAL PURIFICATION OF WATER.

Reservoirs in which vegetable growth or algæ has become profuse are cured by dragging a bag of sulphate of copper behind a boat to and fro over the area of the reservoir, or by dosing the water as it enters. A very minute fraction of this poisonous salt will effectually destroy all growth, and it is said that the salt cannot be detected in the water after the process is complete. The proportion of the salt is varied according to the species to be destroyed and what life must not be destroyed.

Water Sterilisation.

The four processes in common use are: (1) Simple chlorination; (2) the ammonia-chlorine treatment, or chloramination; (3) super-chlorination followed by dechlorination (*a*) by the addition of a chemical anti-chlor, (*b*) by filtration through activated carbon; (4) ozonisation. The activated carbon process is of importance, apart from its use for dechlorination. (See 'Literature,' Papers (2).)

Chlorination and Chloramination.

Chloramination, the ammonia chloride process, is by some water authorities preferred to simple chlorination, and is applied to all, or nearly all, the waters of the Metropolitan Water Board.

Comparing the two processes—it has been observed by J. Bowman, President, 1941, of the Institution of Water Engineers—the chief objection to chloramination is that the retarding influence of the ammonia may seriously interfere with sterilisation unless long contact is possible.

Adequate contact time is necessary before the chlorinated water can be regarded as safe, and information should be sought as to the conditions under which the process is not suitable, before a plant is installed. Such conditions have been referred to (A. P. I. Cotterell) as 'fairly prevalent.'

Superchlorination.—The process comprising superchlorination and dechlorination presents undoubted advantages, as regards the safety and palatability of the treated water, and the ease and flexibility of working conditions. In many cases it is outstandingly the method of choice. (A. P. I. Cotterell.)

Ozonisation.

There were, in 1941, some 90 ozonisation installations on public water supplies in France, 14 in Italy, 5 in Belgium, 4 in Great Britain and 2 in the United States. At Ashton-under-Lyme, the experimental work with the installation (see 'Literature,' Papers (2)) gave the results: cost per 1,000 gal., 0·269*d.*, including electrical energy (at 0·5*d.* per unit), 0·128*d.* That cost, nearly 22*s.* 5*d.* per million gal., could be reduced to 14*s.* 5*d.*, or nearly 0·17 per 1,000 gal., in cases in which the head necessary for the emulsification of the ozone and water could be obtained without pumping.

American Data.—Whiting, Indiana: Cost of plant to treat 4 million gal. per day, 30,000 dollars; average dosage, 11·7 lb., maximum 43 lb. per day; total cost, 4·44 dollars per million gallons.

The Uses of Activated Carbon.

Granular carbon filters are described in the paper, No. 2, 'Literature.' They serve the two purposes, dechlorination and the removal of taste due to causes other than chlorine. The additional expense of such a provision is small, an area of some 80 sq. ft. and about 23 cu. yd. of the filtering medium sufficing, in one case, to deal with 5 million gal. a day. In another installation, the rate of 12 gal. per sq. ft. per min. was adopted.

At Philadelphia, a repulsive-looking river water was first put through charcoal filters, then through slow sand filters, and subsequently chlorinated, emerging as a clear and safe, if a somewhat 'lifeless,' water. (J. Bowman.)

Greenish reservoir water which passed through pressure filters but retained a fishy smell, was rendered palatable and acceptable by injecting finely powdered activated carbon into the main feeding the filters. (W. J. E. Binnie)

Incrustation and Corrosion of Iron and Steel Pipes.

High water velocity is favourable to corrosion and incrustation. Some waters, as chalk-well waters, will simply cause a clean layer of carbonate of lime inside a pipe. Upland waters, especially if containing peat acid, cause rust. Old red-sandstone water neither rusts nor causes deposit. Greensand water causes rust. Filtered water is better than unfiltered.

Iron Bacteria.—About three weeks after water was turned into the Alameda section of the Hetch Hetchy water supply, a 36-in. wrought iron pipe 22 mls. long, the carrying capacity began to drop off at the rate of 0·2 million gals. per day; the normal capacity being 16 million gals. per day. Within 3 weeks the flow had diminished to 13·6 million gals. per day. The entire inner surface of the pipe was covered with a slimy gelatinous growth, ⅛ in. to ¼ in. thick, containing *Schizomyetes Crenothrix*, which originated in the tunnels supplying the pipeline. The use of 1¼ lb. of chlorine and 6 lb. of ammonia per million gals., for some months, progressively improved conditions, preventing the growth of the bacteria.

Tracing Leakages.

To identify and trace the source of underground water, such as leakages from canals, reservoirs, pipes, etc.:—

Fluorescin.—Mix fluorescin with the supposed source of the leakage. One grain is sufficient to colour 100 tons of water.

Indigo.—One grain of indigo dissolved in sulphuric acid will colour a ton of water.
The coloration is more easily detected by placing the water in a very long test-tube and looking through the latter endways.

Stethoscope.—Proceeding along the line of the main in the night hours of least consumption, the patrol nearly closes the valves at principal branches, each in turn, and listens with an instrument which is essentially a large stethoscope, placing its stem against the spindle of the valve. Flow through the valve is indicated by a sizzling sound. Having found the branch main on which the leak is located, similar trials enable the patrol to find the pipe which is leaking.

Diameter of Pipes.

The diameters of pipes are usually calculated by either Eytelwein's or Kütter's formula.
Eytelwein's formula is

$$V = \sqrt{(11704\,RS) + \cdot01698} - \cdot1303$$

where,
V = velocity in feet per sec.; R = Hydraulic mean radius in feet = Area ÷ wet perimeter = D ÷ for circular section of pipe, where D = diameter of pipe in feet; S = sine of inclination = $h \div l$ where h = head in feet; l = length in feet.
An adaptation of this formula which gives good average results is

$$Q = 4\cdot7\,D^2\sqrt{\frac{Dh}{l}}$$

where,
Q = discharge in cub. ft. per min., and D is in inches.
This formula gives rather high results in the case of small pipes with flat gradients, and somewhat low results for large pipes with steep gradients.

Kütter's formula is given in Part II, p. 676.

New and Old Pipes.—As a check on other formulas and on values in tables, we may employ the formula:

$$v = \frac{C}{2}\sqrt{ds}$$

in which v = velocity in feet per second; d = diameter of the pipe in feet; s = slope. The values of c are usually taken as between those for new and for old pipes in the following series, the highest values being for new or perfectly clean pipes, and the lowest values being seldom required in water-supply systems in which the pipes are cleaned or are renewed before they are very old.

½ in.: $d = \frac{1}{24}$; C, 65–46.
1 in.: $d = \frac{1}{12}$; C, 80–56.
3 ins.: $d = \frac{1}{4}$; C, 98–70.
6 ins.: $d = \frac{1}{2}$; C, 105–74.
12 ins.: $d = 1$; C, 109–77.
24 ins.: $d = 2$; C, 111–78.
36 ins.: $d = 3$; C, 112–79.

The highest values thus found correspond with those in the following table, which are for new pipes.

TABLE SHOWING VELOCITY IN FEET PER SECOND, AND SUPPLY IN GALLONS PER MINUTE, FOR LONG PIPES FLOWING FULL.

(See note on the last page of this table.)

Diameter of Pipe in Inches.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Diameter of Pipe in Inches.
	Head of Water divided by Length of Pipe.										
	$\frac{1}{1000}$		$\frac{2}{1000}$		$\frac{3}{1000}$		$\frac{4}{1000}$		$\frac{5}{1000}$		
$\frac{3}{8}$	·173	·05	·278	·08	·363	·10	·436	·13	·502	·14	$\frac{3}{8}$
$\frac{1}{2}$	·212	·11	·336	·17	·436	·22	·522	·27	·600	·31	$\frac{1}{2}$
$\frac{3}{4}$	·278	·32	·436	·50	·562	·64	·670	·77	·770	·88	$\frac{3}{4}$
1	·336	·69	·522	1·07	·670	1·37	·798	1·63	·911	1·86	1
$1\frac{1}{4}$	·388	1·24	·600	1·91	·770	2·45	·911	2·90	1·04	3·31	$1\frac{1}{4}$
$1\frac{1}{2}$	·436	2·00	·670	3·08	·856	3·93	1·02	4·67	1·16	5·44	$1\frac{1}{2}$
$1\frac{3}{4}$	·481	3·00	·736	4·60	·938	5·86	1·11	6·94	1·27	7·91	$1\frac{3}{4}$
2	·522	4·26	·798	6·51	1·02	8·29	1·20	9·81	1·37	11·17	2
$2\frac{1}{2}$	·600	7·64	·911	11·62	1·16	14·76	1·37	17·45	1·56	19·85	$2\frac{1}{2}$
3	·670	12·30	1·02	18·66	1·29	23·64	1·52	27·92	1·73	31·73	3
4	·798	26·03	1·20	39·23	1·52	49·63	1·79	58·53	2·04	66·46	4
5	·911	46·18	1·37	69·79	1·73	88·11	2·04	103·84	2·31	117·80	5
6	1·02	74·64	1·52	111·66	1·92	140·83	2·26	165·77	2·56	187·95	6
7	1·11	111·09	1·66	166·07	2·09	209·22	2·46	216·10	2·79	278·90	7
8	1·20	156·92	1·79	231·11	2·26	294·70	2·65	346·15	3·01	392·46	8
9	1·29	212·71	1·92	316·87	2·41	398·58	2·83	468·35	3·21	530·36	9
10	1·37	279·16	2·04	415·34	2·56	522·08	3·01	613·22	3·40	694·24	10
11	1·45	356·95	2·15	530·42	2·70	666·39	3·17	782·42	3·59	885·56	11
12	1·52	446·66	2·26	663·07	2·83	832·63	3·33	977·29	3·76	1105·8	12
15	1·73	793·27	2·56	1174·7	3·21	1473·2	3·76	1727·8	4·26	1956·7	15
18	1·92	1267·5	2·83	1873·4	3·55	2347·2	4·16	2751·4	4·71	3110·0	18
24	2·26	2652·2	3·33	3909·2	4·16	4891·4	4·88	5727·8	5·51	6470·3	24
30	2·56	4698·8	3·76	6911·3	4·71	8638·9	5·51	10109·7	6·22	11415·0	30

Diameter of Pipe in Inches.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Diameter of Pipe in Inches.
	Head of Water divided by Length of Pipe.										
	$\frac{6}{1000}$		$\frac{7}{1000}$		$\frac{8}{1000}$		$\frac{9}{1000}$		$\frac{1}{100}$		
$\frac{3}{8}$	·562	·16	·618	·18	·670	·19	·720	·21	·770	·22	$\frac{3}{8}$
$\frac{1}{2}$	·670	·34	·736	·38	·798	·41	·856	·44	·911	·46	$\frac{1}{2}$
$\frac{3}{4}$	·856	·98	·938	1·08	1·02	1·17	1·09	1·24	1·16	1·33	$\frac{3}{4}$
1	1·02	2·08	1·11	2·27	1·20	2·45	1·29	2·63	1·37	2·79	1
$1\frac{1}{4}$	1·16	3·69	1·27	4·04	1·37	4·36	1·47	4·67	1·56	4·96	$1\frac{1}{4}$
$1\frac{1}{2}$	1·29	5·91	1·41	6·46	1·52	6·98	1·63	7·47	1·73	7·93	$1\frac{1}{2}$
$1\frac{3}{4}$	1·41	8·79	1·55	9·68	1·66	10·38	1·78	11·10	1·89	11·79	$1\frac{3}{4}$
2	1·52	12·41	1·66	13·56	1·79	14·63	1·92	15·65	2·04	16·61	2
$2\frac{1}{2}$	1·73	22·04	1·82	23·30	2·04	25·96	2·18	27·75	2·31	29·45	$2\frac{1}{2}$
3	1·92	35·21	2·09	38·43	2·26	41·41	2·41	44·29	2·56	46·99	3
4	2·26	73·68	2·46	80·36	2·65	86·61	2·83	92·51	3·01	98·60	4
5	2·56	130·52	2·79	142·30	3·01	153·31	3·21	163·69	3·40	173·70	5
6	2·83	208·16	3·09	226·84	3·33	244·32	3·55	260·81	3·76	277·10	6
7	3·09	308·76	3·37	335·60	3·62	362·21	3·87	386·57	4·09	408·45	7
8	3·33	434·35	3·62	473·09	3·90	509·38	4·16	543·49	4·41	575·83	8
9	3·55	586·82	3·87	639·02	4·16	687·85	4·44	733·83	4·71	777·50	9
10	3·76	767·92	4·09	833·58	4·41	899·75	4·71	959·88	4·99	1016·9	10
11	3·97	979·39	4·32	1066·1	4·65	1147·2	4·96	1223·7	5·26	1298·4	11
12	4·16	1222·9	4·53	1331·3	4·88	1432·0	5·18	1522·7	5·51	1617·6	12
15	4·71	2159·7	5·12	2349·8	5·51	2527·4	5·87	2694·9	6·22	2853·8	15
18	5·18	3426·1	5·65	3737·0	6·08	4019·3	6·48	4284·7	6·86	4536·5	18
24	6·08	7115·4	6·61	7769·1	7·11	8351·6	7·58	8900·4	8·02	9421·1	24
30	6·86	12601·3	7·50	13772·4	8·02	14720·4	8·54	15684·5	9·04	16599·0	30

TABLE SHOWING VELOCITY IN FEET PER SECOND, AND SUPPLY IN GALLONS PER MINUTE FOR LONG PIPES FLOWING FULL (*continued*).

Diameter of Pipe in Inches.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Diameter of Pipe in Inches.
	Head of Water divided by Length of Pipe.										
	$\frac{2}{100}$		$\frac{3}{100}$		$\frac{4}{100}$		$\frac{5}{100}$		$\frac{6}{100}$		
$\frac{3}{8}$	1·16	·33	1·47	·12	1·73	·50	1·96	·56	2·18	·62	$\frac{3}{8}$
$\frac{1}{2}$	1·37	·70	1·73	·88	2·01	1·04	2·31	1·18	2·56	1·31	$\frac{1}{2}$
$\frac{3}{4}$	1·73	1·98	2·18	2·50	2·56	2·94	2·90	3·33	3·21	3·68	$\frac{3}{4}$
1	2·04	4·15	2·56	5·22	3·01	6·13	3·10	6·94	3·76	7·68	1
1¼	2·31	7·36	2·90	9·24	3·40	10·85	3·85	12·27	4·26	13·59	1¼
1½	2·56	11·75	3·21	14·73	3·76	17·28	4·26	19·57	4·71	21·60	1½
1¾	2·79	17·43	3·50	21·84	4·09	25·53	4·63	28·95	5·12	31·98	1¾
2	3·01	24·53	3·76	30·72	4·41	35·99	4·99	40·67	5·51	44·93	2
2½	3·40	43·39	4·26	54·35	4·99	63·55	5·63	71·79	6·22	79·27	2½
3	3·76	69·11	4·71	86·39	5·51	101·10	6·22	114·15	6·86	126·01	3
4	4·41	144·20	5·51	179·73	6·44	210·17	7·27	237·17	8·02	261·70	4
5	4·99	254·50	6·22	317·08	7·27	370·57	8·20	418·02	9·04	461·08	5
6	5·51	404·30	6·86	504·05	8·02	588·82	9·04	663·96	9·97	732·18	6
7	5·99	598·62	7·50	749·83	8·71	870·78	9·82	981·68	10·83	1082·4	7
8	6·44	840·68	8·02	1046·8	9·36	1222·0	10·55	1377·3	11·63	1518·3	8
9	6·86	1134·1	8·54	1411·6	9·97	1647·4	11·21	1856·4	12·38	2046·2	9
10	7·27	1482·3	9·04	1844·3	10·55	2152·0	11·89	2424·5	13·10	2665·9	10
11	7·65	1888·3	9·52	2348·8	11·10	2740·1	12·49	3082·9	13·78	3401·4	11
12	8·02	2355·3	9·97	2928·7	11·63	3416·1	13·10	3838·9	14·43	4239·6	12
15	9·04	4149·7	11·24	5157·0	13·10	5998·2	14·75	6769·5	16·25	7457·0	15
18	10·07	6653·8	12·38	8181·8	14·43	9539·1	16·25	10738·1	17·89	11826·3	18
24	11·63	13664·4	14·43	16958·3	16·81	19754·1	18·92	22228·8	20·83	24474·3	24
30	13·10	23993·0	16·25	29828·1	18·92	34732·5	21·28	39073·2	23·43	43011·1	30
	Head of Water divided by Length of Pipe.										
	$\frac{7}{100}$		$\frac{8}{100}$		$\frac{9}{100}$		$\frac{1}{10}$		$\frac{2}{10}$		
$\frac{3}{8}$	2·37	·68	2·56	·73	2·74	·79	2·90	·83	4·26	1·22	$\frac{3}{8}$
$\frac{1}{2}$	2·79	1·42	3·01	1·53	3·21	1·64	3·40	1·36	4·99	2·54	$\frac{1}{2}$
$\frac{3}{4}$	3·50	4·01	3·76	4·32	4·01	4·60	4·26	4·89	6 22	7·13	$\frac{3}{4}$
1	4·09	8·34	4·41	9·00	4·71	9·60	4·99	10·17	7·27	14·82	1
1¼	4·63	14·77	4·99	15·89	5·32	16·94	5·63	17·95	8·20	26·13	1¼
1½	5·12	23·50	5·51	25 27	5·87	26·95	6·22	28·54	9·04	41·50	1½
1¾	5·57	34·79	5·99	37·41	6·38	39·89	6·76	42·23	9·82	61·35	1¾
2	5·99	48·87	6·44	52·51	6·86	56·01	7·27	59·29	10·55	86·08	2
2½	6·76	86·18	7·27	92·64	7·74	98·73	8·20	104·50	11·89	151·53	2½
3	7·50	137·72	8·02	147·20	8·54	156·85	9·04	165·99	13·10	239·93	3
4	8·71	284·34	9·36	305·50	9·97	325·41	10·55	344·32	15·26	498·17	4
5	9·82	500·86	10·55	537·99	11·24	572·99	11·89	606·13	17·18	876·11	5
6	10·83	795·20	11·63	854·02	12·38	909·42	13 10	959·72	18·92	1389·3	6
7	11·76	1175·3	12·63	1262·1	13·44	1343·7	14·22	1421·2	20·52	2051·3	7
8	12·63	1648·4	13·56	1765·8	14·43	1884·3	15·26	1992·7	22·02	2874·7	8
9	13·44	2221·3	14·43	2384·8	15·36	2538·6	16·25	2684·5	23·43	3871·0	9
10	14·22	2900·4	15·26	3113·5	16·25	3314·2	17·18	3514·5	24 76	5051·3	10
11	14·96	3691·7	16·06	3962·7	17·09	4217·9	18·07	4459·7	26·04	6425·9	11
12	15·66	4601·1	16·81	4938·6	17·89	5256·2	18·92	5557·2	27·25	8004·7	12
15	17·63	8091·3	18·92	8683·1	20·13	9240·3	21·28	9768·3	30·63	14059·7	15
18	19·41	12830·1	20·83	13766·8	22·16	14648·5	23·42	15484·0	33·70	22273·1	18
24	22·59	26545·4	24·24	28477·6	25·79	30296·3	27·25	32018·8	39·17	46016·9	24
30	25·41	46642·4	27·25	50029·4	[illegible]	53218·5	30·63	56238·9	44·0	80770·7	30

TABLE SHOWING VELOCITY IN FEET PER SECOND, AND SUPPLY IN GALLONS PER MINUTE, FOR LONG PIPES FLOWING FULL (*continued*).

Diameter of Pipe in Inches.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Diameter of Pipe in Inches.
	Head of Water divided by Length of Pipe.										
	3/10		4/10		5/10		6/10		7/10		
⅜	5·32	1·52	6·22	1·78	7·02	2·01	7·74	2·22	8·42	2·41	⅜
½	6·22	3·17	7·27	3·71	8·20	4·18	9·04	4·61	9·82	5·01	½
¾	7·74	8·89	9·04	10·37	10·19	11·69	11·24	12·89	12·20	14·00	¾
1	9·04	18·44	10·55	21·52	11·89	24·25	13·10	26·66	14·22	29·00	1
1¼	10·19	32·48	11·89	37·88	13·39	42·67	14·75	47·01	16·01	51·02	1¼
1½	11·24	51·57	13·10	59·98	14·75	67·70	16·25	74·57	17·63	80·91	1½
1¾	12·20	76·21	14·22	88·82	16·01	99·99	17·63	110·14	19·13	119·48	1¾
2	13·10	106·64	15·26	124·54	17·18	140·18	18·92	154·37	20·52	167·45	2
2½	14·75	188·42	17·18	219·03	19·33	246·46	21·28	271·34	23·08	294·29	2½
3	16·25	298·28	18·92	347·32	21·28	390·73	23·43	430·11	25·41	466·42	3
4	18·92	617·47	22·02	718·66	24·76	808·21	25·25	824·14	29·55	964·31	4
5	21·28	1085·4	24·76	1262·8	27·84	1419·8	30·63	1561·7	33·21	1692·6	5
6	23·43	1720·4	27·25	2001·2	30·63	2248·9	33·70	2474·8	36·53	2682·4	6
7	25·41	2539·4	29·55	2953·2	33·21	3319·2	36·53	3651·1	39·59	3957·0	7
8	27·25	3557·6	31·67	4134·9	35·60	4647·9	39·17	5113·0	42·44	5540·9	8
9	28·99	4789·7	33·70	5568·3	37·87	6256·8	41·65	6881·1	45·13	7456·4	9
10	30·63	6246·9	35·60	7262·3	40·01	8161·0	44·00	8974·3	47·67	9724·4	10
11	32·20	7947·9	37·41	9234·4	42·00	10365·4	46·24	11411·6	50·10	12364·4	11
12	33·70	9892·2	39·17	11504·2	44·00	12923·0	48·38	14209·8	52·40	15391·1	12
15	37·87	17380·0	44·00	20192·7	49·42	22679·2	54·33	24932·9	58·85	27009·8	15
18	41·65	27524·2	48·38	31972·0	54·33	35903·4	59·72	39465·9	64·52	42641·5	18
24	48·38	56839·1	56·18	66005·2	63·07	74102·4	69·32	81442·6	75·07	88204·4	24
30	54·33	99731·6	63·07	115785·0	70·80	129971·0	77·80	142825·0	84·25	154688·	30

Diameter of Pipe in Inches.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Velocity in Feet per Second.	Supply in Gallons per Minute.	Diameter of Pipe in Inches.
	Head of Water divided by Length of Pipe.						
	8/10		9/10		1		
⅜	9·04	2·59	9·63	2·76	10·19	2·92	⅜
½	10·55	5·38	11·24	5·73	11·89	6·06	½
¾	13·10	15·00	13·93	15·98	14·75	16·92	¾
1	15·26	31·14	16·25	33·14	17·18	35·04	1
1¼	17·18	54·76	18·28	58·28	19·33	61·61	1¼
1½	18·92	86·83	20·13	92·40	21·28	97·68	1½
1¾	20·52	128·21	21·84	136·42	23·08	144·20	1¾
2	22·02	179·67	23·43	191·16	24·76	202·05	2
2½	24·76	315·71	26·34	335·86	27·84	354·95	2½
3	27·25	500·29	28·99	532·18	30·63	562·22	3
4	31·67	1033·7	33·70	1099·9	35·60	1162·0	4
5	35·60	1815·6	37·87	1931·1	40·01	2040·2	5
6	39·17	2876·1	41·65	3058·2	44·00	3230·8	6
7	42·44	4242·3	45·13	4510·7	47·67	4764·9	7
8	45·50	5910·0	48·38	6315·4	51·10	6671·1	8
9	48·38	7993·0	51·44	8499·2	54·33	8975·8	9
10	51·10	10423·5	54·33	11081·3	57·38	11704·2	10
11	53·70	13252·8	57·08	14088·5	60·29	14880·0	11
12	56·18	16501·3	59·72	17540·4	63·07	18525·6	12
15	63·07	28946·2	67·04	30767·7	70·80	32492·7	15
18	69·32	45811·5	73·67	48690·4	77·80	51116·9	18
24	80·44	94508·9	85·49	100438·1	90·26	106052·8	24
30	90·26	165708·0	95·92	176090·0	101·26	185902·0	30

Note.

The values in the foregoing table are calculated from the formula

$$V = 140\sqrt{rs} - 11\sqrt[3]{rs};$$

where V = velocity in feet per second, r = *hydraulic mean depth* (= area of wet cross section divided by wet perimeter = for full pipes, ¼ diameter) in feet, s = sine of inclination, or total fall divided by total length.

For pipes of greater diameter than above divide the proposed diameter by 4, then twice the velocity opposite to the quotient in the table will be the velocity required, or the corresponding supply multiplied by 32 will be the approximate supply in gallons per minute.

Requisite Pipe Diameters.

The diameters of the pipes required in a water distribution system can be found by progressive calculation from the point of initial full head. The values in the preceding table provide all the data for the drawing of a graph relating: (*a*) head per 10 ft. or 100 ft. of pipe; (*b*) gallons per minute; (*c*) diameter of pipe. When the required diameter lies between two of the sizes actually in use, the larger is usually to be preferred. Thus, the sizes of pipes can be found in planning a system; or the gallons per minute be calculated in an existing system, the allowance being made in the latter case for deterioration of the pipes.

WEIGHTS OF LEAD PIPE REQUIRED BY WATER COMPANIES.

Company.	Weights, per Yard, of Lead Pipe.						Minimum Depth of Under-ground Pipes.	Weight, per Yard, of Warning Pipe.		
	⅜ In.	½ In.	⅝ In.	¾ In.	1 In.	1¼ In.		½ In.	¾ In.	1 In.
	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.		Lbs.	Lbs.	Lbs.
Metropolitan Water Board	5	6	7½	9	12	16	2 ft. 6 ins.	3	5	7
Colne Valley.	5	7	9	11	16	—	3 ft.	—	—	—
Croydon .	4	5	7	9	12	16	2 ft.	—	—	—
East Surrey .	5	6	8	10	14	—	—	—	—	—
" when pressure exceeds 300 ft.	6	7	9	11	16	—	—	—	—	—
South Essex.	—	6	7½	9	12	16	1 ft. 6 ins.	—	—	—
Sutton . .	—	6	7½	9	12	16	2 ft. 6 ins.	3	5	7
West Surrey.	4	5½	—	9	14	20	3 ft.	—	—	—

Sizes of Lead Pipe are Internal. 1¼ is the usual Minimum Size for Closet Pipe, and 9 lbs. per Yard Minimum Weight.

The Number of Bricks and Quantity of Brickwork in Wells and Cylindrical Sewers for each Foot in Depth or Length.

Diameter of Well, in Feet.	Half-Brick Thick.			One Brick Thick.		
	Number of Bricks.		Cubic Feet of Brickwork.	Number of Bricks.		Cubic Feet of Brickwork.
	Laid Dry.	Laid in Mortar.		Laid Dry.	Laid in Mortar.	
1·0	28	23	1·6198	70	58	4·1233
1·3	33	27	1·8145	80	66	4·7124
1·6	38	31	2·2089	90	74	5·3015
1·9	43	35	2·5035	102	82	5·8905
2·0	48	41	2·7979	112	92	6·4795
2·3	53	44	3·0926	122	100	7·0686
2·6	58	48	3·3870	132	108	7·6577
3·0	68	57	3·9760	154	126	8·8357
3·6	79	65	4·5651	174	142	10·0139
4·0	89	73	5·1541	194	159	11·1919
4·6	100	82	5·7432	214	176	12·3701
5·0	110	90	6·3322	234	192	13·5481
5·6	120	98	6·9213	254	209	14·7263
6·0	130	107	7·5103	276	226	15·9043
6·6	140	115	8·0994	296	242	17·0825
7·0	150	123	8·6884	316	260	18·2605
7·6	160	131	9·2775	336	276	19·4387
8·0	170	140	9·8665	358	292	20·6167
8·6	180	148	10·4556	378	308	21·7949
9·0	191	156	11·0446	398	326	22·9729
10·0	212	174	12·2227	438	360	25·3291

RECENT LITERATURE.

(See also Parts I, II and III.

Books.

'Water Supply Problems and Developments.' By W. H. Maxwell. Second Edition. (Review, *The Engineer*, June 18, 1937.)

'Waterworks for Urban and Rural Districts: including the Supply for Mansions and Isolated Buildings.' By Henry C. Adams. (Reviews, *The Engineer*, June 10, 1938; *The Surveyor*, May 13, 1938.)

'The Examination of Water and Water Supplies.' By E. V. Suckling. 1943.

'The Purification of Water Supplies.' By G. B. Williams. 1944.

'England's Rainfall Problem: a Survey of Rainfall, Drought and Distribution.' By H. Spence-Hales and J. Bland. 1939.

Papers and Addresses.

'Application of Experimental Methods to the Design of Clarifiers for Waterworks.' By R. Walton and T. D. Key. Paper No. 5213, Inst. C.E.

'Water Sterilisation: The Choice of Method.' By E. F. W. Mackenzie. *Bull. Inst. San. E.*, November 1940, and *The Surveyor*, February 7, 1941.

'Science and the Divining Rod.' By J. C. Maby, Royal Society of Arts. Extracts, *The Surveyor*, March 15, 1940.

'The Treatment of Water by Ozone.' By M. T. B. Whitson. Paper, Manchester and District Association, Inst. C.E. (See *The Surveyor*, March 22, 1940.)

'Surge Control in Pipe Lines.' By C. F. Lapworth, Inst. W.E. Abstract, *The Surveyor*, July 7, 1945.

'Concreted Steel Communication Pipes.' By J. F. Haseltine. *Journal Inst. C.E.*, April 1943.

'Roughness Factors in Fluid Motion through Cylindrical Pipes and through Open Channels.' By J. Allen. *Journal Inst. C.E.*, April 1943.

'Problems of Increasing Water Consumption.' Address (Presidential), by S. R. Raffety, Institution of Water Engineers. Reported in *The Surveyor*, May 28, 1943, and reviewed June 6.

'Sedimentation in Reservoirs.' By B. J. Witzig. Paper, *Proc. Am. Soc. C.E.*, June 1943.

'The Problem of Fluorine in Water Supplies.' By H. K. Box and H. J. Hodgins. Paper reproduced *Engineering and Contract Record*, May 31, 1944. Reviewed *The Surveyor*, July 14, 1944. Relates to the teeth.

'Rural Water Supplies.' By S. R. Raffety. *Journal Inst. C.E.*, March 1945. Abstract, *The Surveyor*, April 13, 1945.

'Pumping Stations: with Special Reference to Land Drainage and Storm Water Disposal.' By C. Clay. *Journal Inst. C.E.*, March 1945.

'The Use of Calgon in Treating Chalk Well Waters.' By E. G. B. Gledhill and A. W. H. McCauler. Paper, *Inst. W.E.*

'The Evolution of Waterworks Booster Plants.' By J. P. Hallam. Paper, *Inst. W.E.*

Articles.

'Internally "Sleeving" a 36-in. Water Main.' By A. M. Moon. *The Surveyor*, April 18, 1941. The operation consisted in lining the 36-in. main with pipes of nominal 30-in. diameter, which were double-spigot, bitumen-coated pipes of outside steel diameter 32½ in.; 11 ft. 8 in. long and ⅜ in. thick. Five were used to line the cracked length of the main.

'The Case for the Softening of Hard Water.' By D. G. Davies, *The Surveyor*, February 11, 1944.

'Chlorine Treatment to Restore Yields of Wells.' By H. L. White. *Civil Engineering* (U.S.A.), May 1942.

'Measures for Reducing the Silting of Reservoirs.' Editorial, *The Surveyor*, July 10, 1942. The note takes the form of an original exposition and specific measures are described.

Reports and Annual Publications.

Metropolitan Water Board. The annual reports of the Board present important information and data. With regard to the safeguarding of supplies, measures of purification and the continual examination of the waters collected, stored and distributed, special interest attaches to the annual reports, for 1937 and 1938, of the Board's Director of Water Examinations.

'Water Supply Engineering in the United States.' A review, included in the Progress Report for 1937 furnished by the Committee of the Sanitary Engineering Division, Am. Soc. C.E. Published in *Proceedings*, March 1938. Reference is made to the flow of silt-laden water through reservoirs, and much information in that regard has since been forthcoming. The progress report is reviewed in *The Surveyor*, April 22, 1938.

'Chemical Control of the Waters of Streams and Lakes.' An account, in *Engineering News-Record*, January 4, 1940, of investigations carried out in Wisconsin. Includes a description of a method of treating waters with 0·0003 to 0·0012 lb. of copper carbonate per cu. ft. of water to control the larvæ which cause 'swimmer's itch,' by killing the snails in which they develop as parasites. (See *The Surveyor*, February 23, 1940.)

'Water Supply.' Annual Reviews, *The Surveyor*, for 1943, January 21, 1944; for 1944, January 5, 1945.

Third Report of the Central Advisory Water Committee: 'River Boards,' August 1943

ADDENDA, 1949.

WATERWORKS.

Coventry.—Opened in September 1941, new waterworks provided for the supply of a million gals. a day from the River Avon. The cost was about £30,000.

At the end of August 1948, the Minister of Health made an order enabling Birmingham to set aside in its Eden Valley reservoirs an emergency supply of 2,000 million gals. for Coventry. Both towns have schemes for obtaining further supplies. Coventry's scheme is for a supply from the River Severn and the construction of a reservoir at Bredon Hill, the work having been begun in the summer of 1948. The estimated cost of the works is £3,000,000.

Manchester.—The water level of Haweswater Lake was raised 95 ft. by the building of a dam completed in 1940. The waterspread was increased from 346 acres to 974 acres, and the storage capacity increased by some 180,000 million gals. The dam is 1,550 ft. long and 120 ft. in its greatest height.

Dunwan Reservoir.—Situated in the Eastwood and Mearns Water District of Renfrewshire, the reservoir has a capacity of 360 million gals. The catchment is 1,180 acres in extent. The work was completed in 1939.

Rhyl Waterworks.—Inaugurated in June 1939 additions to the town's waterworks comprise a masonry dam, and the doubling in capacity of the filtering station. The building of the dam, which impounds 270 million gals. in Aled Traf reservoir, involved 35,000 cu. yds. of soft and 20,000 cu. yds. of rock excavation.

Talybont Reservoir.—Opened in June 1939, this Breconshire reservoir provides additional storage of water for Newport (Mon.). The waterspread of the reservoir is 320 acres and its capacity 2,527 million gals. It contains 520,000 cu. yds. of earth and 40,000 cu. yds. of clay puddle. Its length is 1,400 ft. and maximum height 97 ft. The catchment of 6,000 acres has an average annual rainfall of 68½ ins.

Glasgow.—The Corporation's new works programme includes: a pumping station, estimate £25,000; another, with water towers, estimate, £90,000; outlet works, estimate, £29,500, pipe-laying (contracts for four out of five sections let), £278,000.

Liverpool.—Falls of rock in the Hirnant tunnel entail the lining of 3,200 yds., expected to be completed by the autumn of 1949. In the meantime, the water will be diverted to the Aber tunnel which also links Lake Vyrnwy with Hirnant. A new storage reservoir, expected to be completed in August 1951, will cost about £750,000.

Farm Supplies.—In August 1948 the proportion of holdings, of 5 acres or more, with farm-house, and having a piped supply: to the house, England, 50 per cent., Wales, 32 per cent.; to the buildings, England, 39 per cent., Wales, 23 per cent.

Rural Water Schemes.—In June 1948 some 656 rural water supply schemes for 3,052 parishes were under consideration by the Ministry of Health, the estimated cost being £23,457,000. In addition, 1,061 schemes relating to 2,146 parishes had been approved, the estimated cost, £12,221,000. Relating to 609 parishes, 350 schemes had been completed at a cost of £1,619,000, Exchequer grants amounted to £46,000.

Water Softening.—The case for the softening of hard waters was stated and proved by Mr. D. G. Davies in an article published in *The Surveyor*, issue of February 11, 1944. The classification of 718 undertakings with published records is, nearly, one-fifth very soft, 0–50 parts per million; one-fifth normal, 50–100 parts per million; three-fifths hard, in the proportions, 14 per cent., 100–150; 14 per cent., 150–200; 21 per cent., 200–300; 6 per cent., 300–400, and, 'exceptionally hard' 2 per cent., over 400. Only 84, or 11·7 per cent. of the 718 undertakings were operating softening processes. It is estimated that hardness in excess of 100 parts per million costs, per 100 parts per million, about 16*s*. per head per annum.

See also Descriptive Section XVIII, Part VI

SECTION XIX

PART I

HYDRAULICS (MECHANICS OF FLUIDS)
(pp. 743–766)
(By P. L. Boucher, Ph.D. (Lond.), A.M.I.C.E.)

PART II

HYDRAULIC TRANSMISSION OF POWER
(pp. 767–799)
(By P. L. Boucher, Ph.D. (Lond.), A.M.I.C.E.)

PART III

CLASSIFICATION OF PUMPS – RECIPROCATING PUMPS – PUMPING FROM WELLS – CENTRIFUGAL PUMPS – ROTARY PUMPS—AIR LIFT PUMPS – HYDRAULIC RAMS (pp. 801–833)
(By J. Foster Petree, M.I.Mech.E., A.M.I.N.A.)

PART IV

WATER TURBINES (pp. 835–846)
(By P. W. Seewer, D.E. (Zurich), M.I.C.E.)

SECTION XIX

PART I

HYDRAULICS (MECHANICS OF FLUIDS)

(Contributed by P. L. Boucher, Ph.D. (Lond.), A.M.I.C.E.)

General Data.

Weight of Water.

Fresh Water (at 62° F.).

Specific gravity of fresh water at N.T.P. (62° F. and 30 inches mercury barometer) is unity
1 gallon (imperial) weighs 10 lb. or 4·543 kg.
1 gallon (U.S.A.) weighs 8·33 lb. or 3·786 kg.
1 cu. ft. weighs 62·355 lb. or 28·315 kg.
1 cu. metre weighs 2,204·6 lb. or 1000 kg. (at 4° C. for weight in kg.).
1 litre weighs 2·2046 lb. or 1 kg. (at 4° C. for weight in kg.).

Sea Water (at 62° F.).

Average: S.G. = 1·03. 1 cu. ft. weighs 64 lb.
Dead Sea: S.G. = 1·24. 1 cu. ft. weighs 77 lb.

Ice and Snow.

Ice: S.G. = 0·922. 1 cu. ft. at 32° F. weighs 57·50 lb.
Snow: 1 cu. ft. of *fresh snow* weighs from 5 to 12 lb.
1 cu. ft. of *wet and compacted snow* weighs from 15 to 50 lb.

EXPANSION AND WEIGHT OF FRESH WATER AT VARIOUS TEMPERATURES.

Temperature. Degrees F.	Weight of 1 Cu. Ft. Lb.	Weight of 1 Gallon. Lb.
32·0 (melting point)	62·418	10·0101
39·1 (maximum density)	62·425	10·0112
52·3 (ordinary calculations)	62·400	10·0072
62·0 (N.T.)	62·355	10·0000
75·0	62·275	9·9871
100	62·022	9·947
125	61·654	9·887
150	61·201	9·815
175	60·665	9·728
200	60·081	9·635
250	58·75	9·422
300	56·97	9·136
400	54·25	8·700
500	51·16	8·204

Coefficients of Volumetric Expansion for Fresh Water are:—

From 32° F. to 212° F. 0·00026420.
From 212° F. to 392° F. 0·00051020.
From 392° F. to 572° F. 0·00056718.

Pressure of Water.

1 ft. head of water at 62° F.	=	0·433 lb. per sq. in. 0·883 ins. of mercury at 62° F. 0·03045 kg. per cm.^{2} 2·2390 cm. of mercury at 0° C. 30·48 cm. head of water at 62° F.
1 lb. per sq. in.	=	2·3069 ft. head of water at 62° F. 2·0416 ins. of mercury at 62° F. 0·07031 kg. per cm.^{2} 5·170 cm. of mercury at 0° C. 70·31 cm. of water at 62° F.
One metre head of water at 4° C. (maximum density)	=	0·10 kg. per cm.^{2} 7·353 cm. of mercury at 0° C. 3·281 ft. head of water at 62° F. 1·422 lb. per sq. in. 2·900 ins. of mercury at 62° F.
One kg. per cm.^{2}	=	10 metres head of water at 4° C. (see figures above for 1 metre head.)
One atmosphere (14·7 lb. per sq. in.)	=	33·947 ft. head of water at 62° F. 30 ins. of mercury at 62° F. 1·0335 kg. per cm.^{2}. 76·0 cm. of mercury at 0° C. 1,033·5 cm. of water at 62° F.

Compressibility of Water.

Compressibility may be defined as the reciprocal of the Bulk Modulus, K, as follows:—

$$\frac{1}{K} = \frac{1}{v}\left(\frac{dv}{dp}\right)_T$$

Where v = initial volume of water,

and $\left(\frac{dv}{dp}\right)_T$ = rate of decrease of volume with pressure under isothermal conditions (T constant).

Without mathematics, K, the Bulk Modulus, may be defined as the unital pressure which, acting on unit volume, would cause unit volumetric contraction, if the compressibility of the water could remain constant under the pressure. Thus, in any practical case, decrease of volume of water caused by pressure is equal to the original volume, multiplied by the applied pressure, and divided by the Bulk Modulus, K.

Compressibility of water is of importance in problems dealing with sudden stoppage and initiation of fluid motion, particularly where water hammer phenomena are involved.

Values of K determined by various authorities range between 284,000 and 300,000 lb. per sq. in. at 0° C., and 339,000 and 352,000 lb. per sq. in. at 50° C.

The value for K of 300,000 lb. per sq. in., or 43,200,000 lb. per sq. ft., may be taken for most calculations.

K for sea water is about 9 per cent. greater than for fresh water.

Viscosity of Water.

The coefficient of viscosity, μ, is defined as the tangential force on unit area of either of two horizontal planes of indefinite extent, separated by unit distance, one plane being fixed and the other moving with unit velocity, the space between the planes being filled with the viscous substance.

The dimensions of μ in C.G.S. units are grammes per centimetre per second.

Coefficients of Viscosity, μ, of Water at Various Temperatures.

Temperature of Water. Degrees C.	0	20	40	60	80	100
Viscosity. C.G.S. Units	0·0179	0·0101	0·0066	0·0048	0·0036	0·0028

(A. G. M. Michell.)

Resultant Pressure and Centre of Pressure.

Total pressure on any submerged area (single face) is equal to the area multiplied by the depth of the centre of gravity of the area below the free surface, and by the unit weight of the fluid.

The position of the *centre of pressure* is determined by establishing the position of the line of action of the resultant total pressure normal to the surface under consideration. This can be done by taking moments about convenient rectangular axes, of elements of the total pressure, and equating their sum to the moments of the total pressure about the same axes.

Discharge of Water Through Orifices.

General expression for orifices, small in comparison with the head of water above their centres, is :—

$$Q = C.A.\sqrt{2gH}$$

Where Q = Discharge, cub. ft. per sec.
A = Area of orifice, sq. ft.
H = Head of water above centre of orifice, ft.
g = Accleration due to gravity, ft. per sec., per sec.
C = Coefficient of discharge determined experimentally.

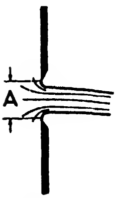


Fig. 1.
Sharp Lipped Orifice.
Average value of C = 0·62.

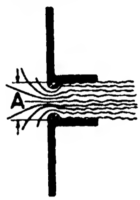


Fig. 2.
Short Tube Orifice (Running Full).
Average value of C = 0·82.

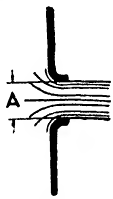


Fig. 3.
Bellmouthed Tube Shaped to Contour of Free Jet.
Average value of C = 0·97.

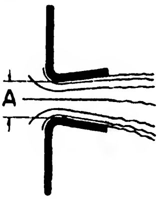


Fig. 4.
Divergent Bellmouth.
Average value of C = 2·0.

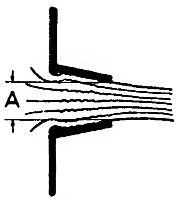


Fig. 5.
Conical Convergent Tube.
Average value of C = 0·90.

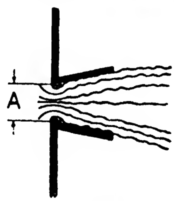


Fig. 6.
Conical Divergent Tube.
Average value of C = 1·46.

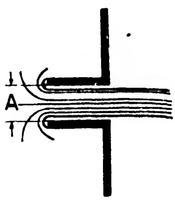


Fig. 7.
Re-entrant Tube (Jet Springing Clear) (Borda's Mouthpiece).
Average value of C = 0·5.

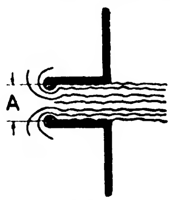


Fig. 8.
Re-entrant Tube (Jet Adhering).
Average value of C = 0·75.

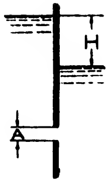


FIG. 9.

Submerged Orifices.

H = difference in level of water surfaces each side of the orifice.

The jet discharges into water and for sharp edged orifices C may be taken as about 0·60.

Large Orifices.—When an orifice is large compared with the head of water over it, variation of velocity from top to bottom lips, due to the increasing head, must (unless the orifice lies in a horizontal plane) be taken into account. The following expression for a sharp edged rectangular orifice is useful:—

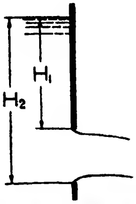


FIG. 10.

$$Q = \tfrac{2}{3}CW\sqrt{2g}\left[H_2^{\frac{3}{2}} - H_1^{\frac{3}{2}}\right]$$

W = width of orifice (unit of measurement 1 ft.).

The value of C may be taken as 0·6 for approximate calculations of sharp edged orifices. Actually C varies slightly with the head of water and the dimensions of the orifice.

Velocity of approach (v_a) may be taken into account by increasing the heads by an amount $h = \dfrac{v_a^2}{2g}$

Submerged Large Orifices.—A submerged large orifice may be treated as a submerged small orifice, since the effective head at all points of the orifice is the same. Velocity of approach may have to be taken into account as described above. The value of C may be taken as about the same as when the orifice is discharged freely into air (actually it is slightly less).

Some Approximate Values of C for Regulator Openings, etc.

For sluices of moderate size in lock gates, etc. .	C = 0·62
For regulator openings between 6 and 13 ft. wide	C = 0·72
For regulator openings above 13 ft. wide . .	C = 0·82
For very large sluices and bridge openings . .	C = 0·92

WEIRS AND NOTCHES.

Formulæ in General Use.

Rectangular Weirs (Sharp Edges).

Q = discharge, cusecs.
L = length of weir, feet.
H = head of water over sill of weir, feet.
n = number of side contractions (2, 1 or 0).
v = velocity of approach, feet per second.
p = height of sill of weir above channel of approach, feet.

Francis.

(1) Neglecting velocity of approach:—

$$Q = 3\cdot33(L - 0\cdot1nH)H^{1\cdot5}$$

(2) Allowing for velocity of approach:—

$$Q = 3\cdot33(L - 0\cdot1nH)\left[\left(H + \frac{v^2}{2g}\right)^{1\cdot5} - \left(\frac{v^2}{2g}\right)^{1\cdot5}\right]$$

N.B.—*The Francis Formulæ apply accurately (within ± 1 per cent.) to weirs from* 8 *to* 10 *ft. wide with heads of from* 0·6 *to* 1·6 *ft. and velocities of approach between* 0·2 *and* 1·0 *ft. per sec. Weir lip should be between* 2 *ft. and* 5 *ft. above bottom of tank.*

Barnes.

(1) For weirs with *two* end contractions :—

$$Q = 3\cdot324\ H^{1\cdot49}\ L^{1\cdot33}\ (L + 2H)^{-0.11}$$

(2) For weirs *without* end contractions :—

$$Q = 3\cdot324\ H^{1\cdot49}\ L^{0.98}\ (L + 2H)^{0.02}$$

Gourley & Crimp.

For weirs *with* end contractions :—

$$Q = 3\cdot10\ L^{1.03}\ H^{1.47}$$

For weirs up to at least 19 *ft. long and heads up to half the length of weir and depth of pool below sill of weir not less than twice the head.*

Rehbock.

For weirs *without* end contractions :—

$$Q = \tfrac{2}{3}\left(0\cdot605 + \frac{1}{320H - 3} + 0\cdot08\,\frac{H}{p}\right)\sqrt{2g}\ LH^{1.5}$$

The Rehbock Formula is easy to apply and appears to give results that are very accurate. See Paper and Discussion : 'Precise Weir Experiments' by Schoder & Turner, Trans. A.S.C.E., Vol. 93, *p.* 999 (1929).

BRITISH STANDARD SPECIFICATION FOR PUMP TESTS.

The rectangular weir is recommended for the measurement of discharges exceeding 250 G.P.M., either a weir with suppressed end contractions or a fully contracted weir being used.

The head is measured at a point upstream at a distance from the weir of approximately six times the maximum head to be measured, and at the side of the channel of approach. If the weir is in the open and its length exceeds 5 ft., two gauges should be used, one on each side, and the mean head adopted.

(1) Weirs *without* end contractions (suppressed weirs).

$$Q = L\left(3\cdot23 + 0\cdot434\frac{H}{p}\right)(H + 0\cdot0034)^{1\cdot5}$$

Allowance for velocity of approach is included in this equation.

The width of channel of approach should be equal to the length of the weir crest and the downstream channel should be the same width for a distance of 2 or 3 ft. from the weir. The weir crest should be at a height above the approach channel floor of not less than one and a half times the maximum head to be measured and not more than 4 ft.

The Nappe should be fully aerated and the downstream water level should not rise higher than 3 ins. below the lip. The area of any ventilating openings should not be less than L × H square inches.

Accuracy to be expected is ± 1·5 per cent. for heads between 2 ins. and 30 ins.

(2) *Fully contracted Weirs.*

$$Q = 3\cdot29(L - 0\cdot1H)H^{1\cdot5}$$

The sides of the channel of approach should not be nearer any point of the weir lip than 4 times the head and the bottom of the channel should not be nearer to the weir lip than 3 times the head, with minimum distances in every direction of 12 ins.

Downstream neither the sides of the channel nor the bed should be nearer any point of the weir lip than 6 ins. and the downstream water level should not rise higher than 3 inches below the weir lip.

The formula applies to rectangular weirs of any length from 12 ins. upwards having complete contractions and applies to heads of from 3 ins. to 24 ins. provided that $\frac{L}{H}$ is greater than 2.

Accuracy to be expected is ± 2 per cent.

See also B.S.S. for Pump Tests, No. 599—1939.)

TRAPEZOIDAL WEIRS.

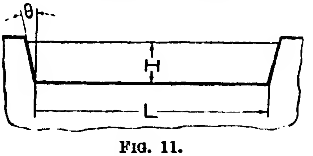


FIG. 11.

Gourley & Crimp.

$$Q = 3\cdot10L^{1\cdot03}H^{1\cdot47} + 2\cdot48H^{2\cdot47}\tan\theta$$

Cippoletti Weir.

If the value of θ is selected in such a way that the discharge over a trapezoidal weir is the same as that over a rectangular weir *without* end contractions of the same length L, then the discharge can be calculated by any of the appropriate formulæ above. The value of determined by Cippoletti is such that tan $= 0 \cdot 25$. Thus a trapezoidal weir with end slopes of 1 horizontal to 4 vertical, is a Cippoletti weir.

Vee Notches (Sharp Edges).

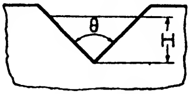


Fig. 12.

General Expression.

$$Q = \frac{8C}{15} \sqrt{2g} \tan \frac{\theta}{2} H^{2 \cdot 5}$$

Where C = coefficient of discharge and varies from $0 \cdot 593$ for a right angle notch to $0 \cdot 62$ for greater angles.

90° *Notches.*

Thomson. $Q = 2 \cdot 536 H^{2 \cdot 5}$

Barnes. $Q = 2 \cdot 48 H^{2 \cdot 48}$

British Standard Specification for Pump Tests.

The vee notch is recommended for the measurement of discharges from 15 to 1,500 G.P.M. For the smaller discharges a half 90° vee notch may be used and for the larger discharges a 90° vee notch. The discharge of water over a half 90° vee notch is half that over a 90° vee notch with the same head.

The head should be measured in the corners of the flume formed by the notch bulkhead if the flume is sufficiently wide, or at the sides of the flume at a distance upstream from the weir, approximately 4 times the maximum head to be measured.

The depth of the bottom of the channel below the apex of the notch should not be less than 6 ins. on the downstream side, while on the upstream side it should not be less than 12 ins. for heads up to 9 ins., nor less than 18 ins. for larger heads. The width of the channel of approach should not be less than 4 ft. for heads up to 9 ins. nor less than 6 ft. for heads up to 18 ins. The water level downstream should not be allowed to rise closer than within 1 in. of the apex of the notch.

90° *Vee Notch.*

$$Q = 2 \cdot 48 H^{2 \cdot 48} \quad \text{(Barnes's Formula)}$$

Half 90° *Vee Notch.*

$$Q = 1 \cdot 24 H^{2 \cdot 48}$$

Accuracy to be expected if every care is taken with setting and reading of the gauges, etc., is $\pm 1 \cdot 5$ per cent. for heads between 3 ins. and 15 ins.

(See also B.S.S. for Pump Tests. No. 599—1939.)

Effect of Velocity of Approach.

Velocity of approach when not allowed for in the formula used may be reckoned as equivalent to an additional head proportional to $\frac{v^2}{2g}$ and added to the head over the sill of the weir. Due to higher surface velocity $\frac{v^2}{2g}$ is usually multiplied by a constant greater than unity, as follows:—

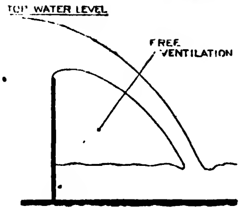
No end contractions, head correction $= 1 \cdot 33 \frac{v^2}{2g}$

Two end contractions, head correction $= 1 \cdot 1$ to $2 \cdot 5 \frac{v^2}{2g}$

(*Smith.*)

N.B.—Velocity of approach must be kept low for accurate work.

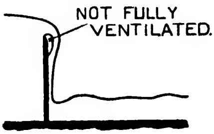
Effects of Shape of Nappe.



Free Nappe.

Fig. 13.

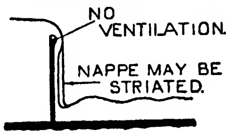
A free nappe, with full bottom contraction is required for accurate measurement of flow.



Depressed Nappe.

Fig. 14.

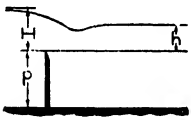
Flow increased some 8 to 10 per cent. above that with free nappe.



Adherent Nappe.

Fig. 15.

Flow 20 to 30 per cent. greater than with free nappe.



Drowned Nappe.

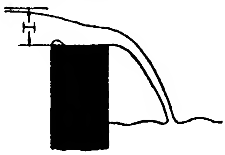
Fig. 16.

Flow about 50 per cent. greater than with free nappe.

Limiting Conditions for Drowned Nappe.

(1) $H > 4p$
(2) $h < \frac{2}{3}p - H$

To calculate discharge of drowned weir, multiply free discharge by $1 \cdot 05 + 1 \cdot 5\frac{h}{H}$

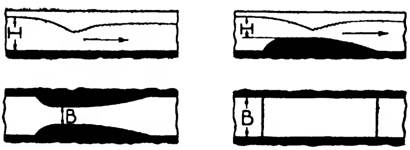


Broad Crested Weirs.

Fig. 17.

Simple approximate expression:—

$$Q = 0 \cdot 35\sqrt{2g}LH^{1 \cdot 5}$$



FLUME WITH SIDE CONTRACTIONS. FLUME WITH HUMP.

Standing Wave Flume.

Fig. 18.

The expression for Broad Crested Weirs is applied:—

The simple approximation is:—

$$Q = 0 \cdot 35\sqrt{2g}BH^{1 \cdot 5}$$

N.B.—For more precise information see 'Hydraulic Measurements,' *Addison* (Chapman & Hall, 1940).

APPROXIMATE DISCHARGE OF WATER OVER A RECTANGULAR WEIR ONE FOOT WIDE WITHOUT END CONTRACTIONS: $Q = 3 \cdot 33\ LH^{1 \cdot 5}$.

Head in Ft.	Discharge. Cusecs.	Discharge. G.P.M.
0·025	0·0131	4·9
0·05	0·0373	13·9
0·10	0·1052	39·4
0·20	0·298	111·8
0·30	0·547	205·0
0·40	0·841	315·0
0·50	1·176	440·0
0·60	1·55	580·0
0·70	1·95	730·0
0·80	2·38	890·0
0·90	2·84	1065·0
1·00	3·33	1248·0
1·10	3·84	1440·0
1·20	4·37	1640·0
1·30	4·92	1845·0
1·40	5·50	2060·0
1·50	6·10	2283·0
1·60	6·71	2510·0
1·70	7·37	2760·0
1·80	8·03	3010·0
1·90	8·70	3260·0
2·0	9·41	3520·0

DISCHARGE OF WATER OVER VEE NOTCHES.

Calculated from B.S.S. Formulæ.

Head. Ins.	90° Notch.		Half 90° Notch.	
	Cusecs.	G.P.M.	Cusecs.	G.P.M.
3·0	0·080	29·778	0·040	14·889
3·5	0·117	43·652	0·058	21·826
4·0	0·163	60·789	0·081	30·394
4·5	0·218	81·412	0·109	40·706
5·0	0·283	105·72	0·141	52·86
5·5	0·358	133·91	0·179	66·95
6·0	0·445	166·16	0·222	83·08
6·5	0·542	202·65	0·271	101·33
7·0	0·652	243·59	0·326	121·77
7·5	0·773	288·98	0·386	144·49
8·0	0·907	339·14	0·454	169·57
8·5	1·054	394·16	0·527	197·08
9·0	1·215	454·20	0·607	227·10
9·5	1·389	519·37	0·695	259·68
10·0	1·578	589·83	0·789	294·91
10·5	1·781	665·68	0·890	332·84
11·0	1·999	747·09	0·999	373·54
11·5	2·232	834·17	1·116	417·03
12·0	2·480	927·01	1·240	463·50
12·5	2·744	1025·8	1·372	512·90
13·0	3·025	1130·6	1·512	565·3
13·5	3·321	1241·5	1·661	620·7
14·0	3·635	1358·7	1·817	679·3
14·5	3·965	1482·2	1·983	741·1
15·0	4·313	1612·2	2·156	806·1

(Condensed from B.S.S. No. 599—1939.)

BELLMOUTH WEIRS.

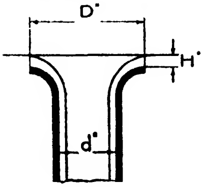


FIG. 19.

Pipe Bellmouths.—Weir flow occurs over the lip of a vertical bellmouth when the head is not so great that the outlet pipe becomes flooded.

Experiments by John Barr show that for Bellmouth weirs with D between two and two and a half times d, the head H should not exceed $\frac{d}{2}$ for pipes smaller than 5 ins. diameter or $\frac{2d}{3}$ for pipes of from 6 ins. to 20 ins. diameter.

Barr determined the following expression for discharge within these limits.

$$Q = 11{\cdot}73DH^{1.35}$$

Where Q = discharge, G.P.M.
D = diameter of bellmouth, inches.
H = head of water over lip of bellmouth, inches.

Large Bellmouth Weirs for Tunnel Outlets.—Recent data on the design of large bellmouth weirs (up to 160 ft. diameter) is furnished in a paper by W. J. E. Binnie ('Transactions of the Institution of Water Engineers,' Vol. XLII, 1937, p. 103), in which such weirs as Taf Fechan, Silent Valley, Pontian Ketchil, Davis Bridge, Burnhope, Manuherika Falls and Jubilee are described and model experiments detailed.

The expression $Q = CLH^{1.5}$ (where L = length of lip) is given for various model experiments (with C varying from 2·06 to 4·04 for various models) and it is concluded that model experiments may be used to establish approximately the relation between Q and H for a prototype. A curtain wall dividing the weir into two semicircular weirs was found to be the best anti-vortex device.

The original paper should be consulted for most useful information on large existing bellmouth weirs and for details of model experiments.

SIPHON WEIRS AND SPILLWAYS.

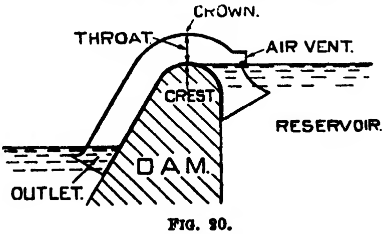


FIG. 20.

Siphon weirs or spillways, usually constructed monolithically with the dam, are employed to give a greater discharge than could be obtained with the same reservoir head and the same length of plain spillway.

Two important features of design are the priming arrangements and the breaking arrangements both of which can be made automatic.

For priming an air pump can be used, an auxiliary 'baby' siphon, a priming weir, a joggled lower limb, and other devices which, at a certain reservoir level, cause the exhaust of air from the siphon until flow commences under suction head.

In breaking a siphon the important object is to admit air to cause flow to cease before the reservoir level has been drawn down to a wasteful extent. This can be achieved by arranging air vents to the siphon crest which are uncovered as the reservoir level falls, or by providing water column or mechanical air valves of special design.

The degree of vacuum under which siphons operate influence the design in an important respect. For high vacuum heads (over 30 ft.) the outlet area is reduced by a nozzle or by tapering the lower part of the outlet limb. For medium heads (20 to 10 ft.) siphons are of constant cross sectional area. For low heads (up to 10 ft.) divergent lower limbs are generally employed with the object of taking advantage of the increased discharge thus obtained.

Coefficient of Discharge.—If a siphon is considered as a pipe or orifice, discharge may be calculated from the total head of the outlet area.

Let Q = discharge of siphon, cusecs.
A = outlet area, square feet.
H = total head, feet.
C = coefficient of discharge.

$$\text{Then } Q = CA\sqrt{2gH}$$

If the throat area is taken instead of the outlet area C may have a value greater than unity.

Efficiency.—A. H. Naylor suggests that efficiency of a siphon should be taken as the ratio of discharge to the discharge of a 'perfect' siphon of the same throat area with perfect vacuum at throat (34 ft. at sea level).

$$\text{This efficiency, } \eta = \frac{Q}{A\sqrt{2g\,h_a}}$$

Where Q = discharge of the siphon.
A = throat area.
h_a = atmospheric pressure.
(units, feet and seconds.)

Values of C and Q in the above expressions can be calculated or estimated from scale model experiments. No general rules for design can be given here, but reference should be made to published literature:—

Brent Reservoir (*Civil Engineering*, August 1937).
Champlain Canal (*Engineering News*, October 13, 1910).
Maramsilli Reservoir (Powys Davies, *Proc. Inst.C.E.* Paper No. 4602).
Bridgeport Dam, Nevada, 1926.
Tests of Siphon Weirs (Corwin & Kidder, *Engineering News Record*, Vol. 109, p. 649).
Tummel Development (*The Engineer*, July 6 and 13, 1934).
Shing Mun Dam (*The Engineer*, January 22, 1937).
Lochaber Water Power Scheme, Laggan Dam (Naylor, *Proc. Inst. C.E.* December 1936).
'Siphon Spillways,' by A. H. Naylor (Edward Arnold & Co. London, 1935).

Venturi Meters, Flow Nozzles and Orifice Gauges.

Fundamental Expression for Flow.

$$H = \frac{V^2\left(\frac{D^4}{d^4} - 1\right)}{C^2\,2g}$$

which may also be written :—

$$Q = 20\cdot82\,AC\sqrt{\frac{H}{\frac{D^4}{d^4} - 1}}$$

Where H = differential venturi head, feet of water.
V = velocity in main, feet per second.
Q = flow in main, G.P.M.
D = upstream diameter, inches.
A = upstream area, square inches.
d = diameter of throat of venturi tube, of flow nozzle or of orifice, inches.
C = coefficient.
F = loss of head, feet of water.

Venturi Meters.—The value of C depends upon the design of meter and the conditions of flow. An average value is 0·976.

The loss of head F is a function of the velocity head $\frac{V^2}{2g}$, of the ratio $\frac{D}{d}$ and it depends also upon the design of the venturi tube. A reliable figure can be obtained only from the makers, F is often less than 1 foot at maximum rate of flow.

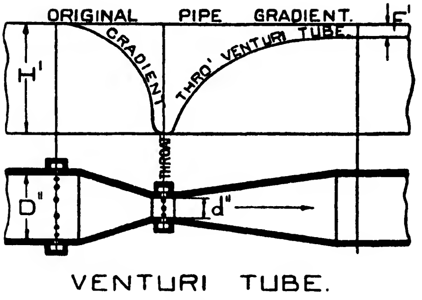


FIG. 21.

The veturi meter is not recommended in smaller sizes than 2 ins. diameter, nor for pipe velocities less than 1 ft. per sec.

There should be at least 10 diameters of straight parallel pipe upstream of the meter.

Accuracy to be expected is ± 1·5 per cent.

Flow Nozzles

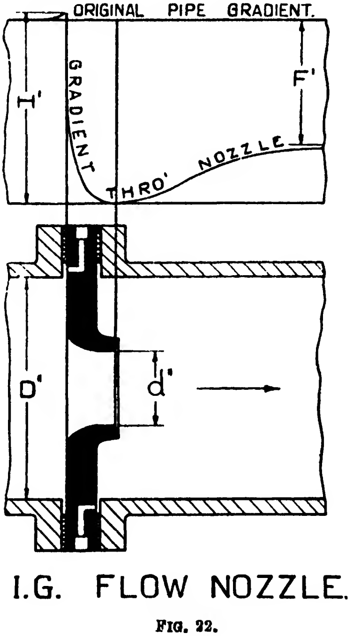


FIG. 22.

The value of C for various ratios of $\frac{D^2}{d^2}$ are given below :—

$\frac{D^2}{d^2}$	C
10	0·985
5	0·980
3·33	0·970
2·5	0·960
2·0	0·950

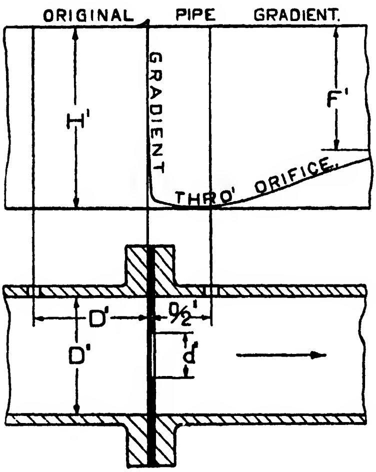
These values are correct for differential heads not less than $\frac{7}{\sqrt{D}}$

The loss of head in a flow nozzle is greater than in a venturi tube and may be from 1 foot to 3 feet or more at normal rate of flow.

There should be at least 10 diameters of straight parallel pipe upstream of the nozzle and not less than 5 diameters of straight parallel pipe of the same diameter downstream.

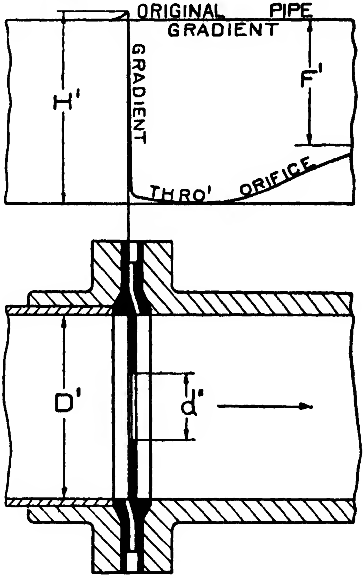
Accuracy to be expected is ± 2 per cent.

Orifice Gauges.—Two forms of orifice gauge are recommended. One consists of a thin plate clamped between pipe flanges and having holes for the pressure take-offs drilled in the adjacent



THIN PLATE ORIFICE.

FIG. 23.



SQUARE EDGE ORIFICE. WITH CLOSE TAPS.

FIG. 24.

pipes. The upstream pressure tap is situated at one pipe diameter upstream from the adjacent face of the orifice plate, while the downstream tap is situated at a distance of $\left(\frac{D}{2}\right)$ from the upstream face of the plate. The other is similarly intended for clamping between pipe flanges, but is of sufficient thickness to contain its own pressure take-offs, located in the corners immediately before and after the orifice.

It is not recommended that an orifice gauge be used for ratios of $\left(\frac{D}{d}\right)$ greater than 0·7. Values of C within this limit are 0·612 for taps at D and at $\left(\frac{D}{2}\right)$ and 0·605 for orifices with close taps.

There should be not less than 10 diameters of straight parallel pipe upstream of the plate and not less than 5 diameters of straight parallel pipe of the same diameter downstream.

The length of parallel hole in the plate shall not be more than $\left(\frac{d}{20}\right)$.

The loss of head F is of the same order as for flow nozzles.

Within the prescribed limits accuracy to be expected is ± 2 per cent.

(See B.S.S. for Pump Tests, No. 599—1939.)

THE PITOT TUBE.

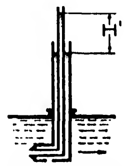


FIG. 25.

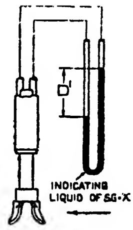


FIG. 26.

The Pitot tube is often the most convenient means of measuring flow in pipelines, especially those of large diameter, and in other forms of conduits and open channels.

In making gaugings in a pipe it is usual to measure the centre velocity and to calculate the discharge by multiplying this velocity by the coefficient (which is a constant represented by the mean velocity across the pipe, divided by the centre velocity) and by the area of the pipe. To obtain the pipe coefficient, the pipe should be traversed with the Pitot tube on 2 diameters at right angles and the average taken of the mean velocity divided by centre velocity for each traverse.

The best form of Pitot tube (the Pitometer) has two orifices which are set to point upstream and downstream. These two orifices are symmetrical and can be reversed to check accuracy. The effect of the upstream and trailing orifices is to produce a greater differential head than is produced by a single upstream orifice.

Expression for Velocity of Flow.

$$V = K\sqrt{2gH}$$

Where V = velocity of flow, feet per second.
K = instrument coefficient (may be taken as 0·990 for ordinary type of Pitot tube).
g = acceleration due to gravity, feet per second per second.
H = differential head between static and impact pressures, feet of water.

N.B.—If indicating liquid has specific gravity x, and D = deflection in feet of this liquid in U-tube, H in the above expression may be written $D(x - 1)$.

Expression for Rate of Flow.

$$Q = 374{\cdot}1\, K_p A V_c$$

Where Q = rate of flow, G.P.M.
K_p = pipe coefficient (mean velocity divided by centre velocity).
A = effective area of pipe, square feet
V_c = centre velocity, feet per second.
(If 'Q' is required in cusecs, substitute 1·0 for 374·1 in above expression.)

A length of straight pipe not less than 15 diameters should be provided upstream of the gauging point and a length not less than 5 diameters downstream.

Accuracy to be expected is $\pm$ 2 per cent.

Current Meters.—Accurate measurements of flow cannot be made with current meters, but in favourable circumstances where the stream of flowing water is of reasonably uniform cross-section and free from serious cross currents and eddies, the current meter, in the hands of an experienced observer, will give results accurate to $\pm$ 3 per cent.

The Salt Velocity Method.—Often, use has been made of the salt velocity method for measuring the flow of large volumes of water. Salt in solution increases the electrical conductivity of water and if brine is introduced into the stream at a convenient point, the passage of water past one or more sets of electrodes at different points of the channel at noted time intervals, recorded graphically, provides means of calculating the flow.

Accuracy to be expected depends entirely upon individual circumstances.

Water Meters.—Various positive and inferential types of proprietary water meters of the integrating type are available. Some of them, particularly the positive reciprocating type of meter, can have remarkable high accuracy (as high as $\pm$ one-fifth of 1 per cent.) and there are many circumstances in which such meters, especially when check calibrated, can be used to great advantage.

Some type of proprietary water meters of the integrating type are described on p. 191.

Flow in Channels and Pipes.

General Equation. (Chezy Formula.)

$$V = C\sqrt{RS}$$

$$\text{Or} \quad Q = CA\sqrt{RS}$$

Where V = velocity of flow, feet per second.

Q = rate of flow, cusecs.

A = cross sectional area of water, square feet.

R = hydraulic radius, feet.

$$= \frac{\text{cross-sectional area of water, A, square feet}}{\text{wetted perimeter, feet}}$$

S = slope of channel or hydraulic gradient of pipe.

$$= \frac{\text{difference of level of free water surface, feet}}{\text{horizontal distance between points of measurement, feet}}$$

This formula is the basis of all formulæ for flow in pipes and channels. But the value of C is variable and not easily established in general terms, and many expressions have been developed for its value in various circumstances.

For very rough calculations C may be taken as 100.

Kutter's Formula. (For C in Chezy Formula.)

$$C = \frac{41\cdot65 + \dfrac{1\cdot811}{N} + \dfrac{0\cdot00281}{S}}{1 + \dfrac{N}{\sqrt{R}}\left(41\cdot65 + \dfrac{0\cdot00281}{S}\right)}$$

Where N = a coefficient denoting the roughness of the conduit.

SOME VALUES OF KUTTER'S N FOR VARIOUS SURFACES. (HORTON.)

Surface.	Perfect.	Good.	Fair.	Bad.
Uncoated cast iron pipe	0·012	0·013	0·014	0·015
Coated cast iron pipe	0·011	0·012*	0·013*	—
Commercial wrought iron pipe, black	0·012	0·013	0·014	0·015
Commercial wrought iron pipe, galvanised	0·013	0·014	0·015	0·017
Smooth brass and glass pipe	0·009	0·016	0·011	0·013
Smooth lockbar and welded 'OD' pipe	0·010	0·011*	0·013*	—
Riveted and spiral steel pipe	0·013	0·015*	0·017*	—
Vitrified sewer pipe	0·010 / 0·011	0·013*	0·015*	0·017
Glazed brickwork	0·011	0·012	0·013*	0·015
Brick and cement mortar; brick sewers	0·012	0·013	0·015*	0·017
Neat cement surfaces	0·010	0·011	0·012	0·013
Cement mortar surfaces	0·011	0·012	0·013*	0·015
Concrete pipe	0·012	0·013	0·015	0·016
Wood-stave pipe	0·010	0·011	0·012	0·013
Plank flumes:				
Planed	0·010	0·012	0·013	0·014
Unplaned	0·011	0·013	0·014	0·015
With battens	0·012	0·015	0·016	—
Concrete lined channels	0·012	0·014*	0·016*	0·018
Cement-rubble surface	0·017	0·020	0·025	0·030
Dry rubble surface	0·025	0·030	0·033	0·035
Dressed-ashlar surface	0·013	0·014	0·015	0·017
Semi-circular metal flumes, smooth	0·011	0·012	0·013	0·015
Semi-circular metal flumes, corrugated	0·022	0·025	0·0275	0·030
Canals and ditches:				
†Earth straight and uniform	0·017	0·020	0·0225*	0·025
Rock cuts, smooth and uniform	0·025	0·030	0·033*	0·035
Rock cuts, jagged and irregular	0·035	0·040	0·045	—
Winding sluggish canals	0·0225	0·025*	0·0275	0·030
Dredged earth channels	0·025	0·0275*	0·030	0·033
Canals, with rough stony beds, weeds on earth banks	0·025	0·030	0·035*	0·040
Earth bottom, rubble sides	0·028	0·030*	0·033*	0·035
Natural stream channels:				
(1) Clean, straight bank, full stage, no rifts or deep pools	0·025	0·0275	0·030	0·033
(2) Same as (1), but some weeds and stones	0·030	0·033	0·035	0·040
(3) Winding, some pools and shoals, clean	0·035	0·040	0·045	0·050
(4) Same as (3), lower stages, more ineffective slope and sections	0·040	0·045	0·050	0·055
(5) Same as (3), some weeds and stones	0·033	0·035	0·040	0·045
(6) Same as (4), stoney sections	0·045	0·050	0·055	0·060
(7) Sluggish river reaches, rather weedy or with very deep pools	0·050	0·060	0·070	0·080
(8) Very weedy reaches	0·075	0·100	0·125	0·150

* Values commonly used in designing.

† In British Indian practice the values would be shifted one column to the left, 0·017 being hardly attainable with earthen channels; 0·020, perfect, 0·0225, good, 0·025 fair, 0·030 bad. See Section XVIII, part IV.

Effect of S, N and R in C in Chezy Formula. (Angus.)

S	N	R	C
0·0001	0·010	1·0	147
0·0001	0·020	1·0	67
0·0001	0·010	10·0	205
0·0001	0·020	10·0	111
0·01	0·010	1·0	156
0·01	0·020	1·0	72
0·01	0·010	10·0	196
0·01	0·020	10·0	105

The above table shows that C varies only slightly and somewhat irregularly with S, while C varies much more rapidly with R and almost inversely with N.

Manning's Formula. (For C in Chezy Formula.)

A simpler expression for C than Kutter's is :—

$$C = \frac{1{\cdot}486R^{\frac{1}{6}}}{N}$$

Where N is the same value as Kutter's N.

Bazin's Formula. (For C in Chezy Formula.)

$$C = \frac{157{\cdot}6}{1 + \frac{M}{\sqrt{R}}}$$

Some Values for Bazin's M.

For smooth cement or planed wood surfaces	M = 0·109
For planks, ashlar, brickwork	= 0·290
For rubble masonry	= 0·833
For earth channels of very regular surfaces	= 1·540
For ordinary earth channels	= 2·360
For exceptional rough channels encumbered with weeds and boulders	= 3·170

Hazen and Williams' Formula. (For Flow in Pipes and Channels.)

$$V = 1{\cdot}319CR^{0{\cdot}63}\,S^{0{\cdot}54}$$
$$= 1{\cdot}319CR^{0{\cdot}13}\,S^{0{\cdot}04}\sqrt{RS}$$

N.B.—C in this formula is not a coefficient in the Chezy Formula. Some values of Hazen and Williams' C for round pipes are as follows :—

For extremely smooth pipes and channels	C = 140
For very smooth pipes and channels	= 130
For good masonry aqueducts	= 120
For new steel riveted pipes and tiled sewers	= 110
For ordinary cast iron pipes, steel pipes 10 years old, and old brick sewers	= 100
For very rough pipes	= 60

Barnes's Formula. (For New Clean Asphalted Cast Iron Pipes.)

$$Q = 47{\cdot}087D^{2{\cdot}769}\,S^{0{\cdot}539}$$

Where D = diameter of pipe, feet.
S = hydraulic gradient of pipe.

Crimp and Bruges Formula. (For Flow in Sewers.)

$$V = 124R^{0{\cdot}67}\,S^{0{\cdot}50}$$

According to Alexander this formula holds for all good ordinary pipe and brick sewers up to 36 ins. diameter, but the coefficient 124 may have to be reduced by as much as 25 per cent. for old sewers and possibly increased by the same amount for large new brick culverts.

Scimemi's Formula. (For Flow in Asbestos-cement Pressure Pipes.)

$$V = 241R^{0{\cdot}68}\,S^{0{\cdot}56}$$

This formula was established by original experiments at Padua in 1925 and was verified at the National Physical Laboratory, Teddington, in 1937.

Turners Asbestos Cement Company recommend that the losses calculated for asbestos-cement pipes by the Scimemi Formula be increased by from 4 per cent. to 7½ per cent. depending upon the size of main and the conditions of laying, to compensate for individual imperfections. The greatest allowance is made for the smallest mains and the least allowance for the largest mains.

N.B.—*In using Scimemi's Formula for asbestos-cement pipes care should be taken to calculate the value of R from the actual internal diameters of the pipes, which are not necessarily equal to the nominal diameters. Four classes of pipes, A, B, C and D, are defined by B.S.S.* 486–1933 *and internal diameters are given in the specification.*

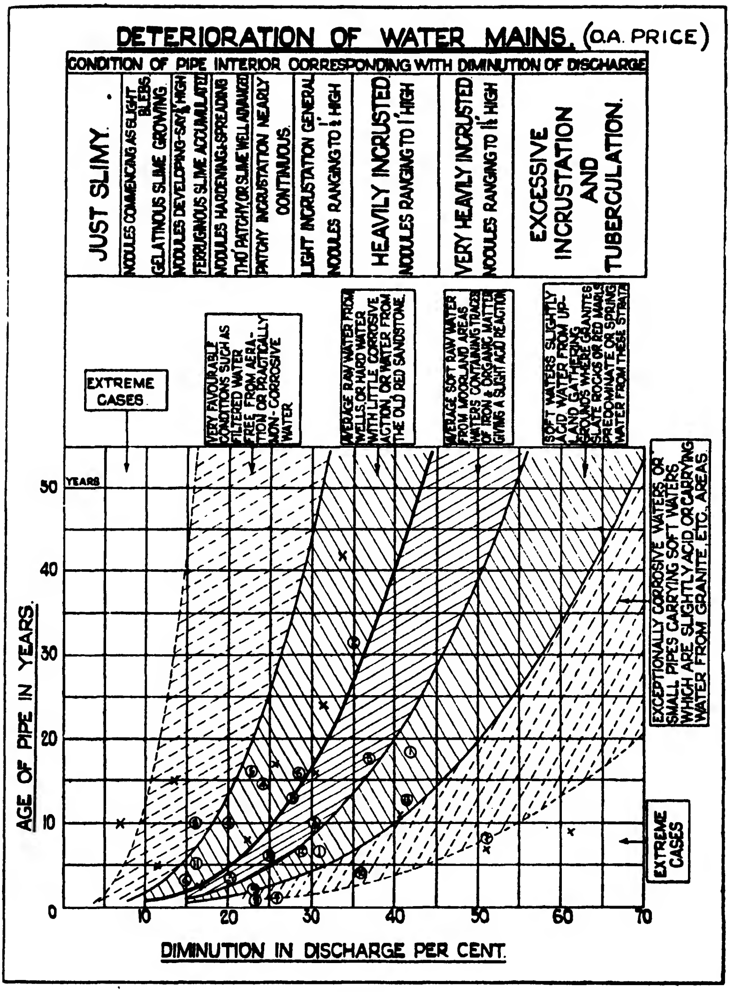


FIG. 27.

The above diagram has been constructed from records of pipeline capacities with various types of water extending over periods up to 42 years. The numbered points represent measured loss of capacity at specific periods of time, each number referring to a particular pipeline for which the type of water and conditions of the pipe interior were known. Unnumbered points refer to pipelines in cases where the quality of the water and the state of the pipe surfaces were not recorded.

Flow of Water in New Clean

Figures taken from Diagram for the Solution of the
Where Q = Discharge in cubic feet per
D = Diameter of pipe in feet
S = Slope or hydraulic gradient.

Hydraulic Gradient—1 in	Internal Diameter 3	4	5	6	7	8	9	10	12	14	15	16	18	20
10	111	245	460	750	1,160	1,660	2,350	3,100	5,250	7,800	9,500	11,500	15,750	21,400
15	89	199	366	615	930	1,335	1,890	2,515	4,200	6,350	7,600	9,350	12,650	17,100
20	76	170	318	525	800	1,160	1,600	2,180	3,590	5,450	6,600	7,900	10,950	14,600
30	62	137	256	424	650	940	1,300	1,750	2,900	4,410	5,400	6,450	8,850	11,900
40	53	117	220	360	560	800	1,130	1,490	2,500	3,700	4,580	5,500	7,500	10,150
50	47	105	197	322	495	705	1,000	1,330	2,220	3,350	4,050	4,900	6,700	9,100
60	43	95	177	294	445	650	910	1,200	2,020	3,050	3,650	4,450	6,100	8,200
70	40	87	161	270	410	590	825	1,115	1,860	2,800	3,390	4,050	5,600	7,500
80	36	81	150	253	381	555	775	1,040	1,725	2,610	3,180	3,800	5,250	7,000
90	35	76	142	239	360	525	725	980	1,610	2,460	2,970	3,550	4,900	6,550
100	33	71	134	225	340	495	670	915	1,530	2,330	2,800	3,370	4,650	6,250
120	30	66	123	207	310	450	630	850	1,400	2,150	2,600	3,100	4,300	5,750
150	27	58	110	184	277	400	565	750	1,250	1,890	2,300	2,770	3,750	5,100
200	23	51	94	157	240	340	480	640	1,080	1,610	1,985	2,390	3,250	4,390
250	21	45	84	139	215	305	430	575	960	1,450	1,770	2,150	2,900	3,900
300	19	41	76	127	194	277	385	525	865	1,310	1,590	1,910	2,600	3,500
350	17	37	69	117	178	255	351	475	790	1,200	1,450	1,750	2,400	3,200
400	16	35	65	108	164	238	330	440	730	1,120	1,340	1,610	2,240	2,990
450	15	33	62	102	155	225	312	418	685	1,070	1,280	1,520	2,100	2,800
500	14	31	57	96	145	214	295	395	655	1,000	1,200	1,440	2,000	2,700
550	13·5	30	55	92	139	202	280	375	620	950	1,150	1,360	1,900	2,530
600	13	28	53	87	133	193	268	358	595	910	1,100	1,310	1,800	2,450
650	12·5	27	50	84	127	184	257	340	570	865	1,055	1,270	1,715	2,320
700	12	26	48	80	123	177	248	329	550	835	1,010	1,210	1,650	2,240
800	11	24	45	74	114	164	230	306	525	780	940	1,130	1,540	2,090
900	10·5	23	42	70	107	154	219	290	480	730	880	1,065	1,450	1,980
1,000	9·8	22	40	66	101	144	205	272	450	680	835	1,000	1,365	1,850
1,200	8·9	20	36	61	93	132	187	250	412	630	755	910	1,235	1,690
1,500	8	17·5	32	54	81	118	165	221	362	560	670	805	1,115	1,485
2,000	6·7	15	28	46	70	102	140	190	318	480	580	690	960	1,290
3,000	—	12	22·5	37	57	82	115	152	255	388	470	565	770	1,040
4,000	—	10·5	19	32	49	69	98	131	220	330	395	480	655	885
5,000	—	9	17	28	44	63	87	117	194	295	353	425	580	785
6,000	—	8·5	15·5	26	40	57	79	107	177	267	325	386	535	710
7,000	—	—	14	24	36	52	72	98	160	248	295	352	490	655
8,000	—	—	13	22	34	49	67	91	150	232	278	330	459	620
9,000	—	—	12·5	21	32	46	64	86	141	220	260	310	430	575
10,000	—	—	12	20	30	44	62	82	136	210	250	300	410	560

Table Gives Discharge

Asphalted Cast Iron Pipes.

Expression $Q = 47{\cdot}087\ D^{2{\cdot}768}\ \ S^{0{\cdot}520}$ (A. A. Barnes.)
second (converted to G.P.M. in tables.)
(converted to inches in tables).

of Main—Inches.

21	22	24	27	28	30	33	36	40	44	48		Hydraulic Gradient—1 in ....
24	28,000	35,000	49,000	53,500	—	—	—	—	—	—		10
19,800	22,600	28,500	39,200	43,500	53,500	—	—	—	—	—		15
17,000	19,400	24,500	33,900	37,000	45,500	59,000	—	—	—	—		20
13,650	15,500	19,850	27,500	30,000	36,500	48,000	—	—	—	—		30
11,700	13,300	17,000	23,400	25,650	31,200	40,900	52,500	—	—	—		40
10,500	11,800	15,000	21,000	23,000	28,000	36,100	46,800	—	—	—		50
9,500	10,750	13,500	18,900	20,800	25,150	33,000	42,000	57,000	—	—		60
8,700	9,850	12,500	17,150	18,900	23,250	30,000	38,000	52,500	—	—		70
8,200	9,250	11,750	16,100	17,750	22,000	28,500	36,000	49,000	—	—		80
7,550	8,700	11,000	15,000	16,700	20,500	26,500	33,900	46,000	58,000	—		90
7,100	8,200	10,400	14,100	15,600	19,500	25,000	32,000	43,000	56,000	—		100
6,600	7,450	9,500	13,000	14,200	17,700	23,100	29,100	39,200	51,000	—		120
5,800	6,650	8,500	11,600	12,700	15,600	20,500	26,000	34,800	45,000	57,500		150
5,000	5,700	7,150	10,000	10,950	13,400	17,600	22,700	29,950	38,500	49,500	10 Ft. per Second.	200
4,500	5,250	6,500	9,000	9,800	12,000	15,700	20,100	27,000	34,400	45,000		250
4,050	4,600	5,800	8,000	8,750	10,800	14,000	18,000	24,100	31,000	40,000	8	300
3,650	4,200	5,350	7,300	8,000	9,800	12,850	16,400	22,200	28,500	36,000		350
3,400	3,900	5,000	6,800	7,500	9,250	12,000	15,300	20,900	26,600	34,000		400
3,220	3,650	4,700	6,450	7,000	8,700	11,400	14,300	19,600	25,000	32,000		450
3,100	3,450	4,450	6,100	6,650	8,250	10,800	13,600	19,000	23,700	30,100		500
2,900	3,300	4,200	5,800	6,300	7,800	10,150	13,000	17,500	22,700	28,800		550
2,800	3,180	4,000	5,600	6,100	7,400	9,650	12,400	16,700	21,800	27,800		600
2,680	3,100	3,805	5,300	5,750	7,100	9,400	11,950	16,000	20,900	26,500		650
2,590	2,900	3,660	5,200	5,600	6,800	8,900	11,400	15,250	19,900	25,400		700
2,400	2,720	3,410	4,780	5,200	6,350	8,350	10,700	14,100	18,600	23,800	5	800
2,285	2,590	3,220	4,500	4,900	6,000	7,700	10,050	13,400	17,400	22,600		900
2,150	2,410	3,040	4,210	4,600	5,650	7,350	9,450	12,650	16,300	21,100		1,000
1,940	2,200	2,765	3,800	4,200	5,150	6,650	8,600	11,500	14,800	19,100		1,200
1,710	1,965	2,480	3,400	3,680	4,590	5,950	7,600	10,250	13,150	17,000	4	1,500
1,485	1,690	2,150	2,920	3,200	3,900	5,150	6,600	8,800	11,400	14,650	3	2,000
1,200	1,350	1,710	2,390	2,600	3,190	4,180	5,350	7,000	9,200	11,800		3,000
1,025	1,165	1,450	2,030	2,215	2,710	3,500	4,550	6,100	7,800	10,000	2	4,000
910	1,040	1,300	1,800	1,980	2,410	3,180	4,050	5,450	6,900	9,000		5,000
830	940	1,190	1,640	1,790	2,210	2,850	3,650	4,900	6,350	8,200		6,000
755	860	1,100	1,500	1,625	2,030	2,630	3,350	4,570	5,800	7,450		7,000
705	805	1,020	1,400	1,525	1,900	2,480	3,180	4,300	5,500	6,695	Velocity	8,000
665	755	960	1,320	1,425	1,790	2,340	2,980	3,990	5,150	6,000		9,000
640	720	915	1,270	1,380	1,700	2,240	2,820	3,800	4,900	5,350		10,000

Velocity 1 Ft. per Second

in Gallons per Minute.

Flow Through Blown-out Joints.

(B.S., Plain, S.S.)

Diameter of Main. Inches.	Increase of Velocity in Main. Feet per Second.		
	100 Feet Head.	200 Feet Head.	300 Feet Head.
4	19	—	—
6	13	18	21
9	8	12	14
12	6	9	11
18	4	6	7
24	3	4	5
36	2	3	3·5

N.B.—If the spigot falls from the concentric position in the socket, the velocity increase and consequently the leakage may be as much as 100 per cent. greater than the figures given in the table.

(*Glenfield & Kennedy, Ltd.*)

Miscellaneous Losses of Head.

Entrance Losses.—When water flows into a pipe from an open reservoir there is a loss of head (H, feet) proportional to the velocity (V, feet per second) in the pipe:—

$$H = \frac{CV^2}{2g}$$

Where C has the following approximate values:—

End of pipe flush with reservoir wall . .	$C = 0·50$
Pipe projecting into reservoir . . .	$= 0·93$
Bellmouth entrance, average . . .	$= 0·10$

Loss at Sudden Expansion (H, feet).

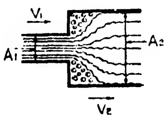


FIG. 28

$$H = \left(1 - \frac{A_1}{A_2}\right)\frac{V_1^2}{2g}$$

or

$$\left(\frac{A_2}{A_1} - 1\right)\frac{V_2^2}{2g}$$

Loss at Sudden Contraction (H, feet).

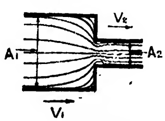


FIG. 29.

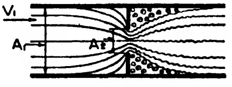
$$H = \frac{CV_2^2}{2g}$$

Where C varies with the ratio $\frac{A_2}{A_1}$

Values of C (Weisbach.)

$\frac{A_2}{A_1}$	0·10	0·20	0·30	0·40	0·50	0·60	0·70	0·80	0·90
C	0·362	0·338	0·308	0·276	0·221	0·164	0·105	0·053	0·015

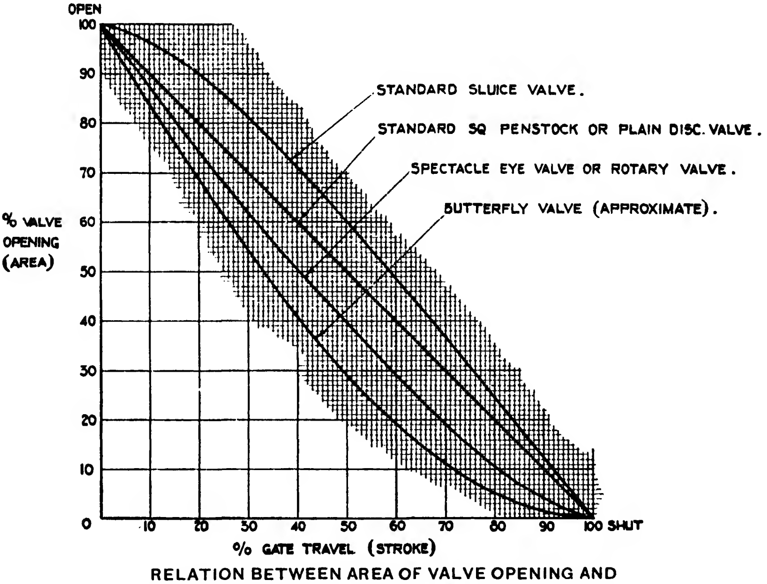
LOSS DUE TO OBSTRUCTION (H FEET).



$$H = \frac{CV_1^2}{2g}$$

FIG. 30.

The above expression can be used for calculating *approximately* the loss of head in sluice valves and other valves at any degree of opening. Curves are given below showing the relation between the area of valve opening and the stroke of valve for sluice valves, penstocks, disc valves, spectacle eye valves, rotary valves, and butterfly valves.



RELATION BETWEEN AREA OF VALVE OPENING AND STROKE OF VALVE FOR VARIOUS STANDARD TYPES OF VALVE

FIG. 31.

Values of C (Bellasis.)

$\frac{A_2}{A_1}$	0·10	0·20	0·30	0·40	0·50	0·60	0·70	0·80	0·90
C	226	48	18	7·8	3·8	2·1	0·80	0·29	0·06

'LENGTH OF PIPING EQUIVALENTS' OF PIPE BENDS.

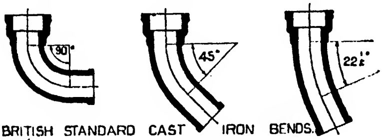


FIG. 32.

Size of Main. Inches.	Angle of Bends. 90°	45°	22½°
2	3·0	2·5	2·0
3	4·5	3·0	2·5
4	5·5	4·0	3·5
6	8·5	7·0	6·0
8	12·0	10·0	9·0
9	15·0	12·5	11·0
10	16·5	14·0	12·0
12	19·0	16·0	14·0
14	24·5	21·0	19·0
15	28·0	24·0	21·0
18	36·0	27·0	23·0

(Equivalents in feet.)

'LENGTH OF PIPING EQUIVALENTS' OF SLUICE VALVES.

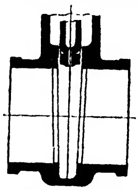


FIG. 33.

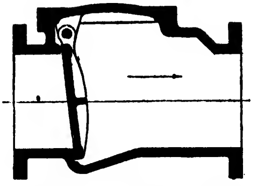
Diameter of Sluice Valve, Inches (*a*).

Length Equivalent, Feet (*b*).

a	2	3	4	6	8	9	10	12	14	15	18
b	1·5	2·0	3·0	5·0	7·0	8·0	9·0	11·0	13·0	14·0	17·0

(Glenfield & Kennedy, Ltd.)

Loss of Head in Ordinary Hinged Door Reflux Valves.



ORDINARY HINGED DOOR REFLUX VALVE — FLOW HORIZONTAL

Fig. 34.

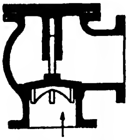
Velocity of Flow. Feet per Second.	Loss of Head in Reflux Valve. Feet.
2	0·24 to 0·35
3	0·29 to 0·38
4	0·33 to 0·39
5	0·36 to 0·42
6	0·41 to 0·46

Losses in flap type Reflux Valves cannot be estimated accurately in the form of a length equivalent. Approximate losses, which are practically independent of the size of valve within the range 1½ ins. to 18 ins., are given in the above table. (*Glenfield & Kennedy, Ltd.*)

Loss of Head in Check Valves at Various Rates of Flow.

(Approximate.)

Right Angle Type with Disc Valve. (*Glenfield & Kennedy, Ltd.*)



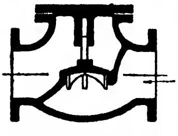
RIGHT ANGLE CHECK VALVE.

Fig. 35.

Flow G.P.M.	Diameter of Valve in Inches. 2	3	4	6
50	4	1·5	—	—
100	12	3·0	2	1·0
200	—	8·0	3	1·25
300	—	18·0	5	1·5
400	—	—	8	2·25
500	—	—	13	3·0
600	—	—	18	3·75
700	—	—	—	4·75
800	—	—	—	6·0

Losses in feet head of water.

Straight Through Type with Disc Valve.



STRAIGHT THROUGH CHECK VALVE.

Fig. 36.

Flow G.M.P.	Diameter of Valve in Inches. 2	3	4	6
50	8	2·5	—	—
100	30	6·0	2·5	1·5
200	—	18·5	6·0	2·0
300	—	—	12·0	3·0
400	—	—	21·0	4·0
500	—	—	—	6·0
600	—	—	—	8·0
700	—	—	—	11·0
800	—	—	—	14·0

Losses in feet head of water.

Fluid Flow through Beds of Granular Material.

Three papers presented to the Institution of Mechanical Engineers by Dr. H. E. Rose and a fourth paper by Dr. H. E. Rose and Dr. A. M. A. Rizk deal with fluid flow through beds of granular material.

In the first paper (1945, *Proc.I.Mech.E.*, Vol. 153, p. 141) the general equation of flow of an incompressible fluid through a bed of granular material is derived by the method of dimensional analysis, this equation being, in the notation given in the paper.

$$\left(\frac{H}{d}\right) = \phi\left[\left(\frac{vdp}{\mu}\right)\left(\frac{dg}{v^2}\right)\left(\frac{h}{d}\right)\left(\frac{D}{d}\right)\left(\frac{e}{d}\right)(f)(Z)(U)\right]$$

By use of beds composed of closely graded, smooth, spherical particles, the effects of variation of the dimensionless parameters (Z), (U), and $\left(\frac{e}{d}\right)$ are eliminated and the relationships between $\left(\frac{H}{d}\right)$ and $\left(\frac{vdp}{\mu}\right)$, $\left(\frac{dg}{v^2}\right)$, $\left(\frac{h}{d}\right)$, and (f) are investigated experimentally

From these tests it is deduced that $\left(\frac{H}{d}\right) \alpha \left(\frac{v^2}{dg}\right)$ and $\left(\frac{H}{d}\right) \alpha \left(\frac{h}{d}\right)$; also, from the author's experimental results augmented by those of other investigators a curve is derived relating the resistance coefficient, ψ, to the Reynolds number defining the flow.

A curve relating the resistance of a bed to the bed porosity is derived in a manner believed to be original.

A curve relating the resistance of a bed to fluid flow and the ratio (D/d) is also derived and verified experimentally to a very limited extent; it is indicated that the connexion probably depends also upon the Reynolds number defining the flow.

The second paper (1945, *Proc.I.Mech.E.*, Vol. 153, p. 148) commences with the extension, from first principles, of the general equation (above) to cover the case of the isothermal flow of a compressible fluid through a bed of granular material.

This extended equation is

$$\frac{gRT}{v^2P^2}(P_1^2 - P_0^2) = 2\,\psi\left(\frac{h}{d}\right)\phi_1\left(\frac{D}{d}\right)\phi_2(f) + log_e\frac{P_1}{P_2}$$

and it is shown that for all normal flow conditions the term $log_e\frac{P_1}{P_0}$ can be neglected. In this equation the functions $\psi_1\,\phi$, (D/d), and $\phi_2\,(f)$ are identical with those for the flow of incompressible fluids.

The validity of this equation is demonstrated by passing air, oxygen, hydrogen, and acetylene through beds of almost spherical sand; for these gases the values of 2,940, 2,630, 42,000 and 3,240 for the gas constant, R, and 0·145, 0·145, 1·045, and 0·085 for the kinematic viscosity, respectively, are obtained. Although the results are not regarded as conclusive the agreements lend support to the view that a single basic equation is applicable to the flow of compressible and incompressible fluids.

The third paper (1945, *Proc.I.Mech.E.*, Vol. 153, p. 154) deals exclusively with the resistance coefficient—Reynolds number relationship for beds of spherical and nearly spherical materials, the relationship being derived from the investigations of the author and of many workers.

As a result of that analysis, it is deduced that the most probable value of the resistance coefficient is given by the empirical expression

$\psi = \frac{1{,}000}{R_e} + \frac{60}{\sqrt{R_e}} + 12$, and that within the transition range between streamline and turbulent flow a very wide margin of uncertainty in the value of ψ exists. For many practical problems, the flow conditions would fall within this transition range and, in view of the uncertainty, a statistical method of dealing with the design problem is suggested.

In the fourth paper (1948, *Proc.I.Mech.E.*,) a more complete curve of wall effect is deduced and the results of the previous papers are considerably extended to cover the cases of beds composed of materials differing widely from the spherical form and also of beds composed of non-spherical materials of mixed shapes and sizes.

For further information reference should be made to the orginal papers.

Fluid Flow through Filtering Media.

A paper by the author ('A New Measure of the Filtrability of Fluids with Applications to Water Engineering,' *Journal of the Institution of Civil Engineers*, February 1947, p. 415) deals with the flow of fluids through filtering media and with the effect on flow of the progressive blockage of the media produced by the solids arrested by the media.

A solution is proposed of certain aspects of the filter blockage problem, based on the conception of the filtrability of fluids, and a mathematical law, derived empirically, is expounded, by which a number of practical filter flow problems become susceptible of calculation.

A new measure of filtrability is defined and methods of measurement described.

For further information reference should be made to the original paper.

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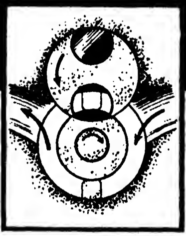

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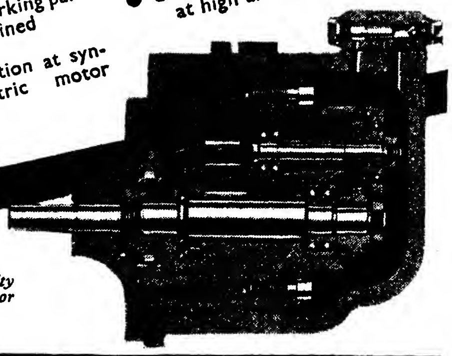

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SECTION XIX

PART II

HYDRAULIC TRANSMISSION OF POWER.

(Contributed by P. L. Boucher, Ph.D. (Lond.), A.M.I.C.E.)

Hydraulic Transmission of Power.

A remarkable static period in the development of hydraulic power has for some time been passed. Although direct application of electric power has rendered obsolete certain types of hydraulic apparatus (hydraulic lifts and hoists, for example), hydraulic power now more than holds its own in some of its original fields and has, in addition, found wide scope. Hydraulic power has been of very great importance in the war effort. The essential simplicity and convenience of hydraulic machinery and the very large positive and flexible efforts at high efficiency which can be exerted against the heaviest loads, are among advantages responsible for the favour with which engineers have already regarded orthodox plant of this type, whilst many modern developments are due to the progressive improvements of materials of construction, and of design, which have enabled working pressures and operating speeds to be raised to figures formerly deemed unobtainable in practice. In addition, highly developed automotive apparatus has extended the use of small scale applications of hydraulic effort by the use of standardised units as in the case of hydraulic automobile brakes, aircraft controls and similar devices.

Working Pressures.

In the past a pressure of 750 lb. per sq. in. was adopted as standard for the hydraulic transmission of power. Now, for heavy work, pressures between 1,500 and 3,500 lb. per sq. in. are normal, whilst pressures of 5,000 and 6,000 lb. per sq. in. are not unusual. Some hydraulic machines with individual pump units work at pressures up to 10,000 lb. per sq. in. Manufacturers of such machinery have facilities for testing components at double these working pressures and pressure gauges indicating up to 70,000 lb. per sq. in. are available.

Pressures within this range give an entirely new significance to this branch of engineering and to its economics, and stimulate the design and production of machinery and controls to rigorous standards essential for such heavy duties. Water itself, usually considered as incompressible for practical purposes, behaves under a pressure of 5,000 lb. per sq. in. as an appreciably elastic fluid, a cub. ft. at atmospheric pressure contracting by nearly 30 cub. ins. with strain energy of some 6,000 ft./lb.* Oils are usually employed as hydraulic media for the highest pressures and for small scale apparatus.

Hydraulic Machines.

There are very few branches of engineering to which hydraulic power is not applicable, and in many different types of machines and apparatus, efforts of from a few ounces to thousands of tons are transmitted hydraulically.

Hydraulic machinery can be classified in several ways, but fundamentally falls into two categories, which include (1) machinery for the provision of power—pumps, accumulators, intensifiers, control valves and pipe work; and (2) machinery for the utilisation of power—presses, actuating machinery, handling plant, machine tools, and automotive mechanisms. Some modern examples of machinery in these categories are described, in this order, below.

Provision of Hydraulic Power.

Pumps.—(See also Section XIX, Part III, p. 801.)

Pumps for the provision of hydraulic power fall into three classes: (1) Slow speed multiple ram pumps which very often, although not always, work in conjunction with accumulator systems; (2) small high speed plunger pumps used for the individual powering of hydraulic machines; and (3) other devices for transmitting hydraulic effort including those for small scale apparatus.

---

* See p. 744, for values of Bulk Modulus K, for water.

Pump Capacity.—In a system without accumulators, the pumps must be capable of delivering the maximum requirements, both in volume pumped and in pressure generated. In an accumulator system the pumps must be capable of delivering at least the average requirements of the system, the accumulators being designed to meet the maximum demand.

Treble Ram Pumps.—The slow speed ram pump is typified by the widely used treble ram machine, which, in the largest sizes, is invariably of horizontal pattern, although for smaller installations vertical designs are employed. Ram pumps are usually driven by electric motors coupled through flexible drives or through heavy reduction gearing. There is considerable variation in ram speeds which are very often less than 200 ft. per min., although speeds of over 500 ft. per min. are used on carefully balanced pumps. Crankshaft speeds of more than 100 r.p.m. are unusual and would, for ordinary designs (see table below), require mechanically operated valves. For hydraulic purposes pumps are usually run at constant speed.

Slip allowances depend upon the size of the pump, being greatest for the smaller machines. An average figure for volumetric efficiency may be taken between 90 and 95 per cent. Mechanical efficiency, exclusive of driving motor, is usually between 75 and 85 per cent.

The following table gives particulars of some representative examples of recent treble ram pump construction.

SOME TREBLE RAM PUMPS OF RECENT MANUFACTURE.

Diameter. Ins.	Stroke. Ins.	Speed. R.P.M.	Ram Speed. Ft./Min.	Delivery. G.P.M.	Pressure. Lb./sq. in.	Motor Power. B.H.P.
4¼	15	60	150	135	1,500	185
4 9/16	15	60	150	150	1,750	250
4½	18	60	180	167	2,240	350
3¾	18	60	180	120	2,500	—
2 11/16	12	65	130	45	3,500	150
2¼	9	56	74	20	5,000	—
2	10	62	103	25	6,000	100
7½	36	84	504	1,432	5,600	2,500

The last machine in the above list is of particular interest as, not only is it an exceptionally large pump, but it works at high pressure and runs at relatively high speed. Unlike the others in the list, it is not associated with an hydraulic accumulator, but has instead two flywheels which assist the 2,500 b.h.p. electric driving motor to meet a peak load of well over 6,000 h.p. This pump, which is totally enclosed with forced lubrication and balanced rotating parts, provides hydraulic power for the operation of a 7,000 ton forging press, one of the largest in the world, in which a single pump provides directly the power for the main forging ram.*

The treble ram pump, running at 60 to 70 r.p.m., and delivering between 100 and 200 g.p.m. at pressures between 1,500 and 5,000 lb. per sq. in., is typical of many recent pump-accumulator installations for hydraulic power supply. In some cases intensifiers are used for the production of finishing pressures, the pump pressure being used directly for the first effort. In other cases the pump is installed in conjunction with a low pressure supply used for filling and initial efforts.

In addition to treble ram pumps, however, there are in use other multiple ram pumps, examples of which are available having 2, 4, 7 and 12 rams to each machine.

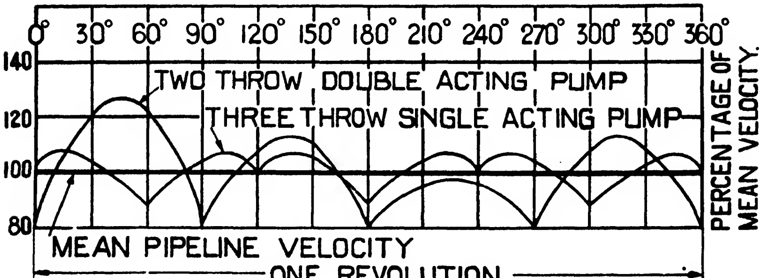

FIG. 1.

* *Engineering*, February 15, 1935, p. 161.

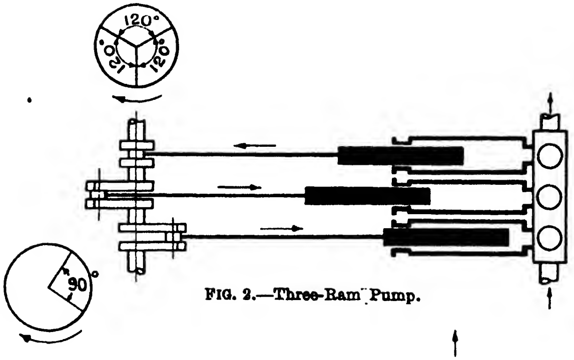

FIG. 2.—Three-Ram Pump.

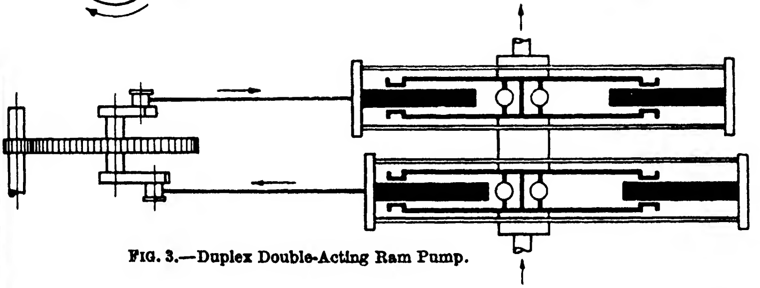

FIG. 3.—Duplex Double-Acting Ram Pump.

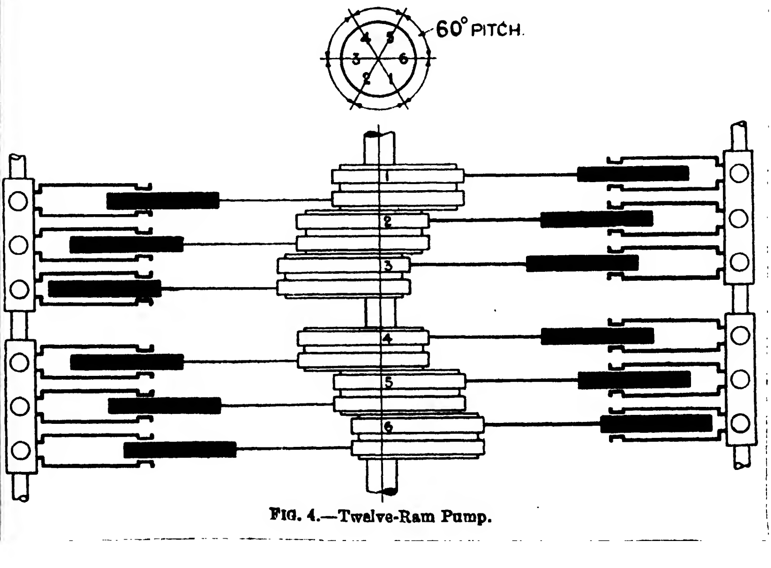

FIG. 4.—Twelve-Ram Pump.

Duplex Double-Acting Ram Pumps.

A type of large slow speed ram pump, suitable for the provision of hydraulic power, is the duplex double-acting machine of which a number have been constructed recently. Details are available * of four machines, not actually used for power purposes, but for pumping oil at pressures of 1,000 lb. per sq. in. These pumps have two separate pumping units driven by a common crankshaft, each unit having two separate rams in line, connected together by crossheads and long rods. The four rams, each approximately 5 ins. diam. with 24-in. strokes, provide, at a crankshaft speed of 40 r.p.m., a capacity of 276 g.p.m. Each pump is driven by a 330 b.h.p. gas engine.

Dual Pressure Multiple Plunger Pumps.

A typical machine of this class, of which a number have been constructed, has 12 rams arranged in groups of three, symmetrically about a common driving shaft. Four of the rams provide a pressure of 2 tons per sq. in., and the 8 other rams a pressure of 0·6 ton per sq. in. The rams are all 2 ins. diam., have strokes of 6 ins., and are driven by eccentrics on the main shaft, which rotates at 80 r.p.m. The deliveries of pressure water are 21 g.p.m. and 42 g.p.m., in relation to the high and lower pressures respectively, and bypass valves are fitted to permit continuous working.

Development of the Conventional Multiple Ram Pump.

Intermediate between the large slow speed multiple ram pumps described above, and the small high speed pumps described below, is a type developed from the conventional ram pump, but arranged so that remarkably high crankshaft speeds are obtained. One example has 7 rams driven by a 7-throw crankshaft rotating at 600 r.p.m. The valves are not mechanically operated, but the slip difficulty at high speed is overcome by the use of small multiple valves which, in the example quoted, are $\frac{3}{16}$-in. diam., with lifts of $\frac{1}{32}$ in.

High Speed Pumps.

For certain applications of hydraulic power, the self-contained hydraulic machine has definite advantages. The development of this type of machine had been made possible by the production of high speed pumps which are very compact, although capable of developing pressures up to 5,000 or even 10,000 lb. per sq. in. Such a pump has been described † working at 5,000 lb. per sq. in., for which a volumetric efficiency of 97 per cent. is claimed at 1,500 r.p.m. High volumetric efficiency is essential to obviate vibration and noise at such speeds, and to reduce flow pulsations.

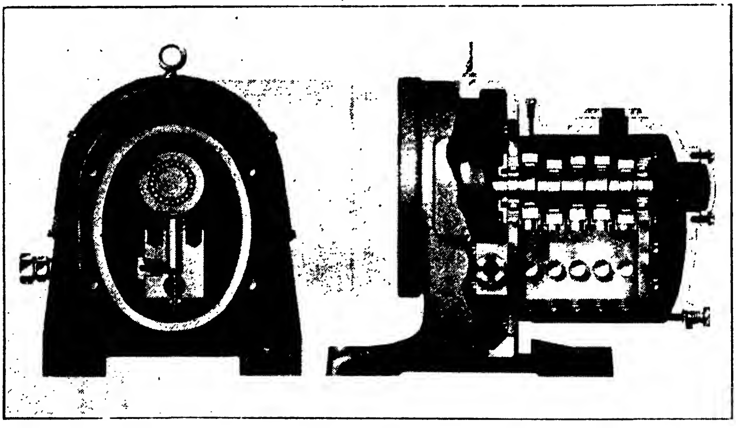

Fig. 5.—Five-Throw Unit-Mounted Pump (*Towler*).

* *The Engineer*, March 22, 1940, p. 227.
† 'Recent Developments in High Speed Reciprocating Pumps,' F. H. and J. M. Towler, *Proc. I. Mech. E.*, Vol. 137, 1937, p. 79.

High speed reciprocating pumps, in which oil is used as the hydraulic medium, are designed to give constant pressure and constant delivery at a constant speed, or to give variable delivery and inversely variable pressure at constant speed. Two-pressure pumps of this type are also made to deal with the idle and working portions of the stroke of the hydraulic machines with which they are associated.

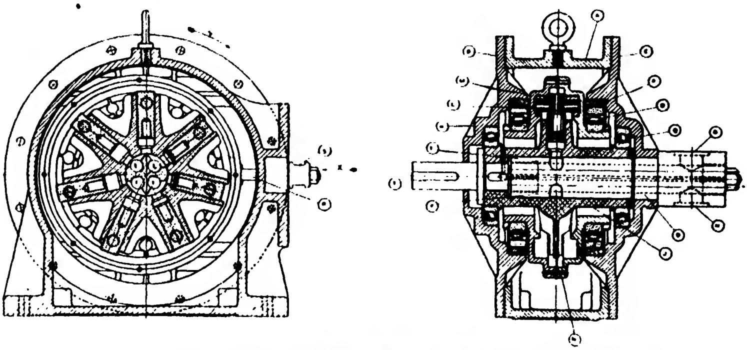

FIG. 6.—Radial Type High Speed Variable Delivery Pump (*Hele-Shaw*).

The rams may operate in barrels located in a parallel group, or the barrels may be set radially around the centre of operation. The rams are driven by eccentrics, by swashplate and connecting rods or by cams, and the valves are spring loaded. The suction valves are invariably drowned; in fact the pumps are immersed in the hydraulic medium, which is a specially prepared oil.

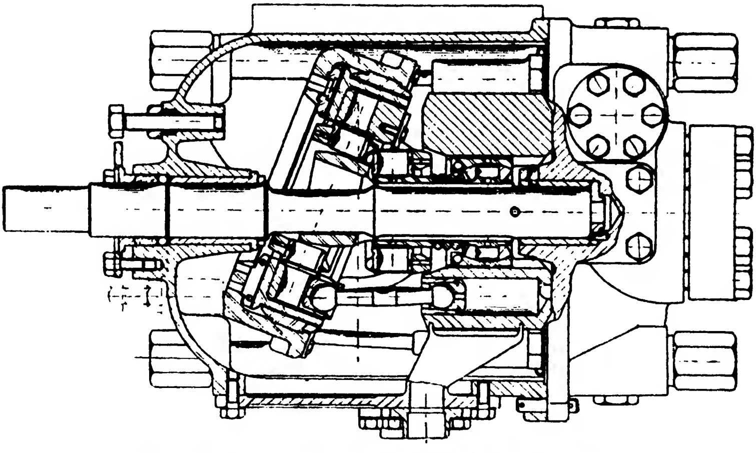

FIG. 7.—Swashplate Type High Speed Variable Delivery Pump.

These and other types of high speed pumps suitable for generating high hydraulic pressures, including rotary pumps and gear pumps of different designs, are described in 'Modern Hydraulic Operation of Machine Tools,' H. O. Town, *Proc. I.Mech.E.*, Vol. 123, 1932, p. 211.

Wave Transmission of Energy.

It is possible to supply energy in the form of longitudinal vibrations, to a fluid column enclosed in a pipe line, and to transmit the energy through the column in such a way that the reciprocating motion of a plunger initiating the vibrations at one end, is reproduced by a plunger at the other end, without flow or circulation of the fluid through the pipe line. Constantinesco has patented a number of applications of this method which have been used in practice. A 3-phase system with three pipe lines, analogous to an electrical 3-phase alternating current system, is usually employed. A 3-cylinder generator with cranks at 120° initiates elastic vibrations or pressure waves in the fluid contained in the pipe lines, and these vibrations are received by the pistons of a 3-cylinder hydraulic motor having the same crank angles. Pressure within the system is maintained by a separate pump which compensates for any fluid slip past the pistons.

An outline of the theory of wave transmission of energy, due to Dr. H. Moss, is given in 'Mechanical Properties of Fluids,' 2nd Ed., 1936, Blackie & Sons, Ltd., London. See also *Proc. I.Mech.E.*, 1923, and Constantinesco's original paper 'The Theory of Sonics,' (The Proprietors of Patents Controlling Wave Transmission, 132 Salisbury Square, E.C.4. 1920).

Hydraulic Accumulators.

High pressure accumulators are of weight loaded and air loaded types, whilst low pressure accumulators are also air loaded, although of different design from the high pressure type.

Accumulator Capacity.—Although an accumulator is installed to provide capacity, this provision must, on account of the large dimensions and heavy loads involved, be kept to the very minimum. The function of the accumulator is to store up excess water delivered by the pumps at times of reduced demand and to give out the water stored when the demand exceeds the pump capacity. If the total amounts of water pumped and used are both plotted against time, a chart of the following type is obtained:—

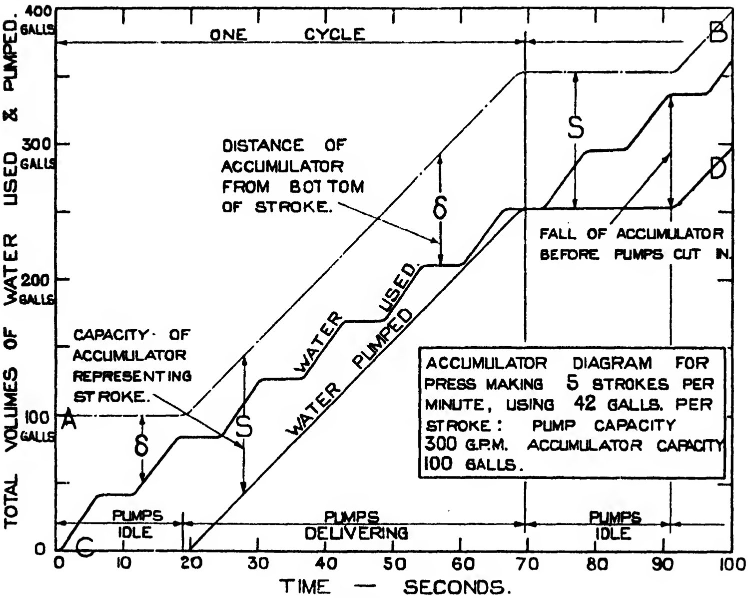

Fig. 8.

The line AB is drawn parallel to the line CD representing the total water pumped at any time so that the ordinate distance S between these two lines represents the stroke of the accumulator. The ordinate distance δ between the line AB and the curve of total water used at any time then represents the distance of the accumulator from the bottom of its stroke, to the same scale that S represents the stroke. When the demand curve drops to the pump delivery line, the accumulator is full and the pumps are stopped or running unloaded, and they do not begin to deliver again until the demand curve has left the line OD. If the demand curve should go beyond the line AB, the accumulator capacity would be too small. From this diagram the rate of ascent and descent of the accumulator can easily be plotted.

Weight Loaded Accumulator.—The weight-loaded accumulator is the classic design but very large examples have been installed in recent years, and important units are under construction.

The weight loaded accumulator consists of a long ram operating vertically in a cylinder. From the head of the ram a crosshead supports a load of cast iron or concrete weights, or a container filled with ballast,* the total load being sufficient to balance the pressure of the fluid pumped into the cylinder. An automatic valve cuts off the supply of pressure fluid towards the end of the upward stroke of the ram, to ensure that the ram shall not be driven out of the cylinder. In many designs there is also a mechanical safeguard in the form of a bayonet joint. Another valve automatically throttles the discharge from the accumulator cylinder towards the end of the downward stroke, in order that the accumulator and weights may come to rest without serious shock.

Rate of Descent.—The relation between ram diameter and stroke is a question of economics. It follows that ram diameters are kept small and long strokes are used. A limiting factor is, however, the maximum permissible rate of descent which must not be such that the momentum of the deadweight load reaches a dangerous value. In many of the largest installations the rate of descent of weight loaded accumulators is kept to less than 1 ft. per sec., although there are large plants in which 2 or even 3 ft. per sec. are allowed. At these higher rates of descent, damaging pressure surges can be set up if the power water being drawn from the accumulator is cut off suddenly.

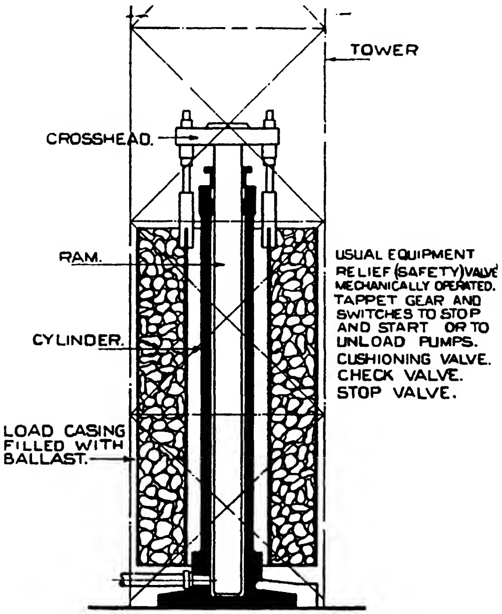

Fig. 9.—Diagram of Weight Loaded Accumulator.

* In a recent case where the pig iron would normally have been used as ballast, for the sake of economy of metal, barytes (barium sulphate) of S.G. 4·0 to 4·8 was used instead.

The table below gives details of some modern weight loaded accumulators, and forms a guide to good practice.

Weight Loaded Accumulators of Recent Manufacture.

Diam. Ins.	Stroke. Ft.	Stroke. Ins.	Pressure. Lb./sq.in.	Capacity. Gallons.	Load. Tons.	Energy Stored. Ft./lb.	Maximum Output H.P. at 1 ft./sec.	Situation.
23½	20	0	750	376	145	6,500,000	590	Tyne Docks.
20½	35	0	800	500	117	9,200,000	475	Port of Bombay.
*24	36	0	800	705	160	13,000,000	650	Leith Harbour and Docks.
27	12	1	800	300	204	5,050,000	830	Port of Calcutta.
30	16	1¼	800	490	250	9,100,000	1,020	Port of Calcutta.
14½	17	0	1,750	122	128	4,900,000	520	Government Factory.
15½	21	6	2,240	176	188	9,100,000	770	Government Factory.
4	7	6	3,360	40	19	315,000	77	Government Factory.

(*Glenfield & Kennedy, Limited.*)

***Differential Weight Loaded Accumulators.*—If the ram of an accumulator is fixed and passes through both ends of the moving storage cylinder, the ram having two diameters, the smaller at the top, very large pressures can be sustained by comparatively small weights. The capacity of such an accumulator is relatively small.**

***Hydro-Pneumatic Accumulators.*—In these accumulators dead weight loading is dispensed with and the load is instead applied by compressed air, either directly, the air being at the same pressure as the hydraulic fluid, or through a differential piston, the air being at a lower pressure than the fluid.**

Direct air loaded accumulators consist of a series of steel air vessels, connected hydraulically and pneumatically, of sufficient combined capacity, together with that of the pump or pumps,

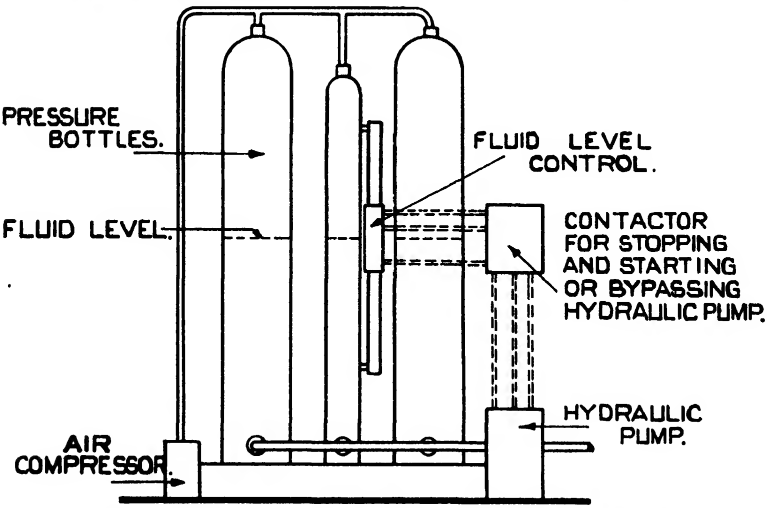

Fig. 10.—Diagram of Hydro-Pneumatic Accumulator, Direct Loaded Type.

* This accumulator is one of the largest in the world.

to supply the system. Running of the pumps is automatically controlled by the rise and fall of the fluid level, the pump being arranged either to start and stop, or to bypass, in accordance with the demand for power. An air compressor supplies air as required to maintain the working pressure. This type of accumulator is relatively light and compact, and can, therefore, be installed on upper floors. It delivers fluid without pulsation of flow at a pressure which is maintained within fine limits and the pressure can be varied as required. Because there are no moving parts except the fluid itself, under an air cushion, no dead weight inertia shocks are generated by sudden cessation of flow.

Self-contained hydro-pneumatic accumulator sets including pumps and automatic control devices, excellently suited to small installations, are manufactured as complete units.*

The differential type of air loaded accumulator is very suitable for larger installations. A good example from the latest practice has a fixed vertical ram 17¾ ins. diam., on which a 2 ft. 1 in. diam. moving cylinder works through a stroke of 13 ft. 7 ins. The upper closed end of this cylinder forms a ram, 2 ft. 10 ins. diam., passing into a 35½-in. diam. fixed solid drawn steel air vessel carried on three columns from ground level. This air vessel is connected to 4 solid drawn steel air bottles, each of 130 cub. ft. capacity, at ground level. Automatic control valves for cutting out and bypassing are included in the equipment. A capacity of 145 gallons at a pressure of 3,000 lb. per sq. in. is obtained, within limits of $\pm$ 5 per cent., the air pressure being 750 lb. per sq. in. The total weight of this accumulator is about 27 tons and its overall height 38 ft.

Similar accumulators of the differential air loaded type can be designed for any pressure and capacity required. It is worthy of note that there is a limit of size to which the solid drawn steel air vessels, which are used in accordance with Board of Trade requirements, can be made. For this reason, in the larger plants, groups of air bottles are used to obtain the required total air storage capacity.

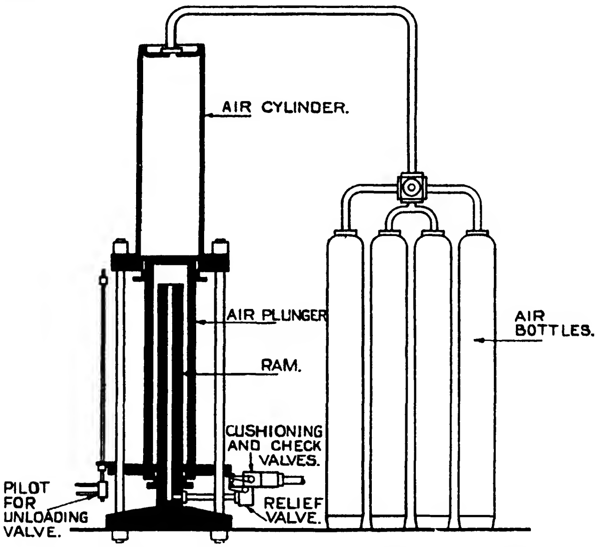

FIG. 11.—Diagram of Hydro-Pneumatic Accumulator, Differential Type.

Low Pressure Air Loaded Accumulators.—Many hydraulic machines, presses in particular, require low pressure water to bring the moving parts of the machine up to the job, to bring the halves of a mould together, etc. Pressures of from 150 to 300 lb. per sq. in. are usually suitable for this work and are obtained from either reciprocating or centrifugal pumps. Standard pressure storage sets are used in which the pump delivers water into a horizontal cylindrical tank, perhaps 6 ft. diam. × 30 ft. long, in which an air cushion is provided.

* *Engineering*, May 20, 1938, p. 574.

Steam Accumulators.—Similar in principle to the differential air loaded accumulator, the steam accumulator employs available steam pressure to load a hydraulic ram through the medium of a suitably proportioned cylinder and piston. In addition, if the steam supply to the pumps is taken through the accumulator steam cylinder, automatic pump control at the limits of the accumulator stròke is readily obtained. Steam accumulators have special application to hydraulic installations where weight loaded accumulators would be unsuitable, as on board ship or in the case of very large plants. In recent years some very large steam hydraulic accumulators have been made for land use. One example has double units with steam cylinders 29½ ins. diam. by 13 ft. 9 ins. stroke, supplied with steam at 169 lb. per sq. in. and stores hydraulic energy at 5,800 lb. per sq. in. for the operation of a 15,000-ton forging press.

Hydraulic Intensifiers.—It is necessary to distinguish between the differential accumulator and the intensifier. The hydraulic intensifier is really a direct acting pump or booster, and it can be operated by water pressure or by steam pressure. Compared with the differential accumulator it is similar except that the high pressure is not provided by external pumps but by the differential loading of water that may be delivered by low pressure pumps, or drawn from mains. The energy stored in the high pressure water is obtained from the low pressure side. The relative effective areas of the rams and cylinders are chosen to suit the available low pressure and required high pressure, allowance being made for frictional losses.

A representative example of a recent hydraulically loaded intensifier has the larger ram 25 ins. diam., on which a pump pressure of 800 lb. per sq. in. acts through a stroke of 8 ft. The smaller ram, 14 ins. diam., delivers water at 2,240 lb., the relative displacements being, of course, in the ratio of the squares of the two diameters. (Not in the ratio of the two pressures, as there are appreciable frictional losses.)

Another recent hydraulic intensifier for a still higher pressure has a low pressure ram 25 ins. diam., using water at 800 lb. per sq. in., and a stroke of 6 ft. The high pressure ram is 8¾ ins. diam. and delivers water at 6,300 lb. per sq. in.

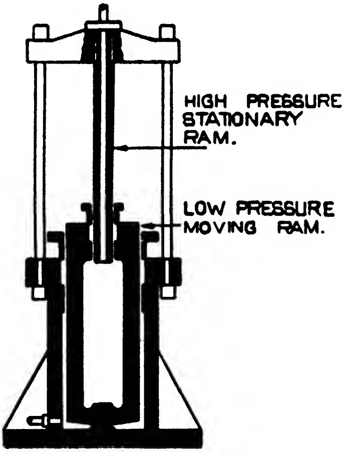

FIG. 12.—Hydraulic Intensifier.

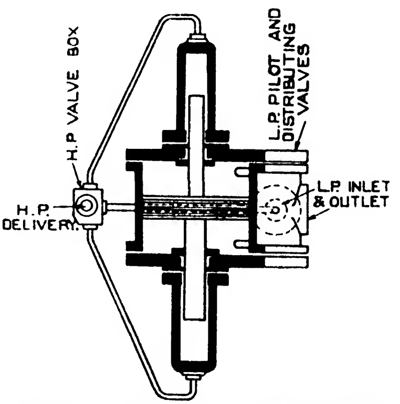

FIG. 13.—Automatic Double Acting Hydrostat Hydraulic Intensifier.

Hydrostat Intensifier.—Another form of hydraulic intensifier for small installations, particularly suited for the development of high pressures for repetition testing purposes, is found in the Hydrostat which, in this appliction, provides by automatic double action a constant supply of high pressure fluid. This machine consists essentially of a piston connected to upper and lower rams which are so proportioned that uniform delivery pressure is obtained on the upward and downward strokes. Valve gear for reversing the strokes is automatic and consists of a distributor valve, pilot operated, for low pressure water, and ordinary pump type suction and delivery valves for the high pressure water.

Control Valves.

Hydraulic control valves include manipulating valves, stop valves, check valves, relief valves, cushioning valves and bypass valves. Full advantage should be taken of modern materials of construction for valves to carry the very heavy loads imposed by the highest fluid pressures, and to withstand the erosive action of high velocity streams which may flow through the valve passages at speeds of several hundreds of ft. per sec.

Valve bodies are made of gunmetal, high tensile cast iron, cast steel or solid forged steel, according to the size and working pressure. Spindles and other internal working parts are of special alloys among which may be mentioned forged aluminium bronze, nickel steel and stainless steel. Bolts are of high tensile steel, and construction generally is of a standard commensurate with the rigorous duties of valves which control hydraulic power at heavy pressures.

Manipulating Valves.—Three main types may be defined: the piston valve, which is, of course, hydraulically balanced; the rotary valve, which is also in equilibrium, and the poppet valve, which can be hydraulically balanced for direct operation or, alternatively, may be operated by pilot action. Representative examples are illustrated by the diagrams. Servo operation, as by the Lockheed system, may be applied to large valves, valves working at high pressures, and valves which must be arranged for remote control (see p. 795).

Various multiple-way valves are used—2-way, 3-way and 4-way—and combinations of multiple-way valves of various degrees may be arranged in ganged units for operation in phase or in sequence. Check valves and bypass valves may require to be incorporated for the control of double pressure systems. Operation may be at the instigation of the operator alone, semi-automatic, or fully automatic.

By an arrangement of semi-automatic valves it is possible to control a machine with a single lever. Low pressure is admitted to the machine at the first movement of this lever and also to an auxiliary cylinder which controls the high pressure valve. When the machine is closed the low pressure rises to its maximum value, and the high pressure valve control piston, which is spring loaded and adjustable, then changes over the machine to the high pressure, check valves protecting the low pressure system.

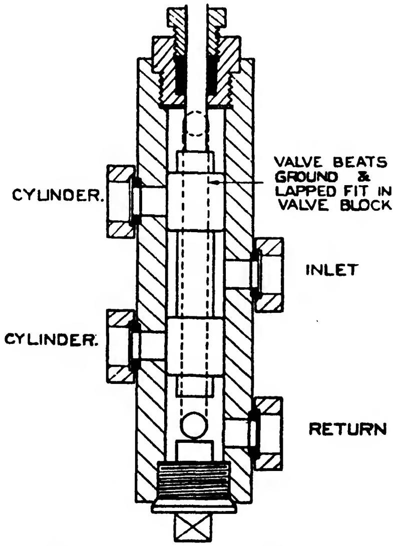

Fig. 14.—Small Two-way Piston Valve.

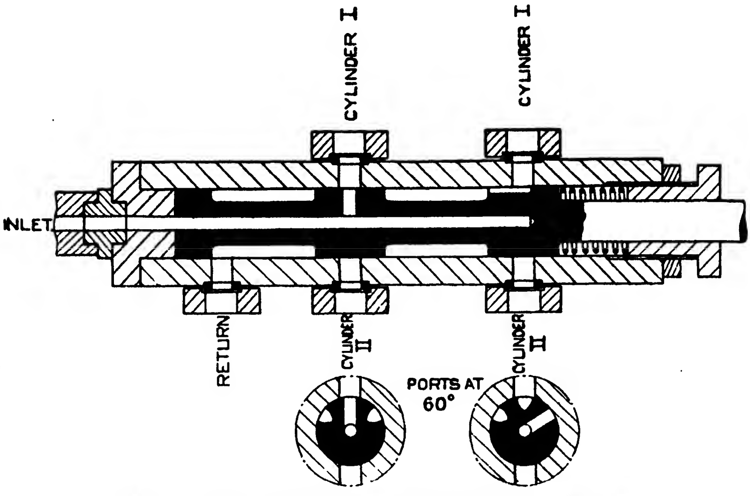

FIG. 15.—Rotary Valve for Controlling Two Cylinders in Sequence.

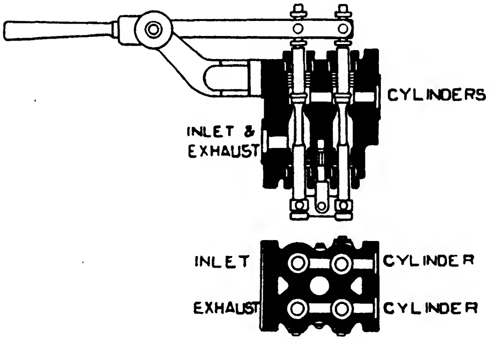

FIG. 16.—Homeyard Type Two-way Poppet Control Valve.

Fully automatic operation can be obtained by time cycle control using, say, a small synchronous motor to operate air switches, which in turn operate solenoids or diaphragms on the main hydraulic valves, in a predetermined sequence. Exact and flexible control of hydraulic machines is readily gained by such arrangements, which have many applications in hydraulic engineering.*

* See, for example : ' Hydraulic Machinery and the Plastic Industries,' M. R. Rhodes, *Proc. I.Mech.E.*, Vol. 141, Sept. 1939.

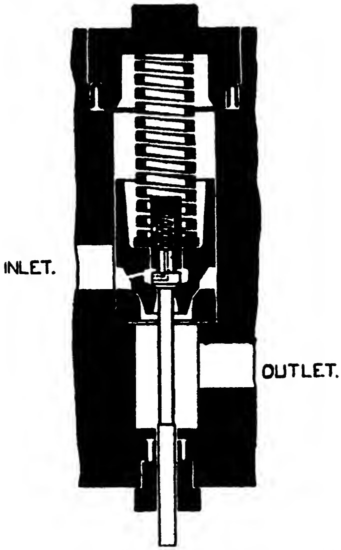

FIG. 17.—Poppet Control Valve with Pilot Action.

Stop Valves.—For pressures up to about 1,500 lb. per sq. in., gunmetal, high tensile cast iron, or cast steel, screwdown stop valves with conical metal to metal valve faces are used, the choice of metal depending upon size and pressure. The valves may be unbalanced in the smaller sizes,

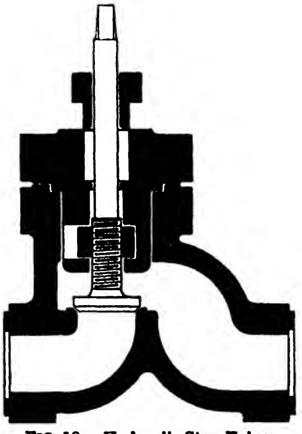

FIG. 18.—Hydraulic Stop Valve, Unbalanced Type.

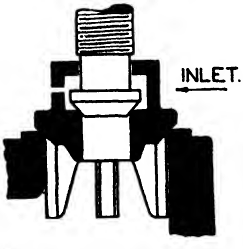

FIG. 19.—Hydraulic Stop Valve, Balanced One Way.

for controlling flow in one direction only, but in the larger sizes, pilot valves are incorporated is the main valves to make possible opening against pressure in the direction of flow. Valves balanced both ways are also used, in which the main valve is extended to form a piston workin in an upper cylinder, to which pressure is admitted by pilot action.

For pressures above 1,500 lb. per sq. in. valve bodies are usually of solid forged steel, and excep in the very smallest sizes, are balanced, the usual design being similar in principle to the poppe manipulating valve described below.

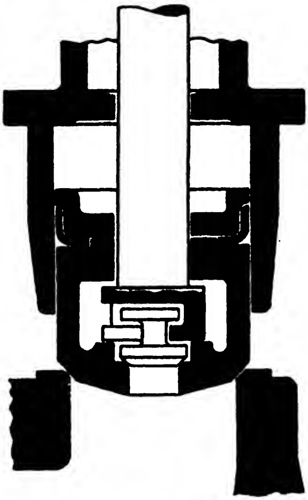

FIG. 20.—Hydraulic Stop Valve, balanced both ways.

Standard hydraulic screwdown stop valves up to 7 ins. diam. are obtainable from most makers for pressures up to 1,500 lb. per sq. in., and in sizes up to 3 ins. diam. for pressures up to about 6,500 lb. per sq. in. Larger valves, and valves for higher pressures, are usually designed to meet individual requirements.

Sluice Valves.—Sluice valves of the round body pattern are suitable for hydraulic service, to act as stop valves, and can be obtained in any size and for any working pressure.

Plug Cocks.—The pressure lubricated plug cock is another type of valve suitable for service as hydraulic stop valve. It has a parallel, non-jamming plug, is self lubricating, and has no gland or packing. Standard plug cocks of this type are available, 2-way and 3-way, in all the usual hydraulic sizes for working pressures up to 3,000 lb. per sq. in.

Check Valves.—The bodies of standard hydraulic check valves are similar to those of screw-down stop valves, and as in the case of stop valves, conical metal to metal seats are used. Most makers supply standard check valves of the same sizes and for the same working pressures as standard screwdown stop valves. Combined check and stop valves are also used.

RELIEF VALVES.

Three distinct types of pressure relief valves for hydraulic duty are in use: Two of these, the ordinary weight loaded and spring loaded valves, are well known: the third, a weight or spring loaded relief valve of differential design, is a development suited to the heaviest pressures.

In choosing a relief valve for a given duty it is necessary first to determine whether the duration of the pressure rises against which it is to mitigate is likely to be long or short; and secondly, whether the volume of water to be discharged is likely to be large or small. In general, for shock pressures, long mains will be subjected to pressure rises of greater duration than short mains, and high pressure rises will require larger discharge rates than will moderate increases of pressure. Also, the larger the pipes in use, the greater is the discharge to be allowed from relief valves. In every case the relief valve should be installed as near as possible to the origin of the disturbance, and it is safest to err on the large size in fixing the diameter.

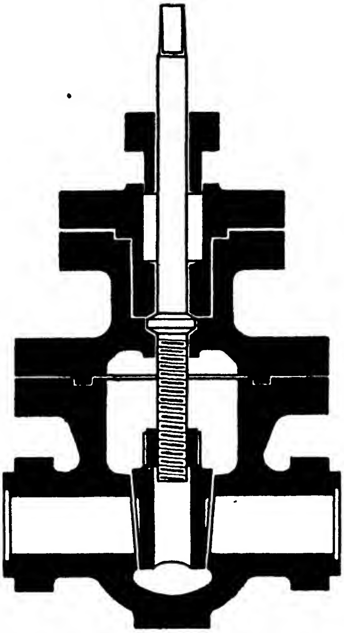

FIG. 21.—Round Body Hydraulic Sluice Valve.

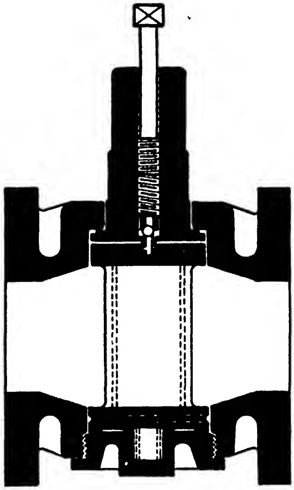

FIG. 22.—Two-Way Lubricated Plug Cock.

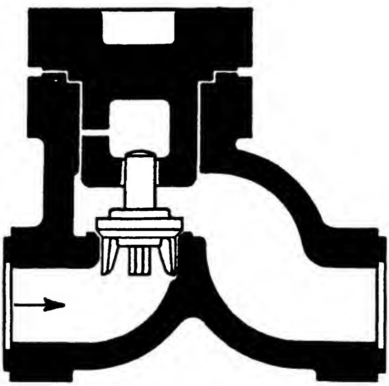

FIG. 23.—Hydraulic Check Valve.

Weight Loaded Relief Valves.—The normal pressure in the system, acting on the valve, is balanced by weights exerting their effort either directly or through a lever, and the pressure at which the valve opens depends upon the effective area of the valve, the loading, and the leverage. When the pressure rises relatively slowly, full bore opening is obtained without any sensible increase of pressure above that which opens the valve. Weight loaded relief valves are not, however, suitable for the relief of momentary increases of pressure, as the inertia of the moving parts, including valve spindles, levers and weights, is relatively high, and for quick action very much larger momentary pressures are required to open such valves than are represented by their dead weight loadings. Pressure surges of very short duration may easily occur without opening such valves.

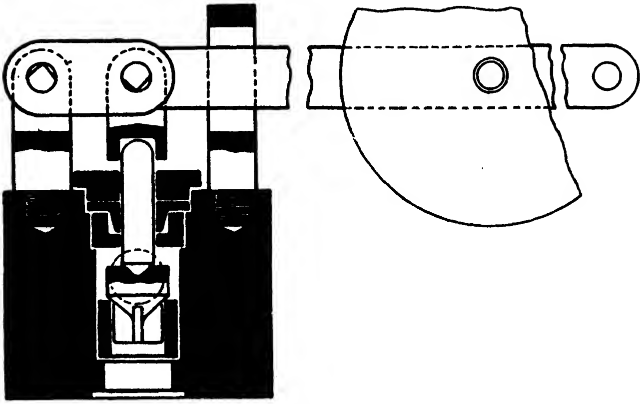

FIG. 24.—Weight Loaded Relief Valve, for High Pressures.

Spring Loaded Relief Valves.—Other things being equal, this type of valve is capable of opening much more rapidly than the weight loaded valve, because the inertia of the moving parts is relatively smaller. Spring loaded valves are therefore to be recommended for the relief of sudden rises of pressure and of pressure surges of short duration. They have the advantage of compactness. Objection is sometimes made to spring loaded valves on the ground that the blow-off pressure increases with the lift of the valve, due to the increased compression of the springs. This increase can be confined within given limits by the use of suitable springs. Discharge pressure can be appreciably affected by the shape of valve, valve ports and valve seats, and there nay be unsatisfactory action, with a tendeucy towards vibration. In well designed valves these effects are not serious.

Differential Relief Valves.—In relief valves of this type the valve element, which is either weight loaded or spring loaded, is similar to that of a poppet control valve, with plunger stem slightly less in diameter than the effective valve diameter. Consequently the total pressure load required to open the valve is very much less than that for ordinary relief valves of the same effective diameter. Thus the dead weight or spring load, which for the heaviest hydraulic pressures might otherwise be unreasonably large, is reduced to manageable proportions, and a much more sensitive relief valve is obtained.

Cushioning Valves and Bypass Valves.

The cushioning valve associated with some weight loaded accumulators is simply a throttle valve on the supply pipe, which is fully open during the rise and fall of the ram, but which is engaged by tappets when the ram approaches the bottom of its stroke, and thereafter closes during the final portion of the stroke, with the object of bringing the ram and its load to rest without shock. A check valve incorporated in the cushioning valve block allows water to flow freely *into* the accumulator.

It is unusual to maintain the delivery of the pumps while the accumulator ram remains at the top of its stroke and during the first part of its descent. The water delivered by the pumps is allowed to circulate back to the suction through a bypass valve (also referred to as unloading

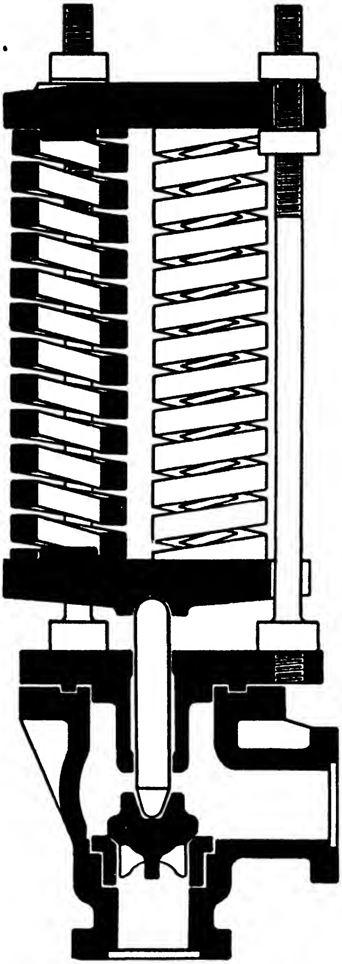

FIG. 25.—Spring Loaded Relief Valve.

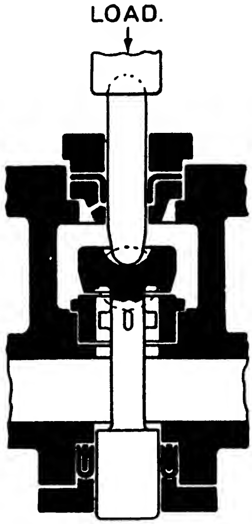

FIG. 26.—Differential Relief Valve.

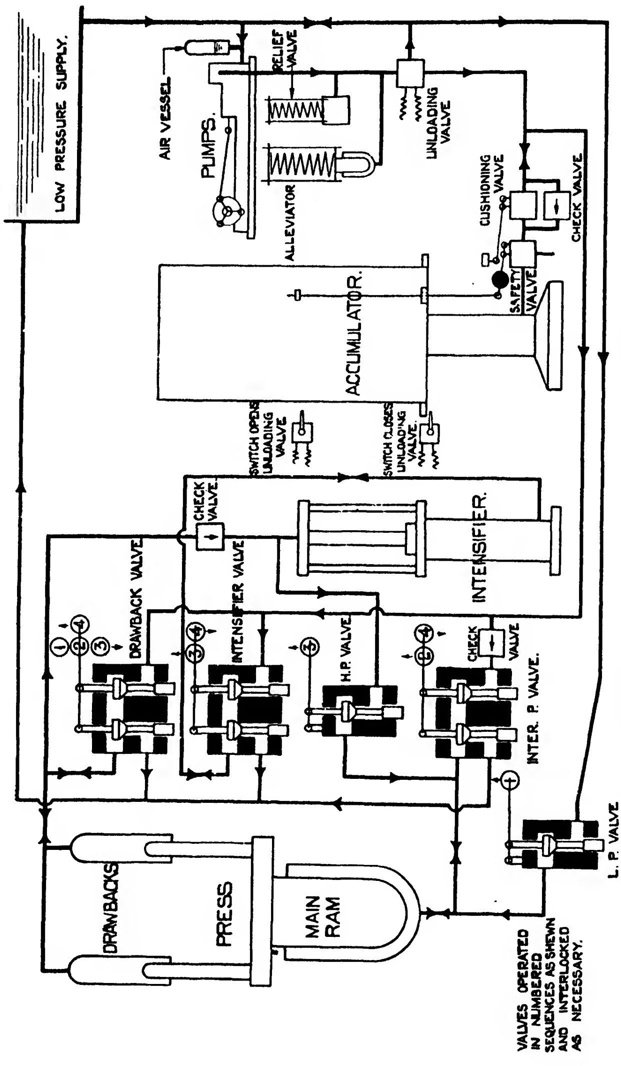

FIG. 27.—Diagrammatic Arrangement of Hydraulic Circuit for Pump—Accumulator—Intensifier Press System.

valve, churning valve or erroneously, as deflecting valve *). Recent examples are available of bypass valve controls operated mechanically and electrically. The actual bypass or unloading valve is the same in both cases, and consists of an hydraulically operated three-branch valve, two branches of which are always in communication between the pump and the accumulator. The third branch, leading back to the suction, is closed while the pumps are supplying the accumulator, but opens when the accumulator reaches the top of its stroke and remains open until the accumulator has descended. A check valve, which is often a combined stop and check valve, on the accumulator side of the bypass valve, closes automatically as soon as the bypass valve opens, and relieves the pressure on the pump side. Hydraulic operation of the bypass valve is controlled by a pilot valve which may in turn be operated either by tappets fixed to the moving parts of the accumulator or, in the case of electrical operation, by limit switches. (See fig. 27, p. 784.) Both the mechanical and electric methods of controlling bypass or unloading valves can be applied to one or more sets of pumps delivering into one accumulator. Where two or more pumps are so controlled, it is best to stagger slightly the switching points so that the pumps cut in and out of supply in sequence and not together.

Air Vessels and Alleviators.

In hydraulic power transmission air vessels, apart from those important ones associated with pneumatic accumulators, are not generally employed except on the inlet connections to ram pumps. These vessels are often referred to as 'suction' air vessels, although the inlet pressure will probably not be below that of the atmosphere. According to Richardson,† the dimensions of an air vessel should be sufficient to provide an air space (a cushion of *compressed* air) of from 5 to 6 times the displacement of one ram plunger. Air vessels of much larger capacity can well be used in all cases where appreciable cyclic fluctuations of velocity occur. They should be fitted as close to the pump as possible.

On the delivery side of hydraulic ram pumps, alleviators are often fitted and should always be fitted to pumps which have noticeable cyclic variations of discharge. Alleviators are also fitted at other points on a hydraulic system to provide elasticity, and damping action on pressure surges which may be set up by sudden variations of flow. Alleviators can be usefully associated with weight loaded accumulators which work at high rates of descent.

The hydraulic alleviator is simply a spring loaded plunger with diameter, stroke, and spring specification determined by the pressure rise which may be allowed to result from the most sudden flow variation likely to occur in a system. The alleviator plunger should be preloaded against stops to the working pressure by compression of the springs, which are of necessity relatively long springs. The plunger stroke is then fully available when the pressure rises above the working pressure, and the characteristics of the springs determine the final rise of pressure which will complete the stroke.

Hydraulic Pipe Work.

Pipes for conveying hydraulic power can be of cast iron or steel for working pressures up to a range of from 900 lb. to 1,200 lb. per sq. in., according to size. Above these pressures lap-welded and weldless steel tubes are necessary. Flanged joints are invariable: in the case of

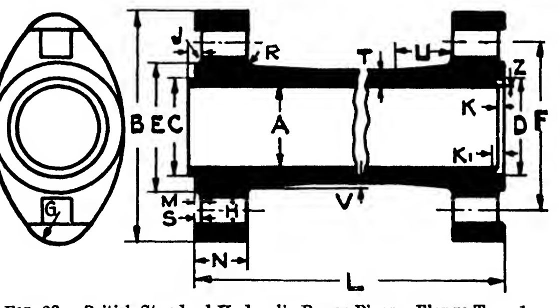

FIG. 28. British Standard Hydraulic Power Pipes: Flange Type 1.

* A deflecting valve is really a safety valve usually opened directly by the accumulator at the top of its stroke, and closed immediately water is released from the system and the accumulator descends. If the pumps are still delivering more water than is being drawn off, the accumulator ascends again and opens the valve. To prevent excessive wear of the accumulator ram at the top of its stroke it is better practice to use an unloading valve.

† 'Function and Design of Air Chambers,' C. G. Richardson, *Power*, July 8, 1913.

cast iron pipes the flanges are integral with the pipes, whilst for steel pipes the flanges are loosely screwed on, the joint being made on the pipe ends.

Joint rings for hydraulic pressure are not made of soft materials as these are likely to be extruded. Copper joint rings are commonly used and the B.S.S. recommendation for soft copper joint rings is that they should be $\frac{5}{16}$ in. thick. The flanges do not meet when the joints are drawn up.

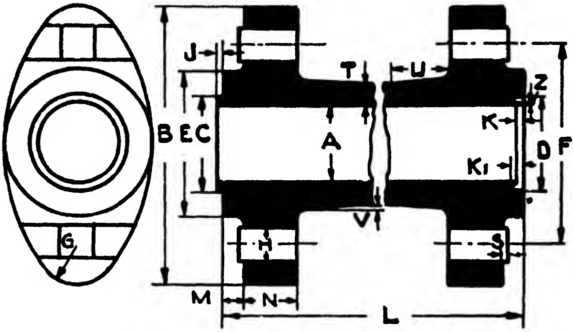

FIG. 29.—British Standard Hydraulic Power Pipes: Flange Type 2.

BRITISH STANDARD CAST IRON HYDRAULIC POWER PIPES.

	Class A. For Working Pressures not exceeding 900 lb. per sq. in.							Class B. For Working Pressures from 900 to 1,200 lb. per sq. in.					
	Flange: Type 1.			Flange: Type 2.				Flange: Type 1		Flange: Type 2.			
	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
A	2	3	4	5	6	7	8	2	3	4	5	6	7
B	$7\frac{1}{2}$	$9\frac{1}{2}$	$11\frac{1}{2}$	$14\frac{1}{4}$	16	20	22	$8\frac{3}{4}$	$11\frac{1}{4}$	14	$16\frac{1}{2}$	19	21
C	$2\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{3}{4}$	$5\frac{3}{4}$	7	8	9	$2\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{7}{8}$	6	7	8
D	$2\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{3}{4}$	$5\frac{3}{4}$	7	8	9	$2\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{7}{8}$	6	7	8
E	$3\frac{7}{8}$	$5\frac{1}{4}$	$6\frac{1}{2}$	$7\frac{5}{8}$	9	12	14	$4\frac{1}{4}$	6	$7\frac{1}{2}$	9	$10\frac{1}{2}$	13
F	$5\frac{3}{8}$	$6\frac{3}{4}$	$8\frac{1}{4}$	$10\frac{1}{4}$	$11\frac{1}{2}$	$14\frac{1}{2}$	16	6	8	$10\frac{1}{4}$	$12\frac{1}{4}$	14	$15\frac{1}{2}$
G	$1\frac{1}{16}$	$1\frac{3}{8}$	$1\frac{5}{8}$	2	$2\frac{1}{4}$	$2\frac{3}{4}$	3	$1\frac{3}{8}$	$1\frac{5}{8}$	$1\frac{7}{8}$	$2\frac{1}{8}$	$2\frac{1}{2}$	$2\frac{3}{4}$
H	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$
J, K	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$
K_1	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
M	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$\frac{1}{4}$	$\frac{1}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$
N	$1\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{2}$	$3\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{3}{4}$	3	$3\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$
R	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$
S	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
T	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{13}{16}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$\frac{5}{8}$	$\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$
U	—	—	—	—	—	—	—	3	3	3	3	3	3
V	—	—	—	—	—	—	—	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$
Z	$\frac{1}{8}$	$\frac{1}{8}$			$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$
Diam. of Bolt	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	2	$2\frac{1}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$
Length of Bolt Overall	$5\frac{5}{8}$	$6\frac{5}{8}$	$7\frac{7}{8}$	$10\frac{7}{8}$	$11\frac{5}{8}$	14	15	$6\frac{5}{8}$	$8\frac{1}{8}$	$11\frac{1}{8}$	$12\frac{5}{8}$	14	$15\frac{1}{2}$

All spigots and sockets to be made to gauge.
All nuts, bolts and screw threads to be British Standard Whitworth.

Cast Iron Pipes.—Cast iron hydraulic power pipes are covered by B.S.S. No. 44—1948 which defines pipes from 2 ins. to 8 ins. diam. for working pressures up to 900 lb. per sq. in. (class A), and from 2 ins. to 7 ins. diam. for pressures up to 1,200 lb. per sq. in. (class B). Standard lengths are 6 ft. and 9 ft. long and short lengths are also made from 1 ft., increasing upwards by increments of 6 ins. to 7 ft. for class A and B pipes, with several special lengths to make up, with the addition of a tee, the standard lengths. B.S.S. No. 44 also defines long tees, short tees, 90° and 45° bends, and blank flanges.

Steel Pipes.—Steel tubes can, of course, be used for any lower hydraulic pressure, but for pressures above 1,200 lb. per sq. in., steel tubes must be employed. They can be lap-welded for the lower range of high pressures, but for the highest pressures solid drawn steel tubes are used. Such tubes, in the smaller diameters, have been made for pressures up to 10,000 lb. per sq. in.

Thicknesses of steel pipes up to 3 ins. diam. are suggested by B.S.S. No. 778—1938, which defines oval steel flanges for hydraulic working pressures of from 750 lb. per sq. in. to 1,500 lb. per sq. in. (up to 3 ins. diam.) and from 1,500 lb. per sq. in. to 4,500 lb. per sq. in. (up to 2 in. diam.).

THICKNESS OF PIPES AND CLASS OF STEEL SUGGESTED BY B.S.S. NO. 778—1938.

Nominal Bore.	Outside Diam.	22/28 tons per sq. in. Ultimate Tensile.						35/40 tons per sq. in. Ultimate Tensile.		
		1,500 lb. per sq. in.			4,500 lb. per sq. in.			4,500 lb. per sq. in.		
		Minimum Thickness.		Approx. Actual Bore.	Minimum Thickness.		Approx. Actual Bore.	Minimum Thickness.		Approx. Actual Bore.
Ins.	Ins.	S.W.G.	Ins.	Ins.	S.W.G.	Ins.	Ins.	S.W.G.	Ins.	Ins.
½	27/32	9	0·144	0·56	6	0·192	0·46	9	0·144	0·56
¾	1 1/16	8	0·160	0·74	—	¼	9/16	8	0·160	0·74
1	1 11/32	7	0·176	0·99	—	5/16	23/32	6	0·192	0·96
1¼	1 11/16	6	0·192	1·3	—	7/16	13/16	—	¼	1 3/16
1½	1 29/32	5	0·212	1·48	—	7/16	1 1/32	—	5/16	1 9/32
2	2⅜	5	0·212	1·95	—	9/16	1¼	—	⅜	1⅝
2½	3	—	¼	2½	—	—	—	—	—	—
3	3½	—	5/16	2⅞	—	—	—	—	—	—

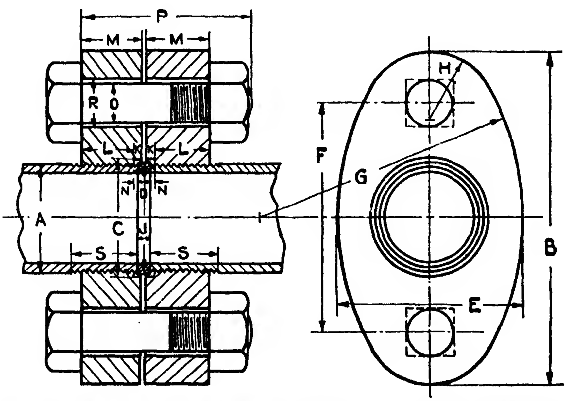

FIG. 30.—British Standard Oval Steel Flanges for Steel Pipes: Working Pressures from 750 up to 1,500 lb. per sq. in.

DIMENSIONS FOR OVAL STEEL FLANGES FOR STEEL PIPES FOR HYDRAULIC WORKING PRESSURES OF FROM 750 UP TO 1,500 LB. PER SQ. IN.

(All dimensions in inches.)

Nominal Bore of Pipe.	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	R	S
1/2	27/32	4 1/4	31/32	15/16	2 1/8	2 3/4	3 11/16	1	5/16	3/16	11/16	7/8	1/16	5/8	2 1/4	11/16	15/16
3/4	1 1/16	4 5/8	1 3/16	1 5/32	2 3/8	3	5	1 1/8	5/16	3/16	11/16	7/8	1/16	5/8	2 1/2	11/16	15/16
1	1 11/32	5	1 15/32	1 7/16	2 3/4	3 3/8	5	1 1/8	5/16	3/16	13/16	1	1/16	5/8	2 3/4	11/16	1 1/16
1 1/4	1 11/16	5 3/4	1 13/16	1 25/32	3 1/8	4	5 5/8	1 1/4	5/16	3/16	15/16	1 1/8	1/16	3/4	3	7/8	1 3/16
1 1/2	1 29/32	6 1/4	2 1/32	2	3 1/2	4 1/2	5 11/16	1 3/8	5/16	3/16	15/16	1 1/4	1/16	3/4	3	7/8	1 3/16
2	2 3/8	7 1/4	2 1/4	2 15/32	4 1/8	5	5 13/16	1 3/8	5/16	3/16	1 3/16	1 3/8	1/16	7/8	3 3/4	1	1 7/16
2 1/2	3	8	3 1/4	3 7/32	4 5/8	5 7/8	5 5/16	1 3/8	5/16	3/16	1 9/16	1 1/2	1/16	1	4 1/4	1 1/8	1 15/16
3	3 3/8	9	3 3/4	3 23/32	5 3/8	6 5/8	5 3/4	1 1/2	5/16	3/16	1 13/16	2	1/16	1 1/8	5 1/4	1 1/4	2 1/16

NOTE.—The above flanges when fitted to the pipes have been designed for a test pressure of 3,000 lb. per sq. in.

DIMENSIONS FOR OVAL STEEL FLANGES FOR STEEL PIPES FOR HYDRAULIC WORKING PRESSURES FROM 1,500 UP TO 4,500 LB. PER SQ. IN.

(All dimensions in inches.)

Nominal Bore of Pipe.	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	R	S
1/2	27/32	4 1/4	31/32	15/16	2 1/8	2 3/4	3 11/16	1	5/16	3/16	13/16	1	1/16	5/8	2 3/4	11/16	1 1/16
3/4	1 1/16	4 5/8	1 3/16	1 5/32	2 3/8	3	5	1 1/8	5/16	3/16	15/16	1 1/8	1/16	3/4	3	7/8	1 3/16
1	1 11/32	5	1 15/32	1 7/16	2 3/4	3 3/8	5	1 1/8	5/16	3/16	1 1/16	1 1/4	1/16	3/4	3 1/2	7/8	1 5/16
1 1/4	1 11/16	5 3/4	1 13/16	1 25/32	3 1/8	4	5 5/8	1 1/4	5/16	3/16	1 7/16	1 3/8	1/16	7/8	4 1/4	1	1 11/16
1 1/2	1 29/32	6 1/4	2 1/32	2	3 1/2	4 1/2	5 11/16	1 3/8	5/16	3/16	1 9/16	1 3/4	1/16	1	4 5/8	1 1/8	1 13/16
2	2 3/8	7 1/4	2 1/4	2 15/32	4 1/8	5	5 13/16	1 1/2	5/16	3/16	1 15/16	2 1/4	1/16	1 1/8	5 1/2	1 1/4	2 3/16

NOTE.—The above flanges when fitted to the pipes have been designed for a test pressure of 6,000 lb. per sq. in.

Solid drawn steel hydraulic tubes for higher working pressures than 4,500 lb. per sq. in. are made in sizes up to 6 ins. diam. Makers adopt flanged joints based on B.S.S. No. 778, but use circular flanges in order to obtain increased boltage.

Very few high pressure hydraulic mains are laid of greater diameter than 4 to 5 ins.

VELOCITY IN HYDRAULIC MAINS.

Pipe lines between pumps and accumulators are usually of relatively large size with water velocities of from 5 to 15 ft. per sec., the higher velocities being used with the higher pressures. In trunk lines larger velocities are often allowed, up to 25 ft. per sec., while velocities of up to 50 ft. per sec. are sometimes permitted through lines serving small individual machines. In general, velocities should be kept reasonably low to reduce frictional losses and the risk of water hammer.

VENTILATION AND DRAINAGE OF HYDRAULIC MAINS.

Air within an hydraulic system can be troublesome and it is very necessary that air vents with suitable valves be provided at all high points and wherever air may collect.

Freezing is another serious cause of derangement and hydraulic systems must, therefore, be fitted with emptying valves at appropriate low points.

ISOLATING VALVES.

Adequate stop valves must be provided for operation and maintenance, and it should be possible to isolate every machine from the rest of the system.

Hydraulic Presses.

A very great variety of work, in many different industries, provides applications for all types of presses of from less than 100 tons capacity up to huge machines of 10,000 tons or more capacity. The scope of press work is indicated by the following list.

Forging	Lead working.
Flanging, bending, dishing, staving	Aircraft production
Punching, shearing	Drawing
Piercing	Foundry moulding
Tube making	Stamping
Extruding	Moulding of plastics
Bushing	Ammunition making
Vulcanising	Gun straightening
Baling	Printing
Brick and tile making	Plywood and veneer manufacture
Extracting	Stretching (sheet metal)
Pelletting	Die casting, etc.

The continued development should be noted of hydraulic presses for special purposes, and the evolution of the self-contained press operated directly from pumps, automatically or semi-automatically, with push button control.

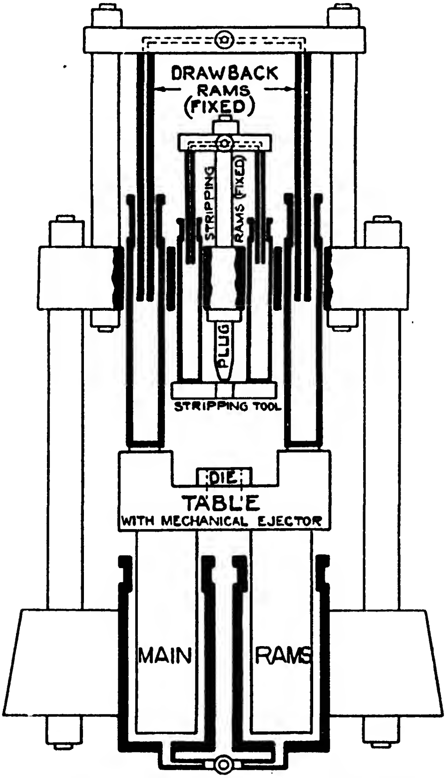

Fig. 31.—Elements of Piercing Type Hydraulic Press.

Presses differ in their characteristics, depending upon the class of work for which they are intended. A forging press, for example, although it may use filling water for bringing the ram down to the job, must be capable of making a series of short working strokes at full load at any point in the total stroke. Presses for specialised operations, such as tube staving, on the other hand, have a fixed working stroke and do not develop full load until the final squeeze, two pressures being normally used. It is not possible to deal with every type of press in these pages; several outstanding examples are, however, quoted.

A Large Forging Press.

An example of a 7,000 tons forging press has a single main ram with a working pressure of 2¼ tons per sq. in., automatically limited to a maximum of 2¾ tons per sq. in., the designed pump pressure being 3 tons per sq. in. Ingots weighing up to 250 tons are dealt with at a penetration speed of 2 ins. per sec. at a rate of ten 5-in. penetration strokes per minute at full load; or, alternatively, with a lesser number of long strokes or a greater number of short strokes in the same period. The return stroke of the main ram is effected by two lifting rams operated by water at 1 ton per sq. in. pressure, which give a total lifting effort of 454 tons. The lowering speed of the main ram is 6 to 7 ins. per sec., and the maximum gap between crosshead and anvil plate is 17 ft. 6½ ins. Water exhausted from the main cylinder goes into a closed low pressure air vessel system under a maximum pressure of 70 lb. per sq. in., and water from this system is used for prefilling the ram cylinder. The main press operations are controlled by a single master lever which governs the admission and exhaust, in turn, of the prefilling water, the high pressure water, and the lower pressure water to the lifting cylinder. The press is operated directly from a high pressure treble ram pump * driven by a 2,500 b.h.p. motor with flywheels, capable of developing a maximum of 6,500 b.h.p. to meet the peak load. The 1 ton per sq. in. supply used for lifting is obtained from an accumulator system which supplies also other smaller hydraulic presses in the same works.†

A much larger forging press,‡ made in Germany for the U.S.S.R., has a capacity of 15,000 tons. This remarkable tool has three main rams, the centre ram exerting 6,000 tons and the two side rams 4,500 tons each. Eight return rams in pairs are also provided, giving a total drawback pressure of from 460 to 690 tons, the weight of the moving parts being 460 tons. The main rams work at a pressure of 5,800 lb. per sq. in., and are supplied by a steam intensifier. The drawback rams use water at 2,900 lb. per sq. in., supplied by three treble ram pumps and a pneumatic accumulator system. This pressure may be intensified to 4,400 lb. per sq. in. Make up water is supplied at 70 lb. per sq. in.

This press can accommodate forgings up to nearly 16 ft. in diameter with a maximum weight of 300 tons. The total weight of the machine and equipment, which extends to a height of 83 ft., is 3,500 tons.

A Staving Press for Large Steel Pipes.—A horizontal staving press, used for forming sockets on welded steel pipes up to 48 ins. diam., uses a pump pressure of 1,500 lb. per sq. in. on the main ram, followed by an intensified pressure of 4,500 lb. per sq. in. The ram has a fixed working stroke. Operations are as follows:—

The pipe with heated end is placed in position in the machine and the socket-to-be is gripped by a split diehead operated hydraulically. The die plug, attached to the main ram, is then entered into the pipe by admitting low pressure filling water to the main cylinder and water at 1,500 lb. to auxiliary jacking cylinders. The pump pressure of 1,500 lb. per sq. in. is then admitted to the main cylinder, the low pressure supply being automatically cut off. When the second movement of the ram has ceased an intensified pressure of 4,500 lb. per sq. in. is admited to the main cylinder, the 1,500 lb. per sq. in. supply being automatically isolated. This completes the formation of the socket and the main ram cylinder is then opened to the low pressure filling water system and the 1,500 lb. per sq. in. pressure is admitted to drawback rams. When the main ram is clear, the diehead is opened and raised (also by hydraulic means), allowing the socketted pipe to be removed.

Six valves, interlocked and controlled by one lever, govern the admission, the isolation and the exhaust of the filling water and of the 1,500 lb. per sq. in. pumped supply to the jacking cylinders, the main cylinders and the drawbacks. Two other valves, operated by a second lever, control the inlet and exhaust of the intensified pressure of 4,500 lb. per sq. in. The small jack cylinders referred to, which use water at 1,500 lb. per sq. in., have the purpose of hastening the travel of the main ram up to the job. In practice the 1,500 lb. per sq. in. pressure supply is often admitted to the main cylinder also, to increase the speed of production. When socketting 48-in. diam. tubes the main ram moves forward 2 ft. 6 ins. in 20 secs. with low pressure filling water in the main cylinder and 1,500 lb. per sq. in. pressure in the jacking cylinders, and at 2 ft. 6 ins. in 5 secs. when 1,500 lb. per sq. in. pressure is admitted also to the main cylinder for the full jacking stroke. Otherwise the 1,500 lb. per sq. in. pressure is admitted only for about 6 ins. of the working stroke, the intensified pressure of 4,500 lb. per sq. in. being used for the final 2 ins. of the working stroke, the speed of the ram slowing up as the work gets harder. The drawback cylinders, using water at 1,500 lb. per sq. in., move the main ram 2 ft. 6 ins. in 5 secs.

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* This pump is referred to on p. 768.

† *Engineering*, Feb. 15, 1935, p. 171.

‡ *Engineering*, Feb. 1, 1935, p. 112.

*Vulcanising Presses for Rubber Products.**—Presses used for the vulcanising of rubber articles include autoclave or open-steam presses such as those used for tyres; dry heat presses used for soles and heels, belting, hot water bottles and similar articles; and presses with several 'daylights,' used for sheeting, tiling and flooring, with drawplates on which the material is built up by hand before being run into the press. For general moulding, dry steam-heated presses having up to '8 daylights' are employed, with single large diameter rams, usually operated on 2-pressure systems through automatic change-over valves.

Hydraulically loaded rolls are used in continuous vulcanising machines for the production of long lengths of rubber materials.

A 10.000 *Ton Rubber Vulcanising Press.*—This outstanding example † of a very large press designed for vulcanising rubber belts and mats has heated platens with a working surface 31 ft. 4 ins. × 8 ft. 4¾ ins. Dealing only with its hydraulic operation, the table is carried on 12 rams arranged in pairs. Pressure water at 1,650 lb. per sq. in. is first admitted to one pair of ram cylinders and the table moved upwards to close the platens on to the work, the other cylinders filling with water form an overhead tank. This movement takes about 10 secs. The pressure of 1,650 lb. per sq. in. is then admitted to all the cylinders, the table moving through the fraction of the stroke necessary to put the load on the material. To increase the load to that necessary for whatever work is being vulcanised, water at any pressure between 1,650 and 4,500 lb. per sq. in. may be admitted from an intensifier with automatic control. The table is lowered by 8 full stroke

SOME HYDRAULIC PRESSES OF RECENT MANUFACTURE.

Purpose.	Maximum Capacity.	Maximum Working Pressure. Lb./sq. in.	Pumps.	Accumulator or Intensifier.	Remarks.
Forging press	15,000 tons	5,800	3 treble ram pumps for drawback rams	Steam intensifier for main rams, pneumatic accumulator for drawback rams (2,900 to 4,400 lb./sq. in.)	Tool has 3 main rams and 8 return rams. Weight of this tool 3,500 tons
Rubber vulcanising press	10,000 tons	4,500	Treble ram pumps, 1,650 lb./sq. in.	Steam Intensifier	
Pipe staving press	48-in. diam. Pipes	4,500	Treble ram pumps, 1,500 lb./sq. in.	Air loaded accumulator and intensifier	
Gear hardening press	1,600 tons	6,250	Vertical long stroke 3-throw variable high speed pump, 350 to 6,250 lb./sq. in.	Accumulator and intensifier	
Moulding press for plastics	1,300 tons	4,480	Treble ram	None	
Locomotive wheel press	300 tons		Treble ram	None	4 main rams, each with 2 return rams
Sheet metal stretching press	150 tons		High speed pumps	None	
Moulding press for plastics	75 tons		High speed 2-throw with adjustable maximum pressure	None	Single ram
Portable press for bending and stretching	50 tons	2,240	High speed ram pump 1,000 to 1,500 r.p.m.	None	Press mounted on 4-wheeled truck

* 'Rubber Machinery and Process Plant,' E. Morris, *Proc. I.Mech.E.*, June 1939, Vol. 141, p. 110.

† *Engineering*, Feb. 9, 1934, p. 156.

return cylinders in a few seconds. The speeds of the rams are controlled automatically and the vulcanised material is produced in correct thicknesses within very fine limits.

Air loaded accumulator and intensifier plant serve this large press.

*Moulding Presses for Plastics.** —Double injection moulding presses of horizontal design are used for the production of small articles from thermo-plastics. The powder, contained in a heated cylinder, is forced into the mould at the first movement of the ram and the return stroke of the ram ejects the finished article from the mould, whence it falls into a container. These presses give continuous production, but it is important that the hydraulic supply should be free from pulsations.

For larger mouldings single acting vertical presses are used with drawback rams, and ejector rams for the finished articles.

The suitability of hydraulic presses for the moulding of plastics is to be noted. The forming or moulding process must have a degree of irregularity, the closing mould being free to adapt its stroke elastically to the flow of the material. This condition is perfectly fulfilled by an hydraulic press with suitable power supply. The application of individual high speed, high pressure pumps to presses for plastics moulding is a prominent development.

Hydraulic Actuating Machinery.

Hydraulic power has important applications in civil and mechanical engineering, to the actuation of machinery—*i.e.* the application of forces to control the relative movements of machine members, as distinct from press work in which the hydraulic effort is directed to the shaping of material in course of manufacture.

Although electricity is now widely used for many driving duties that were formerly undertaken by hydraulic power, there are many cases in which hydraulic power, for various reasons, is still preferred.

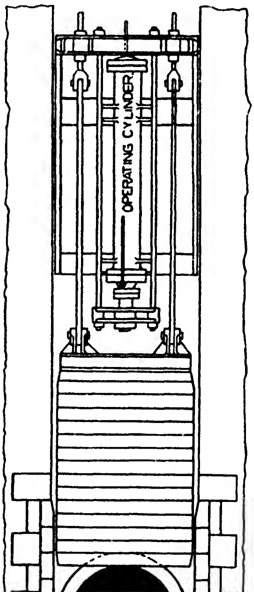

Fig. 32.—Hydraulic Dock Sluice Machine.

* 'Hydraulic Machinery and the Plastic Industries.' M. R. Rhodes, *Proc. I.Mech.E.*, September 1939, Vol. 141, p. 143.

Sluices and Lock Gates.—In dock practice, before the general application of the electric motor, there was great convenience in having a central hydraulic power station, with pumps and accumulators, for the provision of the relatively heavy efforts needed to operate dock sluices and lock gates as well as coal hoists, cranes and capstans. Most of the large docks still retain their hydraulic power plants and have, in many cases, modernised them. Pressures of from 700 to 800 lb. are used. Gates are often operated through hydraulic engines of the 3-cylinder radial type, and direct acting rams and cylinders are also employed as, for example, in the operation of sluices.

Hydraulic power has been applied with success to the automatic operation of sluices or penstocks in sewage installations. In one of the latest installations,* the largest sewage disposal works in the Empire, six large grit chambers for storm water have penstocks installed at each end, those at the inlets being 54 ins. wide × 114 ins. deep, and those at the outlet ends 72 ins. wide × 96 ins. deep, in both cases for operation under unbalanced heads of up to 12 ft. The penstock frames are extended to carry hydraulic actuating cylinders, the design ensuring that all the operating forces are confined to the sluice structure. A working pressure of 500 lb. per sq. in. is generated by two three-throw ram pumps each of 5 g.p.m. capacity, with a pressure medium of oil of low freezing point. A pneumatic type accumulator is used with automatic control of the pumps by means of float switches on the fluid tank, which bring in each pump at a slightly different pressure. A standby hand-operated pump enables pressure to be generated even in the event of the failure of the electric supply.

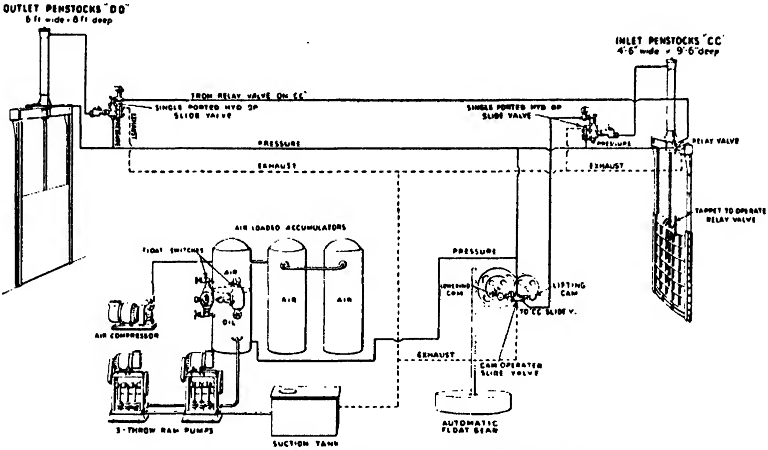

FIG. 33.—West Middlesex Sewage Disposal Scheme.

(Diagrammatic arrangement of Hydraulic Pressure Equipment and Automatic Control of Storm Water Penstocks.)

The control valves, by means of which the penstocks are opened in sequence, are operated by float gear which responds to a given rise of level in the main sewer, the opening of the outlet penstock in each case being controlled by the opening of the inlet penstock. This automatic gear is independent of external power and would not be affected by dislocation of electricity supplies which might be the result of a severe storm.

Bridges.—There are numerous examples of bridges in which hydraulic power is used for lifting and slewing machinery. Some details are given in Section XIV, Part IV, p. 525.

Adjustable Floors.—Even when electricity is fully available, advantage is taken of the special characteristics of hydraulic power when circumstances are appropriate. An excellent modern example of the conversion of electrical power to hydraulic power for the convenient manipulation of a heavy and cumbersome load, is found in the 720 tons adjustable floor at Earls Court.† This floor is in three sections, each 65 ft. × 95 ft., and two rams, each 28 ins. in diam., are employed

* West Middlesex Sewage Disposal Works.
† *The Engineer*, December 24, 1937, p. 707.

to raise each section to the level required, after which permanent stilts take the weight. Operation of 'chocks,' by means of which the floor load is transferred through the stilts to the reinforced concrete load distributing beams in the bottom of the pool, is also hydraulic, and when the stilts have taken up the weight of the floor and its loading, the hydraulic power can be turned off the main lifting rams.

Light Alloy Casting Machines.—Hydraulically operated continuous casting machines for light alloys have been used successfully in recent years and their use has been greatly stimulated by the war-time demand for aluminium and magnesium production.

Testing Machines.—For driving materials testing machines of various kinds, from the Brinell hardness testing unit with its tiny hand pump, up to the largest tensile and compression machines, hydraulic power is a convenient and flexible agent.

Pressure testing machines for valves, pipes, fittings and general castings range from relatively elaborate machines with semi-automatic control valves for repetition testing, to simple hand pumps for connection to pipe lines or blanked off castings. Very high pressures are sometimes applied, reaching 20,000 lb. per sq. in.

Hydraulic Transmissions.—There has been great development in mechanisms for the transmission of power through fluid couplings, fluid torque converters (Descriptive Sec. XIX (II), hydraulic transformers, and hydraulic variable speed gears (Descriptive Sec. XIX (II), and many ingenious devices for transmitting, transforming and receiving motion and energy hydraulically, have been evolved. Among successful applications of hydraulic drives, those to machine tools are of outstanding importance (Sect. XXII (I), p. 1022).

Power operation of ship's main and emergency steering gear and of slipway haulages * are among successful applications of hydraulic drives in which exceptionally heavy and fluctuating loads have to be dealt with.

Bulkhead Doors.—Watertight apparatus for ships, in conformity with the regulations of the British Ministry of Shipping, Lloyd's, Bureau Veritas and the International Convention, include watertight bulkhead doors which, in many of the most important installations, are controlled hydraulically. The chief requirements affecting the hydraulic operation of watertight bulkhead doors are as follows:—

It must be possible to close all the doors simultaneously from the bridge. Under normal conditions of control it must be possible for any door to be operated locally under power or by hand, or from the bulkhead deck as required. When closed from the bridge any door must be

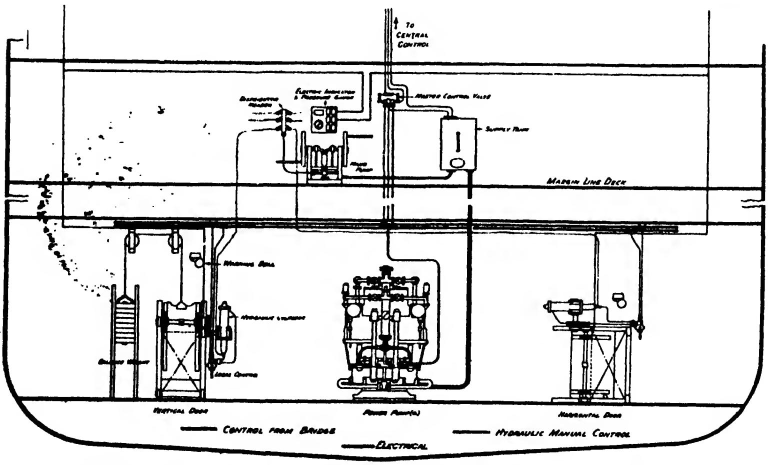

FIG. 34.—Hydraulically Controlled Watertight Bulkhead Doors—Stone System.

* 'Recent Developments in the use of Hydraulic Power,' H. O. Town, *Proc. I.Mech.E.*, 1940, Vol. 143, p. 129.

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capable of being opened locally, but must automatically reclose when the controls are released. It must be possible to open any door from the local position at any time under power or by hand.

Hydraulic power is particularly suited to the fulfilment of these requirements and several systems have been evolved.

In the extensively applied 'Stone' system a working pressure of 700 lb. per sq. in. is used, the hydraulic fluid being delivered either by steam pumps or by electrically driven pumps of the rotary type, with air accumulator storage and automatic running control over an open circuit. The bulkhead doors are of the sliding type working vertically or horizontally, and are opened and closed through the agency of hydraulic cylinders and appropriate control valves, necessary provision being made for operation under power, and by hand in the event of failure of power, in accordance with the regulations.

In a proposal to apply the Glenfield-Lockheed system of remote hydraulic control to the operation of watertight bulkhead doors (provisionally approved by the British Ministry of Shipping) a pressure of about 800 lb. per sq. in. is provided by electrically driven transmitter pumps automatically controlled, with pressure storage of the hydraulic medium. Pairs of Lockheed slave cylinders actuate the doors by direct effort. This system provides the following hydraulic controls:

The doors can be opened or shut individually from the bridge using selector valves, or they can all be closed or opened together using a master control.

Doors can be opened or closed locally under power, or by hand, but they will return to the positions decided by the bridge.

Local power controls can be locked, but only in the door shut positions.

Doors can be closed and opened from the bulkhead deck by hand.

When the doors are closed by local or bulkhead deck hand controls, the power circuit is isolated, thus preventing the bridge from opening the doors. This power circuit lock can be released from either the local or bulkhead deck control positions.

A door that has been partly closed by hand from the local position can only be closed and not reopened from the bulkhead deck position.

It is of interest to note that the hand power in this system is also applied hydraulically, the hand controls being actually small hydraulic pressure transmitters independently connected to the door slaves, with appropriate automatic isolating valves in the circuits.

Small Scale Automotive Mechanisms.

Exceptional progress has been made with highly developed small scale automotive mechanisms or servo motors, some of which have become familiar even to laymen. The Lockheed and Bendix Cowdrey automobile brake systems are examples.

Aircraft Developments.—Hydraulic actuation, through remote controls, of aircraft mechanisms has been widely adopted. Hydraulically operated apparatus for aircraft includes retractable undercarriages and landling flaps of high speed craft, bomb compartment doors and releasing equipment for war planes, gyroplanes, composite apparatus for mid air launching, and auto pilots.*

Lockheed Hydraulic Remote Controls.—The Lockheed system of remote hydraulic control employs a range of very highly developed standardised components, from which automotive mechanisms can be built up for many purposes. The essential components of this system include hand operated and motor driven rotary transmitters, selector valves in ganged units, slave cylinders of several types, various automatic circuit control valves, and pressure tubing. Specially blended oil is used as the hydraulic medium. High overall efficiencies of the order of 90 per cent. are claimed for this system in comparison with mechanical gearing which for many operational purposes has an overall efficiency of only 25 to 30 per cent.

Rotary Transmitters.—Two-cylinder and 4-cylinder units are employed giving pressures up to 700 lb. per sq. in. and deliveries which depend upon the throw of the crankshaft, of which there is a choice of three for each unit. The transmitters have incorporated in them valve arrangements which give reverse directions of fluid output for opposite directions of rotation of the crankshaft so that double acting systems can be operated where both push and pull efforts are required from the receiver unit. An adjustable relief valve is also included in the transmitter, or separate 2-way relief valves which make it possible to give two different pressures, depending upon the direction of rotation of the crankshaft. When the handwheel is released or the motor

* See *Journal, I.Mech.E.*, Vol. 143, No. 2, 1940.

stopped, the hydraulic circuit is 'open' and variations of fluid volume due to temperature changes have no effect.

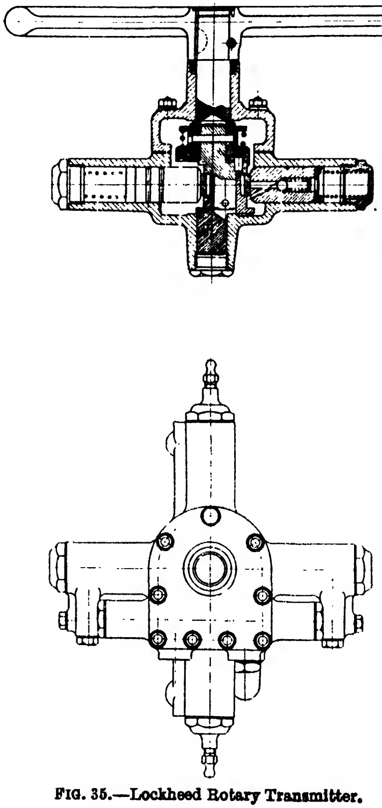


FIG. 35.—Lockheed Rotary Transmitter.

Slave Cylinders.—Slave cylinders are of several types all of which consist essentially of a small bore cylinder (the usual sizes are between $\frac{3}{4}$ in. and 3-in. diam.) fitted with a piston, in

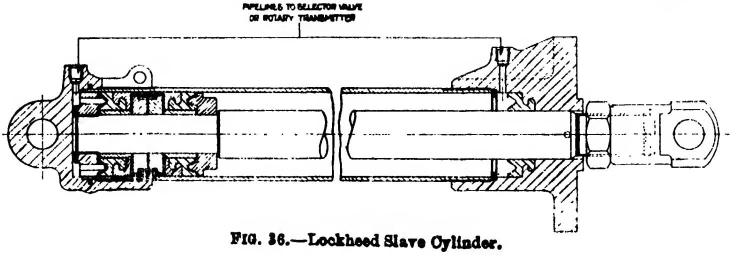


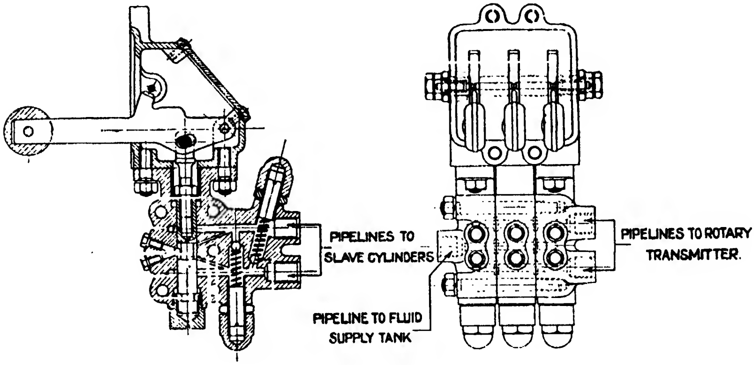
FIG. 36.—Lockheed Slave Cylinder.

which is incorporated a mechanical locking device to ensure that the piston can only be moved in the cylinder when internal fluid pressure is applied, and that it cannot be moved by external effort. Any mechanism operated by a slave cylinder remains locked in the position determined by the rotation of the transmitter, as soon as the handwheel is released. When heavy reactive loads are involved, an hydraulic lock is employed as well. Slave cylinders without the auto-lock are used for certain purposes.

Pairs of hydraulically synchronised slave cylinders are used in cases where mechanical interconnection between the two is appropriate but difficult or impossible to attain.

The standard auto-lock slave cylinder can be fitted with indicator transmitters or limit switches for the transmission of visual indications of the movements of the piston.

The cylinders are single ended giving differential efforts or double ended to give equal efforts on both strokes.



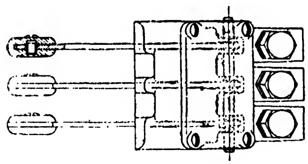


FIG. 37.—Lockheed Selector Valves.

Selector Valves.—Selector valve units consist of up to nine separate selector valves ganged together, each valve operated by its own lever and so arranged that pressure fluid from one rotary transmitter may be directed to operate slaves at any one of a number of remote receiving points, by depression of the appropriate lever. Banks of selector valves can be employed. The levers are positively interlocked so that only one valve may be selected at a time. Electrical switching arrangements can be incorporated for position indication.

Applications.—Among many successful applications of the Lockheed hydraulic remote control system may be mentioned Lockheed-Babcock remote damper control gear, the Glenfield-Lockheed system for the remote control of valves, marine steering gear and marine reversing equipment, accelerator and throttle control, the opening and closing of windows, and a wide variety of similar mechanisms.

Hydraulic Remote Control of Valves.—The conventional system for the hydraulic control of valves, often utilising water at ordinary distribution pressures, has disadvantages among which may be mentioned the large hydraulic cylinders which are often necessary, and the necessity to maintain the pressure on the piston in order to keep the valve in the desired position, creeping of the valve usually being unavoidable. Moreover, with wear of operating valves and piston leathers, the tendency to creep becomes very pronounced.

Mechanical locks, auxiliary to the conventional hydraulic equipment of a valve, have had limited applications and their use is not extensive. The Lockheed system applied to valves is differentiated from the conventional system by the smallness of the slave cylinders, resulting from the use of high pressure and by the fact that the hydraulic piston incorporates a mechanical lock which is applied automatically whenever pressure is taken off the piston at the completion of every operational movement. When heavy reactive loads are involved, as in large valves and sluice gates, owing to the weight of the moving parts, an hydraulic lock is included in the circuit in the shape of an automatic valve in the pressure pipe leading to the lower side, say, of the cylinder.*

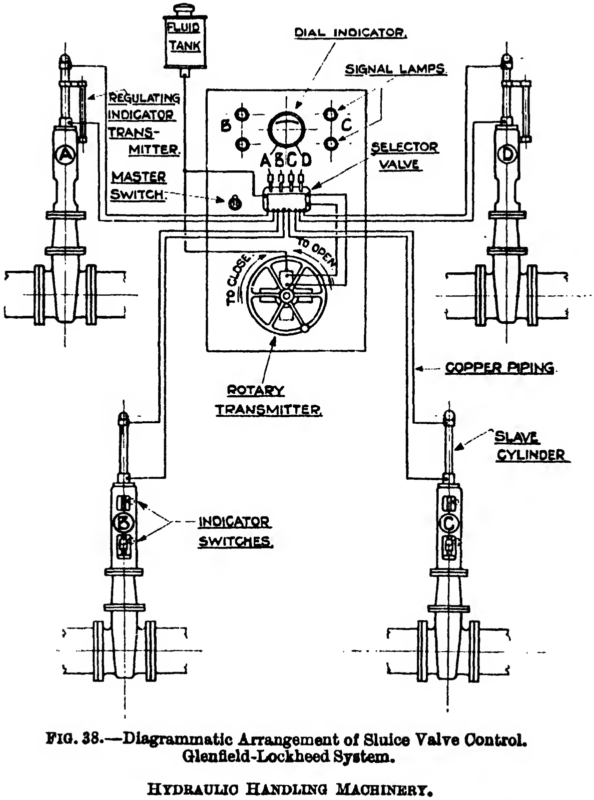


FIG. 38.—Diagrammatic Arrangement of Sluice Valve Control. Glenfield-Lockheed System.

HYDRAULIC HANDLING MACHINERY.

Cranes, Capstans, etc.—Hydraulic cranes, derricks and hoists have been largely superseded by electrically operated machines but there are, nevertheless, a large number of such hydraulic lifting devices still in use at docks and harbours, in shipyards and in many other industrial undertakings. Motor car inspection jacking platforms are well-known examples of recently specialised lifting devices. In most cases the working pressure is between 700 and 1,500 lb. per sq. in. Similarly, although electrically driven capstans are much more commonly installed now than hydraulic capstans, the latter machines remain in quite extensive use, and examples have been installed quite recently at several important sites.

* 'A New System of Hydraulic Remote Operation for Control Valves,' E. Bruce Ball, jun., *Journal of I.Mech.E.* 1940, Vol. 143.

Where existing hydraulic power systems are available there is quite good reason to consider the use of the silent and robust hydraulic machine for heavy lifts and pulls. Simplicity, compactness, absence of fast running components and ease of control are among the advantages presented by hydraulic operation of lifting machinery. Although in the past, economical working was only obtained under full load conditions, with modern variable pressure and variable delivery hydraulic pumps, this objection need not be sustained.

Friction of Hydraulic Packing.

Hydraulic Leathers (see Sect. XIX (III), p. 806).

Compression Packings.—(Braided Flax or Hemp).

$$F = 0 \cdot 26 \text{ PD (Houghton)}.$$

Where F = friction (lb.);
P = total load on plunger (lb.);
D = plunger diameter (ins.).

Hydraulic Pressure Gauges.

Standard Bourdon type pressure gauges with solid drawn steel spring tubes of special alloys are manufactured for pressures up to 32 tons per sq. in. Gauges should be calibrated to twice the working pressure they are to register. Pulsating pressures and sudden variations of pressure should not be allowed to act directly on hydraulic gauges and protection should be provided in the form of fine regulating valves which should be adjusted to reduce the movement of the pointers to the smallest perceptible tremor.

Each gauge should be calibrated against a deadweight testing machine and an individual dial should be written. Frequent checking of the calibrations is desirable.

See also Descriptive Section XIX, Part II:

Bell's Asbestos and Engineering Ltd.
Exactor Control Co. Ltd.
Keelavite Rotary Pumps and Motors Ltd.
Vickers-Armstrongs Ltd. (Variable Speed Gear Department).

SECTION XIX

PART III

CLASSIFICATION OF PUMPS — RECIPROCATING PUMPS — PUMPING FROM WELLS — CENTRIFUGAL PUMPS — ROTARY PUMPS—AIR LIFT PUMPS—HYDRAULIC RAMS.

(By J. Foster Petree, M.I.Mech.E., A.M.I.N.A.)

PUMPS.

The function of a pump is the raising and forcing of liquid, either by transferring it from a lower to a higher level or by performing equivalent work in maintaining pressure or overcoming resistances to flow. The work done is divisible into the three sections of suction, hydraulic losses, and delivery, all three being expressed in feet of 'head'; a column of fresh water 2·3 ft. in height representing a pressure at its base of 1 lb. per sq. in. The total resistance the pump must overcome to deliver the liquid, expressed in this way, is the 'total gauge head' (fig. 1). Liquid in motion has also kinetic energy, and this is convertible into head by the relation

$$h = \frac{v^2}{2g}$$

where h = the head in feet, v = velocity in ft. per sec., and g = the acceleration due to gravity, usually taken in pumping calculations as 32 ft. per sec. per sec. By transposing the equation into

$$v^2 = 2gh \text{ or } v = 8\sqrt{h}$$

the corresponding velocity can be obtained when the head is known.

SUCTION LIFT.

The height to which liquid may be forced depends upon the design of the pump and the power applied to drive it; but the suction lift is due solely to the pressure of the external atmosphere, and its maximum height depends upon the ability of the pump to produce a vacuum in the suction pipe. The theoretical maximum is 34 ft., this being the height of the water column corresponding to normal atmospheric pressure at sea-level; but air leaks, pipe friction, and the vapour pressure existing over any free liquid surface in the suction system, combine to reduce the practicable maximum to about 25 ft., although some reciprocating pumps have been known to prime themselves and draw steadily from a vertical depth of 30 ft. It is advisable to limit the vertical lift still further, to about 20 ft. for reciprocating pumps and 15 ft. for centrifugal pumps, unless all suction losses can be accurately predicted and a greater lift is found to be feasible.

As atmospheric pressure alone raises the liquid into a pump, any reduction whatever in the pressure difference between the inside of the pump and the outside air has a direct equivalent in feet loss of possible lift. Owing to the reduced atmospheric pressure, altitude above sea-level reduces the available lift by a foot for every 1,000 ft. of height, assuming the temperature to remain unchanged. Any rise in temperature, at any level, also reduces the possible lift at that level by promoting vaporisation, which creates a back pressure against the pressure of the atmosphere. The loss of lift from this cause, in the case of water, is about 2 ft. for every 10° Fahr. For this reason a pump dealing with hot water must be placed below the supply level to ensure a flow to the pump inlet by gravity. These limitations, due to natural laws, are independent of design.

SUCTION PIPING.

Good design can lessen hydraulic losses by avoiding such obstacles to free flow as sharp bends or sudden changes of section. As the frictional resistance in pipes varies approximately as the square of the velocity it can be diminished by using pipes of ample size; hence, suctions are often made larger than delivery piping, to limit the velocity to about 3 ft. per sec. for reciprocating pumps. This figure is often applied to centrifugal pumps also; but the absence of pulsations in a centrifugal pump suction makes a higher speed possible—5 ft. or 6 ft. per sec.—without increasing the total frictional loss, the higher surface friction of the pipe walls being offset by the reduced eddy-making. Pipes and castings should be water-tested for flaws or porous spots, and care taken in making and maintaining joints, to guard against air leaks. A very small leak will allow

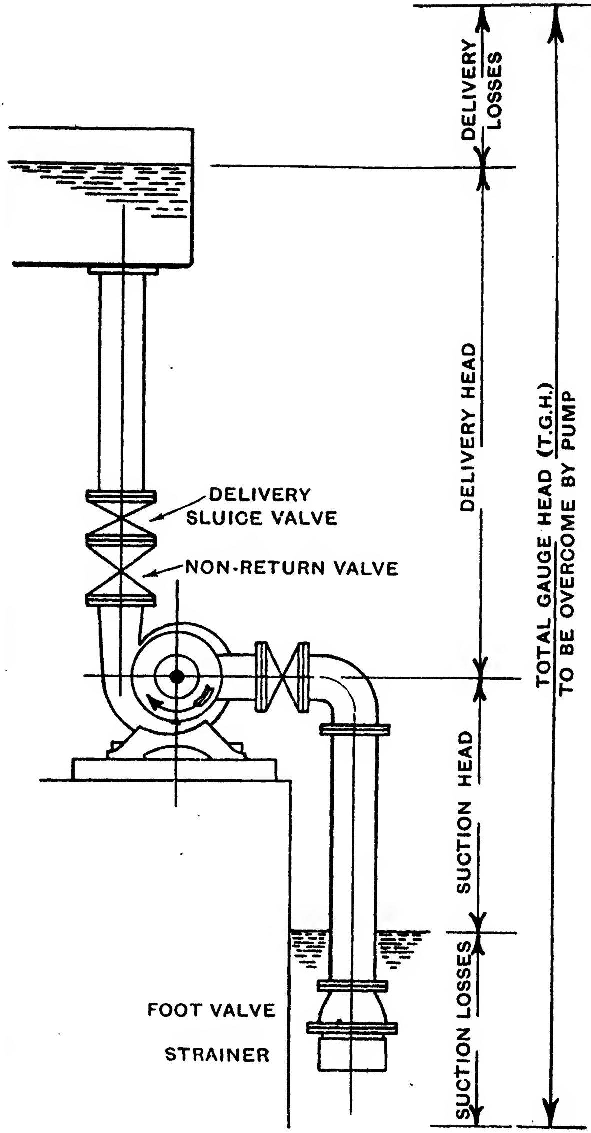
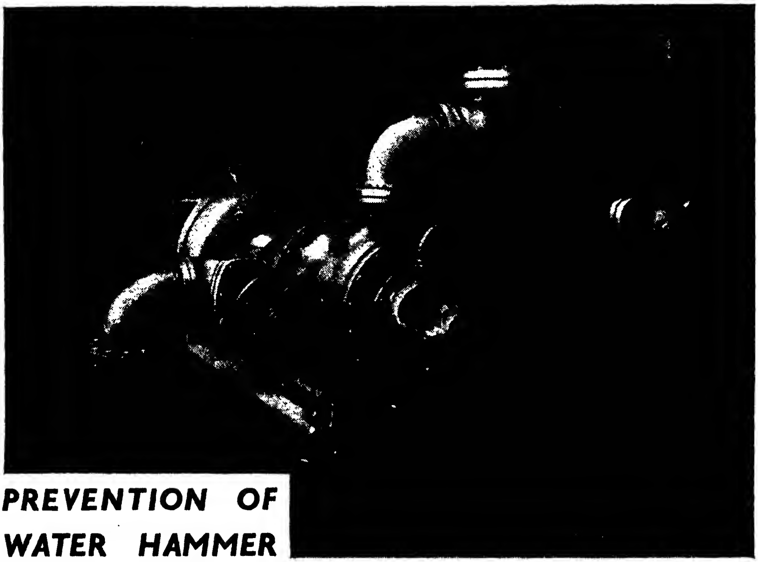


FIG. 1.—Work done by a Pump.



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the pipe to empty when the pump is stopped, even though a foot-valve may be fitted, as foot-valves are not readily accessible, and are seldom quite tight.

FOOT-VALVES AND STRAINERS.

The hinged flaps of foot-valves must have enough lift to give a clear passage at least equal to the area of the pipe. If a pump must be frequently stopped and restarted a full-bore sluice-valve may be fitted in place of the foot-valve, to ensure that the pipe remains full when the pump is not working. 'Shock loss' at entry to a suction pipe, due to the sudden change in direction of the water, can be almost eliminated by fitting a bell-mouth piece of suitable design (see R. W. Allen, *Proc. Inst. Mech. Eng.*, 1923). Suction strainers, in water that is normally clean, should have a total area through the holes at least three times that of the pipe. When pumping from rivers this proportion may be doubled, and a swivel bend or other means provided for raising the strainer to the surface for cleaning. For continuous operation of large pumps for river and sea-water (such as the circulating pumps of electricity stations), some form of revolving or traversing screen is necessary, cleaned by a mechanical scraper or a water spray, or both in conjunction. Ships plying in waters liable to ice have a live steam connection to the circulating intakes for melting accumulated ice. This device, believed to have been fitted first in the *Great Eastern* steamship, is also effective in dislodging jellyfish, jammed by the inrush of water against the suction gratings; or a connection may be made from the discharge of another pump and water under pressure pumped out through the choked grating.

Classification of Pumps.

Although one principle of operation blends into another in some border-line cases, the majority of pumps can be placed in one or another of the following categories:

Reciprocating Pumps, subdivided into (*a*) lift pumps, and (*b*) force pumps;
Centrifugal Pumps, comprising (*a*) volute pumps, and (*b*) turbine pumps;
Rotary Pumps other than centrifugal pumps; and
Air Lift Pumps.

Other means for raising liquids are the scoop-wheel; the chain pump, in which a series of discs on an endless chain lifts water up a pipe; the Archimedean screw—perhaps better classed as a conveyor than as a pump; the pneumatic or steam ejector and the steam injector; the hydraulic, or Montgolfier, ram (see p. 831) in which the kinetic energy of a large volume of water at a low head is used to raise a small quantity of the same water to a high head; and the Humphrey gas pump, in which the explosion of a gas charge above the surface of the water in a closed chamber forces some of it through the discharge valve, and the subsequent surging of the remainder admits a further supply, and also draws in and compresses a fresh gas charge, when the cycle is repeated.

Various forms of axial-flow pump have been developed for pumping from boreholes. Some are true screws and do not differ in principle from the Archimedean screw, but others are partially centrifugal in their action, and some are high-speed turbine pumps with diffusion passages for converting a part of the velocity of flow into pressure between the successive impellers.

The air lift, though styled a 'pump,' is in essence a water-chimney, and functions similarly to the chimney of a boiler plant by utilising the difference in density between the inside and outside of the rising main to produce upward flow.

RECIPROCATING PUMPS.

The simplest form of reciprocating pump, with a valve in the bucket only, is of limited utility, and needs no detailed description. A modernised form, often power-driven, is the portable diaphragm pump used by contractors for dewatering trenches. A non-return valve on the suction of the simple bucket-pump improves the regularity of performance without extending the working range; but if the top of the barrel is closed and a second non-return valve fitted on the delivery the scope is greatly extended, as the upward movement of the bucket will then lift the water to any desired height, limited only by the power applied and the strength of the parts. This type is still used in waterworks, and for extracting air and condensate from surface condensers. It is single-acting, and the rod is in tension during the working stroke. To give the effect of double-action two pumps are used, connected by a rocking beam or a crankshaft and rods so that the buckets act alternately.

The efficiency of this form of pump needs careful checking to detect leakage past the bucket valve; a drawback not present in the force pump, in which the reciprocating member may be either a packed piston working in a barrel or a plunger working through an external gland. The suction and delivery valves of force pumps are housed in chests with bolted covers, allowing of examination and renewal without disturbing the pumpwork. The plunger pump is slightly cheaper to make than the piston pump, as the chamber need not be machined internally. Piston pumps are either single-acting or double-actng, and are suitable for clear liquids and pressures up to about 250 lbs. per sq. in. The plunger pump, usually single-acting, is preferable where the delivery head is high or the liquid carries grit in suspension, as leakage past the plunger can be seen at once and the packing adjusted while the pump is working. Both types require an air vessel on the delivery, and if the driving speed fluctuates, on the suction side also, to prevent damage by shock. Some cushioning may be provided by air spaces formed in the discharge valve chest, but there must be no pockets in which air might collect in the suction system.

Power Pumps.

If a force pump has the piston or plunger actuated by a connecting rod from a crankshaft driven by external power through belt or spur gearing, it is known as a 'power pump.' The type is suitable for continuous duty over a wide range of heads, but not for conditions involving rapid changes in speed or quantity, as the speed of the prime mover cannot readily respond to varying conditions in the pump. For example, if the inflow to the suction does not completely fill the pump barrel severe shock results when the momentarily unloaded plunger strikes the water. If excessive pressure or the closing of a valve interrupts delivery the pump still runs at full power, and practically full speed, and may generate such a head as to burst a valve chest or break the teeth of the driving gears; whereas a direct-acting steam pump (q.v.) would merely stop when the total pressure on the plunger equalled the total pressure on the steam piston, restarting automatically as soon as the plunger load again fell below the steam load. The suction pipe of a power pump must run full under all conditions, and a relief valve should be fitted to limit the delivery pressure. The rate of suction inflow, therefore, determines the maximum piston or plunger speed at which the pump can be run without producing cavitation and consequent water-hammer. The cavitating speed can be calculated by Goodman's method (*Proc. I.Mech.E.*, 1903, p. 123).

Duplex single-acting power pumps have the cranks set at 180° apart, duplex double-acting power pumps at 90°, and triplex pumps at 120°. As lineshaft and motor speeds are seldom suitable for direct belt-driving, belt pulleys are fitted on a countershaft carrying a pinion which engages a large gear wheel on the end of the crankshaft. Another drive is from a pinion on the motor shaft through double-reduction gearing to the pump crankshaft; but this gear is liable to become noisy in use and lacks the resilience of the belt drive, shocks in the pump being transmitted through the train of gears to the prime mover unless a flexible coupling is incorporated.

Pump speeds in excess of 100 revolutions per min. of the crankshaft are rare unless the valves are mechanically operated, the usual rates ranging from 60 to 75 r.p.m. in medium-sized pumps (5 ins. to 9 ins. plunger diameter) down to 35 or 40 r.p.m. in pumps with 15 ins. plungers. Fuel pumps for oil engines and pressure pumps for oil-operated machine tools may run at speeds exceeding 1,500 r.p.m. of the driving shaft. Such pumps are of small size, have a short stroke and are actuated by cams or, preferably, by eccentrics. They have a drowned suction, very short and direct passages and, almost invariably, spring-loaded valves. Single-barrelled power pumps are used only in small sizes, *i.e.*, under 6 ins. diameter of plunger or piston. In larger sizes, and often for duties within the capacity of a single pump, the duplex or triplex types are preferred for their greater uniformity of flow. For very high-pressure hydraulic duty (4 tons or 5 tons per sq. in.) special pumps are made, having the barrels and valve chests machined from solid steel forgings.

Proportions of Branches.

The suction and delivery pipes of power pumps are usually of the same bore as the pump branches, no taper pipes being fitted. To minimise losses the suction diameter should not be less than two-thirds of the plunger or piston diameter, and in long-stroke pumps (*i.e.*, with a stroke/bore ratio greater than 1·0) may equal the plunger diameter. Pumps for hot or viscous liquids may have suction branches 1½ times the plunger diameter; but this adds greatly to the weight and cost of piping, and should not be necessary if an adequate positive head is maintained on the suction and the pipe system has no sharp bends or sudden restrictions. The table represents current practice in the suction and discharge velocities of power pumps of the principal types.

	Velocity in ft. per sec. of Suction.	Delivery.
Triplex single-acting plunger pumps	2 to 4	2½ to 5
Triplex double-acting piston pumps	2½ „ 4	3 „ 4½
Duplex double-acting piston pumps	3½ „ 5	5 „ 7
Simplex single-acting piston pumps	2 „ 3	2½ „ 3

The low speeds of flow adopted for the simplex pumps are due to the small average size of this type. In small pipes the friction loss is relatively greater than in large pipes, and as it varies as the square of the velocity, high flow speeds are detrimental to efficiency.

Approximate Calculation of Flow in Pipes.

The quantity in gallons per minute flowing through a pipe of diameter D ins. at a velocity of 1 ft. per sec. is given approximately by twice the square of the diameter ($2D^2$); but a quicker and more exact method on the slide-rule is to use the formula:

Gallons per minute at 10 ft. per sec. $= \dfrac{D^2}{0{\cdot}049}$, as follows:—

Set the 7 on Scale C to the pipe diameter in inches on Scale D. The right-hand 1 of Scale B then indicates on Scale A the quantity in gallons per minute at 10 ft. per sec.; and a movement of the cursor to the 9, 8, 7 or other value on Scale B (representing velocities in ft. per sec.) gives the quantities corresponding to those velocities on Scale A. Alternatively, setting the cursor to a known quantity on Scale A gives the speed of flow on Scale B. The formula is based upon Imperial gallons of 277·274 cu. ins.; to convert the quantities to U.S. gallons multiply by 1·2.

Capacity of Reciprocating Pumps.

If D = Dia. of pump barrel in inches, S = Stroke of pump in inches, N = Number of working strokes per minute, and G = Gallons per hour; then

$$G = \frac{D^2 \times S \times N}{6} \text{ approximately.}$$

Actually the result is 2 per cent. less than the volume swept by the pump piston; *i.e.*, it gives the output at a volumetric efficiency of 98 per cent. This is not an impossible efficiency, but 95 per cent. is probably the best obtainable in most cases. The remaining 5 per cent. is mainly leakage past the piston and gland. A small gland leakage may be tolerated as a seal against the ingress of air on the reverse stroke. A reciprocating pump with a long suction pipe may deliver, against a low head, more water than corresponds to the plunger displacement, *i.e.*, the volumetric efficiency may exceed 100 per cent., the momentum of the long water column, acquired during the suction stroke, holding both valves open after the delivery stroke has begun.

Discharge from Reciprocating Pumps.

Table of Constants which, Multiplied by Effective Number of Strokes per Minute, give the Discharge in Gallons per Hour.

Stroke of Pump, in Inches.	Diameter of Pump, in Inches.												
	2	2½	3	3½	4	4½	5	5½	6	7	8	9	10
1	0 6795	1·061	1·528										
1½	1·019	1·592	2·292	3·120									
2	1·359	2·123	3·057	4·160									
2½	1·698	2·654	3·821	5·200	6·795								
3	2·038	3·185	4·586	6·240	8·154	10·31							
3½	2·378	3·715	5·350	7·280	9·513	12·03	14·86	17·98					
4	2·718	4·247	6·115	8·320	10·87	13·74	16·98	20·54	24·45				
4½	3·057	4·777	6·879	9·360	12·23	15·45	19·11	23·16	27·50				
5	3·397	5·308	7·644	10·40	15·59	17·19	21·23	25·68	30·06	41·60			
5½	3·737	5·839	8·408	11·44	14·95	18·91	23·35	28·25	33·63	45·76			
6	4 077	6·370	9·173	12·48	16·31	20·63	25·48	30·81	36·68	49·92	65·20	82·52	
6½	4·416	6·901	9·937	13·52	17·67	22·35	27·60	33·39	39·73	54 08	70·63	89·40	
7	4·756	7·432	10·70	14·56	19·02	24·07	29·73	35·95	42·79	58·24	76·06	96·27	
7½	5·096	7·962	11·46	15·60	20·38	25·79	31·85	38·52	45·84	62·40	81·49	103·1	
8	5·436	8·494	12·23	16·64	21·74	27·50	33·97	41·09	48·91	66·56	86·93	110·0	135·8
8½	5·778	9·024	12·99	17·68	23·10	29·22	36·10	43·65	51·95	70·72	92·36	116·9	144·3
9	6·115	9·555	13·76	18·72	24·46	30·94	38·22	46·23	55·00	74·98	98·10	123·7	152·9
9½	6·455	10·08	14·52	19·76	25·82	32·66	40·34	48·79	58·50	79·04	103·5	130·6	161 3
10	6·795	10·61	15·28	20·80	27·18	34·38	42·47	51·35	61·12	83·20	108·7	137·5	169·8
10½	7·135	11·14	16·04	21·84	28·54	36·10	44·59	53·90	64·20	87·36	114·1	144·4	178·3
11	7·474	11·67	16·80	22·88	29·89	37·82	46·71	56·50	67·26	91·52	119·5	151·2	186·8
12	8·154	12.74	18·34	24·96	32·60	41·27	50·94	61·63	73·35	99·84	130·4	165·0	203·7
13	8·832	13·80	19·86	27·04	35·32	44·71	55·18	66·77	79·46	108·1	140·2	178·8	220·7
14	9·512	14·86	21·39	29·12	38·04	48·14	59·43	71·90	85·58	116·5	152·1	192·5	237·7
15	10·19	15·92	22·92	31·20	40·75	51·58	63·67	77·04	91·69	124·8	162·9	206·2	254·7
16	10·87	16·99	24·45	33·28	43·47	55·02	67·92	82·17	97·81	133·1	173·8	220·0	271·7
17	11·55	18·05	25·98	35·36	46·19	58·46	72·16	87·31	103·9	141·4	184·7	233·8	288·6
18	12·23	19·11	27·51	37·44	48·90	61·90	76·41	92·45	110·0	149·7	196·2	274·4	305·7

Mechanical Efficiency of Reciprocating Pumps.

As mechanical efficiency varies with the design of a pump and still more according to the standard of maintenance, only general figures can be given for reciprocating pumps as a class. Other things being equal, a piston pump is rather more efficient than a plunger pump owing to the large glands of the latter, embracing the full plunger diameter; and in all types a better efficiency accompanies increase in size of the pump. For small piston pumps 60 per cent. is a good figure, rising to 85 per cent. and even more in large waterworks pumping engines. Corresponding sizes of plunger pumps would have efficiencies from 3 to 5 per cent. less when pumping water, with a further reduction of at least 10 per cent. if pumping such liquids as tar or heavy crude oil. This must be taken into account when computing the boiler pressure for steam pumps.

Boiler Pressure required for Pumping.

A direct-acting steam pump with an 8-in. diameter steam cylinder and a 6-in. diameter pump barrel is to be used for pumping fresh water to a height of 100 ft.; the water being supplied to the pump suction at a pressure of 10 lbs. per sq. in. What working pressure will be required in the boiler?

The mechanical efficiency of a pump of this size, in good working order, is about 65 per cent. One foot head of fresh water represents a pressure of 0·433 lb. per sq. in., and the pressure to be pumped against is, therefore, $100 \times 0{\cdot}433 = 43{\cdot}3$ lbs. per sq. in., plus the pressure necessary to overcome fluid friction in the discharge pipe, with all losses at bends, valves, etc. Under ordinary conditions the total pipe loss would not exceed the equivalent of 10 lbs. per sq. in., and, assuming this value, it can be cancelled by the specified pressure of 10 lbs. per sq. in. on the suction. The net resistance to movement of the pump piston is, therefore, the amount due to the delivery head alone, acting on the area of the 6-in. piston, and its value is $(6 \times 6 \times 0{\cdot}7854) \times 43{\cdot}3 = 1{,}224$ lbs. The mechanical efficiency being 65 per cent., the total resistance to be overcome by the steam piston is $\frac{1{,}224}{0{\cdot}65} = 1{,}883$ lbs. To overcome that resistance the effective pressure that must be exerted on a direct-acting steam piston 8 ins. diameter would be $\frac{1{,}883}{(8 \times 8 \times 0{\cdot}7854)} = 37{\cdot}46$ lbs. The boiler pressure must equal the effective pressure on the piston plus the loss of steam pressure due to throttling in the steam ports and friction in the pipe. Allowing for a back pressure of 4 lbs. per sq. in., and assuming a reduction by throttling of 10 lbs. per sq. in., the actual pressure in the boiler must be at least $37{\cdot}46 + 4 + 10 = 51{\cdot}46$ lbs. per sq. in. As boilers seldom work continuously at the full designed pressure, a boiler rated at 60 lbs. per sq. in. would be selected for this duty.

Direct-acting Pumps.

Steam reciprocating pumps are described as 'direct-acting' when the steam piston and pump piston are both mounted on the same rod so that the delivery pressure in the pump is produced by the direct thrust of the steam pressure without intervening gears or linkwork. Both pistons are double-acting, and the pumps may be either horizontal or vertical. They range from small pumps delivering three or four gallons per minute, to big triple-expansion units for waterworks pumping, fitted with Corliss or drop valves to give a sharp cut-off and a high degree of expansive working in the steam cylinders.

In a simplex direct-acting pump the valve of the steam cylinder is operated by a swinging link with one end pivoted on the pump framing, the other end being attached to the reciprocating pump rod. Duplex pumps have two steam cylinders and two pump barrels side by side, the valve of either steam cylinder being actuated from the rod of the other pump. If simple valves are used they have no lap or lead, the initial steam pressure being carried for the full stroke; this ensures that one steam port is always open so that the pump cannot become stalled by the valve sticking when both ports are covered, but involves a somewhat heavy steam consumption. The valve is not rigidly attached to the valve rod, but can slide upon it by a predetermined amount; thus the stroke of the rod exceeds the travel of the valve by the amount of 'lost motion,' and the valve pauses for a time at the end of each stroke, allowing the full steam pressure to follow the piston almost to the other end of the cylinder.

A simplex pump is made to work expansively by using an auxiliary or pilot valve operated directly by the valve rod and admitting steam to a piston which forms the main valve of the steam cylinder. Where automatically variable delivery is required, as in boiler feeding, a float tank is added, the vertical movement of the float controlling the steam supply. (Centrifugal feed pumps are treated under 'Turbine Pumps,' p. 814.) For industrial uses the horizontal pump is favoured, but on board ship the vertical type is preferred owing to its compactness and, in addition to boiler feed-water supply, is used for fuel and lubricating oil, bilge, fire, ballast, and general services.

Materials of Pump Ends.

Iron pump ends, with steel rods and iron or bronze fittings, are used for pumping asphalt, ammonia, tar, potassium cyanide, drainage water, sewage, and paint; with all parts in contact with the liquid made of iron, for naphtha, creosote, gas-works pumping, soda pulp, caustic soda, and alkaline liquids in general; lead-lined throughout, or with lead-lined barrel and lead covers and valves, for concentrated acids; glass-lined, for milk.

Bronze pump ends, or iron with bronze lining, are used for wood pulp and paper 'stuff,' alcohol, whisky, molasses and sugar syrups, boiler-feed water, drinking water, tan liquor, beer, salt water, vinegar, fuel oil and lubricating oils, and concentrated nitric acid; with all parts in contact with the liquid of bronze, for acid mine water, chlorate of lime, and dilute acids. Stainless steel and various nickel alloys are now extensively employed in pumps for corrosive liquids, liquid food-stuffs, etc.

Leather Packings.

A cupped leather ring or U-leather forms a self-tightening packing much used for rod glands and pump pistons subject to high pressures. It is effective if merely inserted in a groove of square section, the water pressure inside the ring forcing it against the rod or cylinder wall; but the constant bending at the fold as the pressure is applied and relaxed causes rapid failure unless some support is provided, and the life may be only a week or ten days. A metal supporting ring within a U-leather gives long life, and a further improvement is a shaped gland or backing ring (usually of gunmetal if separate from the gland) to fit over the curved outer surface. The inner

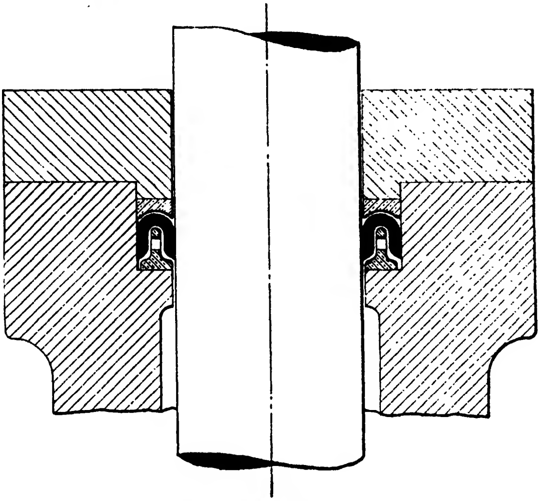

FIG. 2.—U-Leather Packing.

ring should not fill the leather, and should be drilled with holes to ensure a free flow of the pressure water. (See fig. 2.)

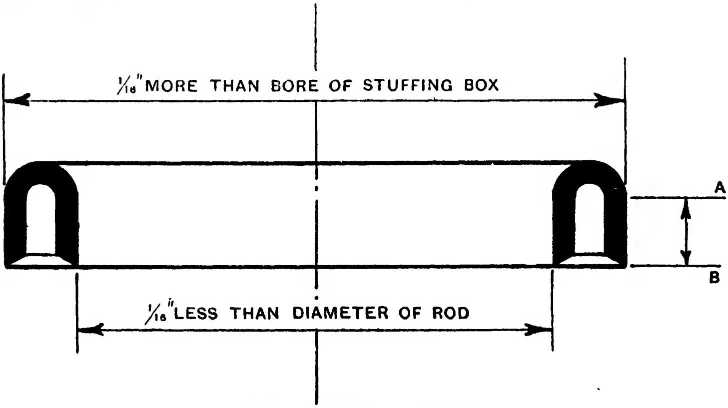

FIG. 3.—Section of U-Leather.

Leather packings wear badly in dirty or gritty water, and salt water, even though clean, has the same effect, as it corrodes and pits machined iron or steel surfaces. Intermittent working demands absolutely clean fresh water, as grit may lodge between the leather and rod during idle periods, causing serious scoring when pressure is again applied. Wherever possible sufficient pressure should be maintained to keep the leather expanded in the gland at all times. To ensure close fitting the inner diameter of a new leather should be slightly less than the rod diameter, and the outer diameter of the leather slightly more than the bore of the stuffing-box. (See fig. 3.)

Lubrication of Leathers.

The working conditions should be stated when ordering leathers, as high- and low-pressure packings may be tanned by different processes, calling for different lubricants. The makers' recommendations should, therefore, be invited, and followed. Leathers in store should be dressed periodically with the recommended oil (neatsfoot and sperm oils are commonly used) to keep them pliable, and may with advantage be kept on wooden formers rounded to fit the inner space of the ring. They should not be hung on pegs, nor nested one on top of another. Immediately before fitting, tallow or oil should be thoroughly worked into the fibres by hand. *Mineral oil speedily rots leather and must never be used.*

A preservative dressing which will also counteract slight roughness of the metal surfaces is known as Entwistle's lubricant, and contains 50 parts stearine or tallow, 20 parts soapstone or powdered mica, 2 parts graphite, 10 parts neatsfoot or other suitable oil, 10 parts powdered zinc, lead, copper or other soft metal, and 10 parts oil of tar. By the reciprocating motion of the rod the particles of metal and graphite are distributed over the surface and fill the pores and minute cavities, so producing a smooth finish.

Friction of Leathers.

The friction of leathers involves so many variables and is so much affected by some of them, *e.g.*, any eccentricity of loading, that no formula can give an exact value. The various formulæ, based on experiments with hydraulic rams, give results which at best are only approximate, and may differ widely. That of John Hick, based on elaborate tests with rams ½ in., 4 ins., and 8 ins. diameter, at pressures up to 6,000 lbs. per sq. in., has the form

$$F = C \times DP$$

where F = total friction in lbs., D = diameter in ins., P = pressure in lbs. per sq. in., and the coefficient C may vary from 0·03 for a well-lubricated leather in good condition, to 0·047 for a new and stiff, or badly-lubricated leather. Later tests by Tuit and by Goodman gave much higher frictional losses than Hick's, and Prof. Goodman, from his own and the previous experiments, developed the formula

$$F = 0{\cdot}08P + C$$

where F = frictional resistance in lbs. per sq. in. of water pressure, P = water pressure in lbs. per sq. in., and the constant C, based on the diameter D ins. of the rod or piston, has values from $\frac{100}{D}$ to $\frac{250}{D}$ for leathers in good and poor condition respectively. This formula gives a friction of the order of 10 per cent. of the total pressure. Experiments by Martens (Germany) and Davis (U.S.A.) on hydraulic testing machines indicate 5 per cent.; but the earlier tests of Tuit (*Engineering*, June 15, 1888) ranged as high as 18·8 per cent., even with well-lubricated leathers and the moderate pressure of 284 lbs. per sq. in. Two leathers were used in this test, however, and therefore it appears that 10 per cent. loss is a reasonable approximation.

The friction occurs almost wholly at the point A (fig. 3) and is little affected by varying the depth A–B. Increased depth does not make the packing more effective, and there is a practical benefit in a shallow form; as the ring is moulded under pressure from a flat sheet and deep drawing, by straining the fibres at the bend, may lead to early failure by cracking.

Pumping from Wells.

For wells under 30 ft. a hand pump with a 4-in. barrel and a stroke of 9 ins. or 10 ins., delivering about 24 gallons per minute, is as large as one man can conveniently work. For wells of 30 ft. to 70 ft. depth, a 3½-in. barrel may be fitted. For wells over 70 feet deep, three 3-in. barrels are preferable, worked by rods from a three-throw crank driven through a spur wheel and pinion.

Deep-Well Pumps.

In deep-well pumping the pump must be placed at or below the rest level of the water. A reciprocating borehole pump comprises a gunmetal barrel about 1 in. less in diameter than the well lining tube, screwed upon a rising main about ½ in. bigger in diameter than the barrel, so that the bucket and foot-valve can be drawn up for repair; and a pump rod carrying a gunmetal plunger, which passes through a gland in the top of the pump barrel. Alternatively a bucket pump may be used, the rising main being a continuation of the barrel to the surface. The foot-valve is contained in a heavy gunmetal casting resting at the bottom of the lining tube, and has a projection above the guard on to which the bucket can be screwed in order to lift the valve to the surface if required.

It is usual to balance all the weight suspended from the crankpin, plus one-half of the work done in lifting the column of water. If there is a considerable lift above the surface, this can be done in part by a differential plunger at the surface. Other methods employ a weighted lever or a weight sliding in guides and attached by wire ropes passing over pulleys, to the crosshead. Pump rods may be of steel, or of pitchpine (to reduce weight), coupled by wrought iron plates. The crankshaft is driven either by spur gearing from a countershaft fitted with fast and loose belt pulleys, or by worm gearing and motor. If electric motor or oil-engine drive is fitted, it is desirable to provide a clutch for use when starting. Worm gearing should be completely enclosed in an oil bath; and a similar casing may be used with advantage for spur gearing.

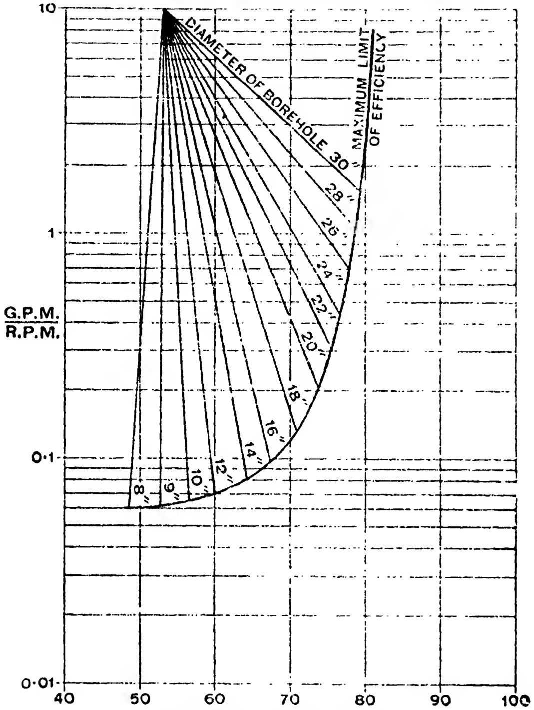

FIG. 4.—Influence of borehole diameter on pump efficiency.

Plants of these types will deliver up to 100 gals. per min. from wells as deep as 500 ft. For larger deliveries the diameter of the well must be increased, at some extra capital cost, which is a charge on the water raised. Instances are some boreholes of the South Staffordshire Waterworks, 30 ins. diameter, which yield 790 gals. per min. (See paper by F. J. Dixon, *Trans. Inst. Water Engrs.*, 1920.)

CENTRIFUGAL AND AXIAL-FLOW BOREHOLE PUMPS.

For deep wells the centrifugal borehole pump is a multistage unit driven at high speed by a motor on the surface. As the spindle is vertical there is no deflection to restrict the number of impellers. The eye of the impeller is large in proportion to the diameter over the vane tips, and the vanes are, therefore, necessarily short, so that the head per stage can seldom exceed 20 ft.; hence 15 or 20 impellers may be required for a deep bore. The bearing in the suction end, below the pump, usually consists of lignum vitæ strips, lubricated by a small supply of water from the bore. If the water is gritty a rubber sealing ring is also fitted, to ensure a pressure inside the bearing sufficient to prevent grit being carried in. The intermediate shaft bearings are of similar type, and supported by spider rings held between the flanges of the rising main, with which the shaft is concentric. The span between bearings is usually from 7 ft. to 10 ft.

The larger the bore the more efficient can be the design of the pump; but as the size of the pump is increased, so also is the diameter of the shafting, with a consequent increase in the frictional loss due to the presence of the shaft and bearings within the rising main. This loss also increases at higher rates of revolution, so that each design represents a compromise between the various factors. Fig. 4 illustrates the influence of borehole diameter upon pump efficiency, and fig. 5 the frictional loss for various shaft diameters and speeds: these diagrams being taken from the paper by Sherwell and Pennington on 'Centrifugal Pump Characteristics,' *Proc. Inst. Mech. Eng.*, 1933. Reference may also be made to 'Modern Methods of Raising Water from Underground Sources,' by R. S. Allen and W. E. W. Millington, *Proc. Inst. Mech. Eng.*, 1931, where particulars are given of oil engine and belt drives through bevel gearing in conjunction with high speed centrifugal borehole pumps.

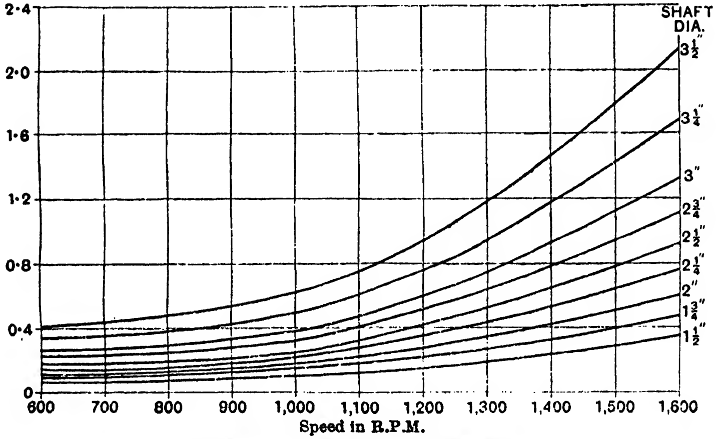

FIG. 5.—Friction of borehole pump shafting.

Suction strainers, if fitted, should have an area through the holes of ten or twelve times the area of the suction orifice. Foot-valves are seldom necessary, and may be undesirable if a strainer is provided and the water carries sand in suspension, as they prevent the backwash through the strainer when the pump stops, which is of value in clearing the strainer holes. Ample headroom is required in the building housing the plant, for initial assembly and subsequent inspection; and provision for attaching lifting eyes must be made, as the pump is assembled unit by unit while suspended over the borehole.

CENTRIFUGAL PUMPS.

The centrifugal pump acts by the effect of centrifugal force on the mass of water rotating at any instant in the disc or impeller (sometimes also called the 'wheel' or 'runner'). Radial flow is thus caused and the water leaves the impeller in the relative direction of the exit angle of the vanes, with a velocity compounded of the radial velocity and the velocity of rotation of the tips of the vanes (see fig. 15). The absolute velocity of the water is outwards, and in the direction of rotation, and part of the energy due to the water velocity is converted into pressure at the outlet from the casing.

The two main types are the Volute Pump (fig. 6), in which the impeller discharges directly into a spiral casing of uniformly increasing cross-section; and the Turbine Pump, in which a ring of tapering guide passages surrounds the impeller, reducing the water velocity and increasing the pressure until discharge takes place either to the inlet of the next impeller or to the delivery branch. Turbine pumps have usually two or more stages, as a volute pump can be designed in most cases to suit conditions that a one-stage turbine pump could cover, and is cheaper to make.

Centrifugal pumps were formerly thought suitable for low heads only, but developments during the present century have so improved the efficiency and widened the scope that they are now made for practically every water-pumping duty undertaken by reciprocating pumps, up to pressures exceeding 1,200 lbs. per sq. in. They are also used for pumping light oils, many chemicals, milk, meat extracts cement slurry, sewage, paper pulp, gravel, and even small coal; but are not suitable for dealing with liquids of a viscosity much in excess of that of water. Centrifugal pumps for crude oils have been made, but highly viscous liquids are handled more effectively by reciprocating pumps or some form of positive-displacement rotary pump. (See p 824.)

Advantages.

1. Uniformity of flow with no reversal of direction; permitting higher pipe speeds and therefore smaller and lighter pipework.

2. Compactness; leading to a much reduced building cost as compared with reciprocating pumps of equal output, in the larger sizes.

3. Good efficiency at high rotative speeds, permitting direct drive by steam turbine, internal combustion engine or electric motor of moderate size.

4. Silent operation, and freedom from vibration.

5. Low maintenance cost.

Volute Pumps.

Casings.

The casing of a volute pump is designed to maintain a uniform rate of flow all round the impeller, which, for some duties, may project into it; this construction being adopted when a wholly external volute would unduly increase the size, weight, and therefore the cost, of the pump. The casing may be of single- or double-suction type, in conjunction with either open or shrouded impellers. In the single-suction design the whole of the water pumped enters through one side of the impeller. It is cheap to make, and is, therefore, preferred for pumps of the smallest sizes, in which the unbalanced thrust on the impeller is too small to need special attention; also for vertical-spindle pumps of larger size for well, dock, and sewage pumping, as a short, straight inlet pipe can be fitted.

For horizontal-spindle pumps above 6 ins. diameter branches the double-suction type, which is inherently in hydraulic balance, is almost universal. It is also used for some vertical-spindle dock and sewage pumps of large size owing to the compact form of the casing. Double-suction

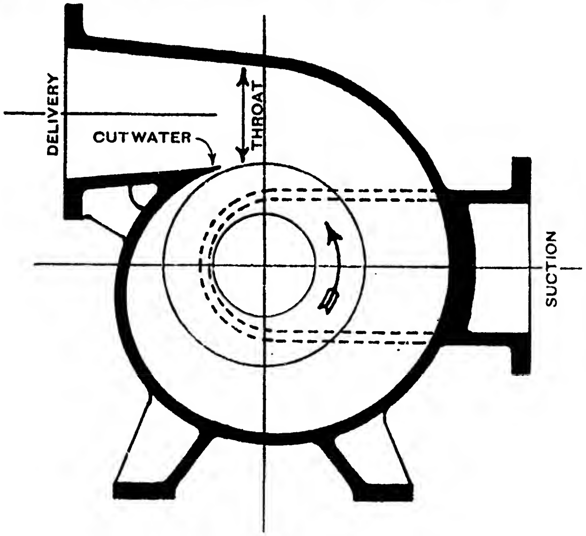

FIG. 6.—Section through casing of volute pump.

casings may have either two removable end covers, extended to form bearing brackets (as in fig. 7), or a bolted joint on a plane through the axis of the spindle, usually on the horizontal centre-line. If the casing is split horizontally it is desirable to arrange both suction and delivery branches below the joint so that the top half of the casing can be lifted without breaking any pipe joints. Two volute pumps are sometimes used in series when the quantity is too great for a turbine pump to handle economically, or the speed of the prime mover is low for a single impeller. For low heads and large quantities two low-lift pumps may be combined in a single casing with two volutes to form a 'twin parallel' pump, and there are instances of three impellers in parallel, although these are not numerous.

Casing Details.

The volute sections are proportioned on the basis of the cross-sectional area at the 'throat' (see fig. 6), which is designed to pass the specified quantity at a velocity varying from $0 \cdot 3\sqrt{H}$ to $0 \cdot 5\sqrt{H}$ ft. per sec., where H = total head in feet. Half-way round the spiral the area is approximately half the throat area, and at intermediate points in proportion. The taper is usually continued out to the delivery branch to convert more of the velocity energy into pressure, and may be carried further by a separate taper pipe. The taper in the pipe should be gradual, to minimise eddy losses; 1 in 8 on the diameter being a usual figure. (See also Prof. A. H. Gibson, 'The Design of Volute Chambers,' *Proc. Inst. Mech. Eng.*, 1913; and papers by the same author on flow through taper pipes and passages, *Trans. Roy. Soc. Arts*, 1910, and *Trans. Roy. Soc. Edin.*, 1911.)

Supporting bearings should be outside and separate from the casing, no part of the revolving weight being taken by the glands; but steady bearings for the impeller are formed in the casing around the 'eye,' or inlet, of the impeller. As the rubbing speed at these points is high, renewable rings are usually fitted in the casing, and often on the impeller as well. Wear also takes place where the discharge from the impeller impinges on the edges of the volute, especially if the water carries sand in suspension, and renewable rings may also be fitted at this point. They are always advisable in gunmetal casings.

The guiding principle in designing casing passages is to avoid sudden changes in the velocity or direction of flow, the water being accelerated uniformly into the eye of the impeller in a smoothly curving path, and the flow at exit from the impeller reduced in speed with the minimum turbulence to the point where it enters the rising main.

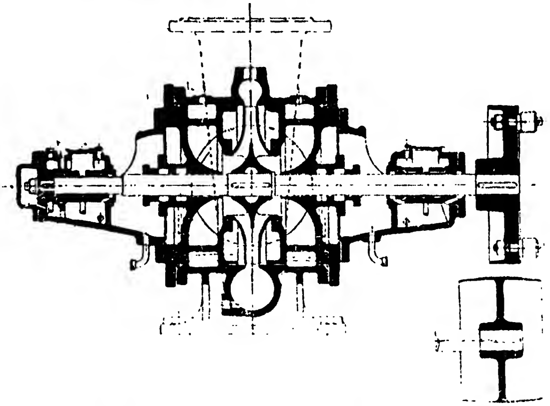

FIG. 7.—Section of volute pump, showing bearing brackets.

TURBINE PUMPS.

In turbine pumps a 'diffuser,' or ring of tapering guide passages, takes the place of the volute; the passages reducing the velocity and increasing the pressure of the water before leading it into the eye of the next impeller, or, after the final stage, into the delivery end cover. Most diffuser arrangements embody a tangential outward flow, the return to the next impeller being either spiral or radial. (See A. E. L. Chorlton, 'Construction of Turbine Pumps,' *Proc. Inst. Mech. Eng.*, 1917.) Turbine pumps have been made with ten stages in the one casing, but a practical limit is set to the number of stages by the unsupported length of the spindle between bearings; the intermediate bushes, eye rings, etc., being designed to prevent inter-stage leakage and not as supports for the rotating parts. Hence, eight stages is the usual maximum, and if the specified head requires more the pump is divided into two sections with the driving motor between. In such a case the motor bedplate is often extended to one side, with machined slides on which the motor can be withdrawn without disturbing the pumps.

Casings and Details.

Although horizontally-divided casings find some favour in America and Germany, most British makers prefer either the ring-type construction (as in fig. 8) or the 'cylinder' or 'drum' type (fig. 9); some controversy existing as to their respective merits, though the advantag appears to rest with the ring type. In the drum type the series of stages, each unit comprising impeller, guide ring, and perhaps a separate return ring, is inserted in succession into an outer cylindrical shell, closed at the ends by the suction and delivery covers. The assembly is neat in appearance, but has the drawbacks that there is no check on the tightness of the joints between stages, and internal rusting of the casing may make withdrawal of the parts very difficult; whereas the ring type of casing is easy to assemble and take apart, and any leakage is evident at once.

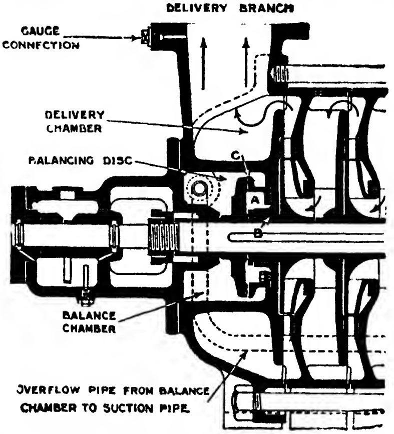

FIG. 8.—Delivery end of turbine pump (ring-type casing) showing hydraulic balancer.

The single-inlet impeller (type A, fig. 14) is almost standard practice in turbine pumps, but as it is not in axial balance the tendency to move towards the suction end must be counteracted. This is partly done by a bearing ring on the back of the impeller, which limits the area exposed to delivery pressure. Holes drilled through the impeller are intended to equalise the pressures on both sides within the ring, but the device is only partly effective and some additional means must be adopted in high-lift pumps. The most usual is the hydraulic balancer shown in fig. 8,

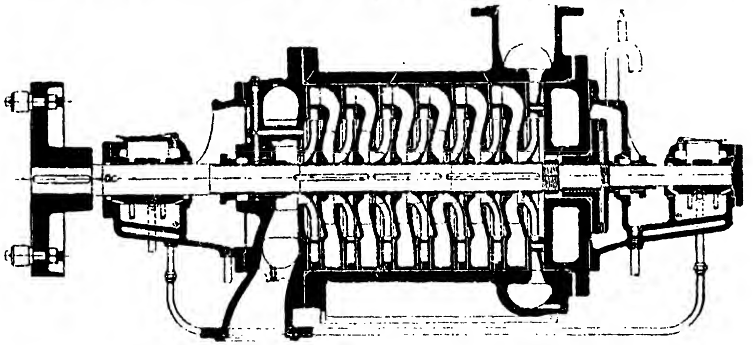

FIG. 9.—Section of multi-stage turbine pump (drum-type casing).

comprising a disc secured to the spindle within the delivery end cover, where it rotates with a small axial clearance over the faced end of a chamber A, to which pressure water is led through a small external pipe from a suitable stage of the pump, or through the clearance B. As the spindle moves towards the suction end pressure builds up behind the disc and forces it towards the delivery end; this increases the running clearance C and allows the pressure water to escape until equilibrium is restored.

Prof. Gibson has shown (*Trans. Roy. Soc. Arts*, 1910) that there is least hydraulic loss in tapering channels when the angle between the walls is about 11°; but this is difficult to arrange without undue size of casing, and 15° is the usual compromise. Velocities between stages are high and smooth surfaces, therefore, essential, gunmetal or bronze diffuser sections being used in iron casing rings. Impellers are of gunmetal, bronze, or Monel metal, the bosses forming a continuous sleeve on the spindle; with two keys, at 180° apart, in the larger sizes, to assist dynamic balancing. For high lifts the delivery cover is of cast steel; suction covers may be cast iron or cast steel. Small pumps may be of bronze throughout. For pumping corrosive mine water cast iron covers are sometimes lined in part with bronze.

Turbine Pumps for Boiler Feeding.

Turbine pumps are now used to feed boilers at all working pressures up to 1,250 lbs. per sq. in., and with feed water temperatures up to 400° Fahr. The duty calls for special design to ensure stable performance, as the delivery pressure must be kept constant almost irrespective of the steam demand on the boilers. The specified continuous overload rating of the boiler governs the pump design, and to prevent excess pressure in the feed range at light loads some control of pump speed is required. In motor-driven sets this is often obtained by some form of hydraulic coupling. Feed pumps are provided in duplicate, either set being able to meet the full steam demand. Some engineers still prefer reciprocating pumps as a stand-by, as they give positive displacement of the water and will either force open a sticking check-valve or come to a stop, which would at once be noticed; whereas a centrifugal pump would continue to run although no water was passing. It is now quite usual, however, to rely entirely on centrifugal sets; half of the pumping plant being motor-driven, and half by steam turbines, as a safeguard against an electrical breakdown. The steam-turbine drive lends itself to sensitive automatic regulation, the high speed promotes efficient turbine operation, and the exhaust can be used directly for feed-heating. The turbine governor responds to both steam pressure and boiler load variations and usually acts by controlling auxiliary steam nozzles. An overspeed governor is fitted to shut the set down completely in such an event as a burst pipe or failure of the water supply to the pump. The compactness and economical working of these sets are strong points in their favour, especially for large evaporations.

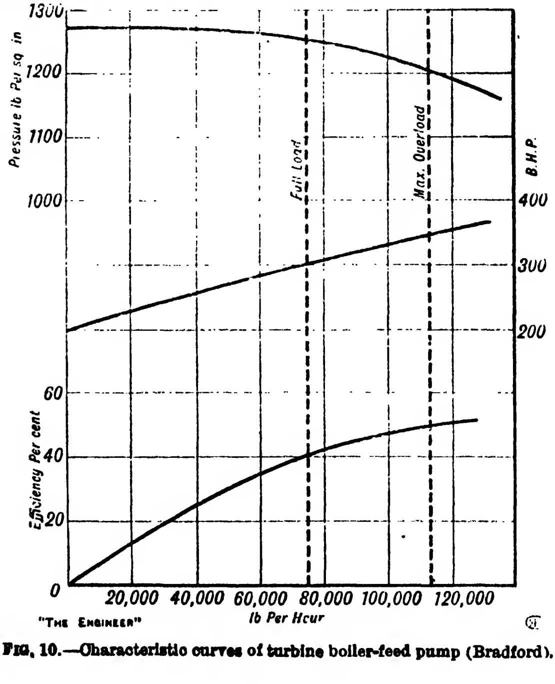

Fig. 10.—Characteristic curves of turbine boiler-feed pump (Bradford).

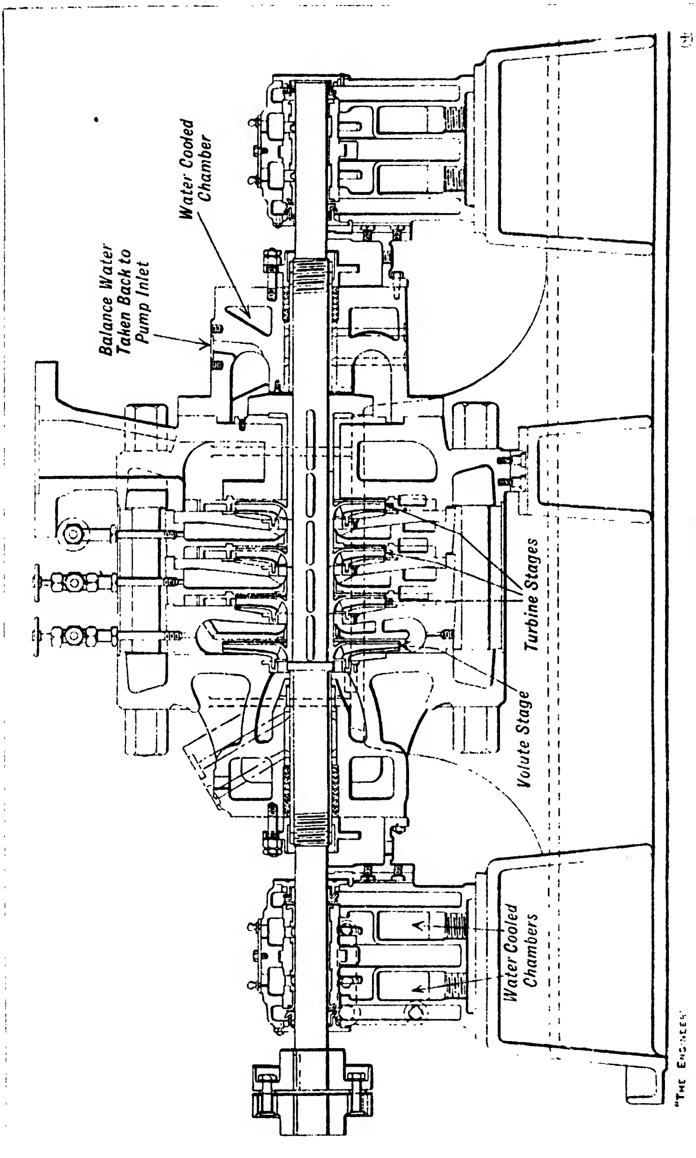

Fig. 11.—Section of 4-stage boiler-feed pump (Bradford).

Fig. 10 shows the characteristic curves of a turbine boiler-feed pump at Bradford (Yorks.) electricity station, delivering 75,000 lbs. per hour against 1,250 lbs. per sq. in. pressure, with an overload rating of 112,500 lbs. per hour at 1,200 lbs. per sq. in. One set, electrically driven at 2,950 r.p.m., comprises a 2-stage low pressure pump and an 8-stage high pressure pump. The other unit is steam turbine-driven at 4,000 r.p.m., and consists of a single-stage low pressure pump and a 4-stage high pressure pump (fig. 11). For full description see *The Engineer*, Oct. 23, 1931.

Impellers.

Impellers may be either single-inlet or double-inlet, having in each case either open-type vanes supported by a single disc, or the vanes enclosed between two side plates or 'shrouds.' The applications of the single-inlet and double-inlet arrangements are described in discussing the pump casings (p. 811).

The open-type impeller is cheap, easy to mould and machine, and is not liable to clog with debris, being self-clearing to a considerable extent; but the side friction loss is high if the running clearance is small, and the leakage from delivery back to suction is serious if the side clearance is increased to avoid the friction. Hence this type is not suited to high heads or speeds.

The shrouded impeller is a stronger construction, but is less simple to make and can only be machined externally. Internal finishing of the passages must be done by hand with file and emery cloth, and very little is practicable in small impellers. Side leakage is almost eliminated by the shrouds, but disc friction, which varies approximately as the cube of the revolutions and the fifth power of the diameter (see Gibson & Ryan, *Proc. Inst. C.E.*, 1910) becomes important at high speeds and may absorb 10 per cent. or more of the power input. Fibrous matter in the water soon chokes a shrouded impeller unless the number of vanes is so far reduced as to impair the efficiency, *i.e.*, to less than six. Some impellers, for gravel-pumping and similar duties, have only one or two passages and will pump anything that can pass through the suction pipe; but the efficiency is then usually low.

The vane angles at inlet and outlet are designed to pick up the water at the entrance or 'eye' and deliver it to the volute or guide passages with minimum loss. Some loss by 'shock' is inevitable, especially at inlet, where the water strikes the vane at a different angle for every change in operating conditions. The impeller is, therefore, designed to give optimum performance under one set of conditions, known as the 'normal,' and 'percentage characteristic curves' are plotted from which the performance can be estimated for other loads and speeds. (See p. 817.)

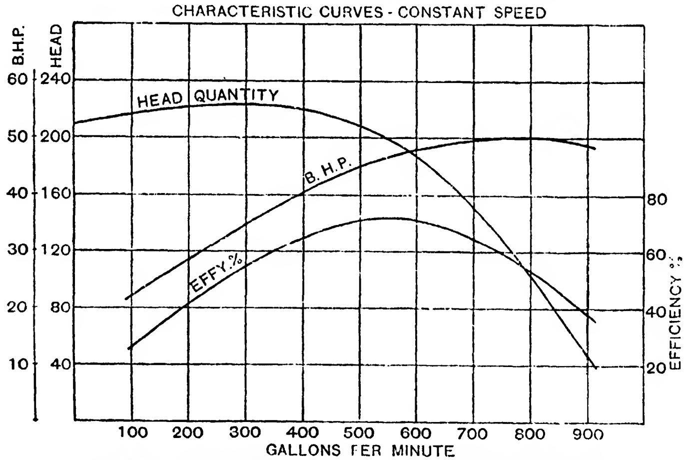

FIG. 12.—Constant speed characteristic curves of a centrifugal pump.

Characteristic Curves of Performance.

The variations of head, efficiency and power absorbed, plotted at constant speed on a basis of varying quantity, form the usual 'characteristic curves' of a centrifugal pump. Fig. 12 shows a typical set, from a pump designed to deliver 550 gals per min. against a head of 200 ft. with an efficiency of 72 per cent., taking 46·5 b.h.p. at the pump coupling. If the actual speed differs from the assumed constant speed of the design the head and output are also affected, but can be corrected to the constant speed values by applying the rules:

Head varies directly as the square of the r.p.m.
Quantity varies directly as the r.p.m.
B.H.P. varies directly as (r.p.m. × Head).

The percentage variation curves are constructed in this way from the constant speed performance curves, the abscissæ being varying quantities expressed as percentages of the normal designed quantity, and the head, efficiency and power similarly as percentages of the normal values. Thus in fig. 13, plotted for the same pump as fig. 12, the point marked '100–100' represents 550 gals. per min., and 200 ft. head on the head-quantity curve; 72 per cent. on the efficiency curve; and 46·5 b.h.p. on the power curve. To take an example illustrating the use of the curves. What would be the performance of this pump at a head of 170 ft.?

The new head of 170 ft. is 85 per cent. of the normal head of 200 ft., shown on the curve at 100; and reference to the Head-quantity line shows that the capacity at 85 per cent. head has increased to 119 per cent. of the normal, or 1·19 × 550 = 655 gals. Referring now to the Efficiency curve at the point corresponding to this increase in capacity, it is seen that the efficiency will be 95 per cent. of the normal, or 0·95 × 72 = 68·4 per cent. From the b.h.p. curve it is found that the power required is 106 per cent. of the normal, or 49·5 h.p.

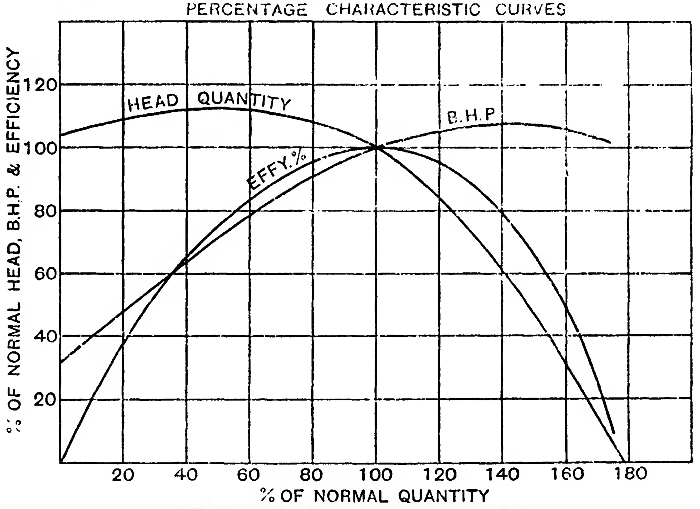

FIG. 13.—Percentage characteristic curves.

IMPELLER PROPORTIONS: SPECIFIC SPEED.

The proportions of impellers for different duties may be compared in various ways to bring out the effect of different features, the efficiency obtainable for a given duty depending mainly upon the relationship between output, head and speed of rotation. Comparison by the ratio of the mean radius at the vane inlet to the radius at the vane tip (R_1/R_2) touches only head and speed, without reference to output; and the ratio D/B, where D = outer diameter of impeller, and B = width at tip, is of limited utility without other particulars. The most useful expression for combining the effects of changes in output, head and speed is the 'Specific Speed,' denoted by the symbol N_s. This must not be confused with the 'specific speed' of a water-turbine; it is not a measure of movement, but a number indicating the 'type' of the impeller, derived by applying the principle of similarity to impellers which are geometrically identical, differing only in the ratio of their corresponding linear dimensions.

$$\text{Specific speed } (N_s) = \frac{N\sqrt{Q}}{H^{\frac{3}{4}}}$$

where
N = speed of rotation;
Q = output;
H = total head per impeller.

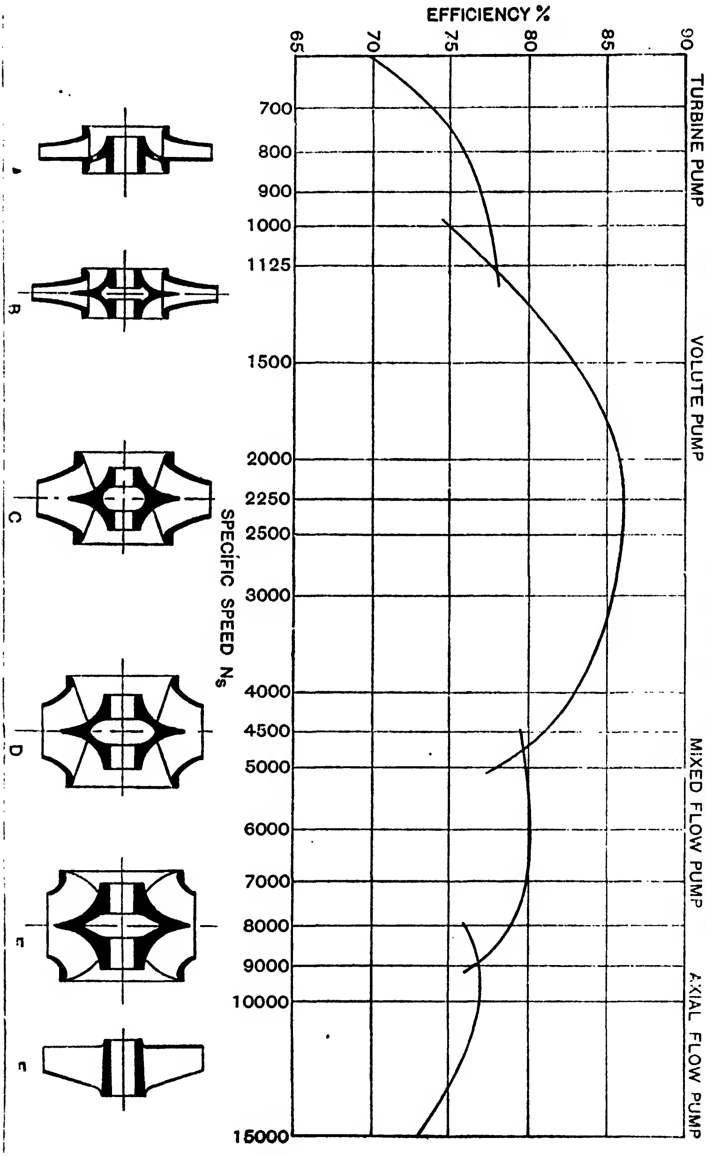

FIG. 14.—Relation between specific speed and efficiency of centrifugal pumps.

In British practice N is usually taken in revolutions per min., Q in gals. per min., and H in ft.; but other units may be used, and it is, therefore, important when comparing values of N_s from different sources (*e.g.*, in comparing a British and an American pump) that the units should be known so that necessary corrections can be made. If Q is taken in gals. per min., the values of N_s are 19·34 times as great as those obtained from a Q in cub. ft. per sec.; if Q is in U.S. gals., N_s is 21 18 times the value for Q in cub. ft. per sec.

In evaluating N_s on a slide-rule the denominator $H^{\frac{3}{4}}$ can be obtained as the fourth root of H^3. The table below gives values of $H^{\frac{3}{4}}$ from H = 20 ft. to H = 200 ft., by increments of 5 ft. As N_s refers to one impeller only in a multi-stage pump this range covers most requirements.

The relationship between specific speed, efficiency obtainable in practice and the general appearance of the impeller is shown in fig. 14, redrawn from a diagram by T. Y. Sherwell and R. Pennington (*Proc. Inst. Mech. Eng.*, 1933). Type A is the single-inlet impeller used in turbine pumps, where N_s does not exceed 1,100. Between N_s = 1,125 and N_s = 1,500 a medium-lift impeller of type B would be suitable; at still higher values of N_s, a low-lift impeller of type C. As N_s rises beyond 2,000 a mixed-flow impeller (type D) becomes necessary, with the vanes carried down into the eye; developing at N_s = about 7,000 into the 'Francis-vane' form (type E), which is partly a screw. Type F is the axial-flow or screw pump, combining a high revolution speed, very large output and low head.

The terms 'low lift,' 'medium lift,' and 'high lift' give a general impression of duty requirements, but are no longer a safe guide to impeller type. At the World Engineering Congress in Japan in 1929 it was suggested that they should be abandoned, and a simple division adopted into 'single-impeller and 'multiple-impeller' pumps; but specific speed more accurately indicates the kind of pump and form of impeller required for any given purpose.

SPECIFIC SPEED.

VALUES OF $H^{\frac{3}{4}}$ (OR $\sqrt[4]{H^3}$) FOR HEADS OF 20 FT. TO 200 FT., BY INCREMENTS OF 5 FT.

Head in ft. = H.	$H^{\frac{3}{4}}$	Head in ft. = H.	$H^{\frac{3}{4}}$
20	9·46	115	35·12
25	11·18	120	36·25
30	12·81	125	37·39
35	14·39	130	38·50
40	15·91	135	39·60
45	17·38	140	40·70
50	18·80	145	41·79
55	20·19	150	42·86
60	21·56	155	44·18
65	22·90	160	44·99
70	24·19	165	46·04
75	25·49	170	47·08
80	26·75	175	48·11
85	28·00	180	49·15
90	29·22	185	50·17
95	30·43	190	51·18
100	31·62	195	52·18
105	32·80	200	53·18
110	33·92		

DISC FRICTION OF IMPELLERS.

Calculations of power lost in disc friction are usually based on the work of Gibson and Ryan (*Proc. Inst. C.E.*, 1910) from which is derived the formula:

$$\text{Horsepower lost} = \frac{4\pi f\left(\frac{\pi N}{30}\right)^{n+1}}{550\,(n+3)}\left(\frac{D}{2}\right)^{n+3}$$

in which D = impeller diameter in feet;
N = revolutions per minute;
f and n = constants, depending on the peripheral velocity of disc and the degree of roughness.

Within the usual range of tip speeds the values of f vary from 0·0033 to 0·0045, and the values of n from 2·0 to 1·75. By taking the approximate values of f = 0·004 and n = 2, the formula may be simplified to

$$\text{Horsepower lost} = \frac{N^3 D^5}{15{\cdot}25 \times 10^8}$$

Velocity Diagrams for Impeller Vanes.

The various velocities of impeller and water at inlet and outlet are shown graphically in fig. 15, being represented to scale by the triangles ABC and DBF respectively. At inlet, BA is the radial velocity, set out to scale: BC, drawn parallel to the tangent to the eye at A, represents to the same scale the peripheral velocity of the vane tip at inlet; CA, to the same scale, represents the velocity and direction of the water particles along the surface of the vane. The angle BCA = the inlet angle ϕ of the vane, BC being parallel to the tangent at A.

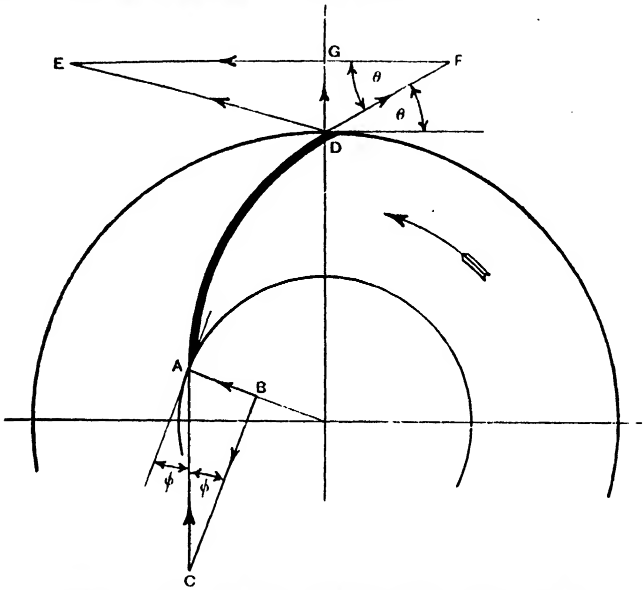

FIG. 15.—Velocity diagrams at inlet and outlet of impeller vanes.

At outlet the water leaves the vane at D, making an angle θ with the tangent at D, and having a velocity represented to scale by DF. FE, parallel to the tangent at D, represents the peripheral velocity of the impeller at the vane tip; and DE is the absolute velocity of the water particle, being compounded of the radial velocity DG and the 'velocity of whirl' GE.

$$\text{The radial velocity at inlet} = \frac{\text{Flow in cub. ft. per sec.}}{\text{Area of eye in sq. ft.}}$$

$$\text{and at outlet} = \frac{\text{Flow through impeller in cub. ft. per sec.}}{\text{Circumferential discharge area in sq. ft.}}$$

due allowance being made for the thickness of the vanes in calculating the area of discharge. In new designs the tip width may not be known when the velocity diagrams are drawn, and the radial velocity must then be assumed as a basis for the impeller dimensions. It may range from 5 ft. to 10 ft. per sec. in volute pumps, and to 15 ft. or even 18 ft. per sec. in turbine pumps.

Setting Out Impeller Vanes.

Impeller vanes are usually curved either to uniform circular arcs or involutes. A single involute curve can seldom be drawn to give both inlet and outlet angles correctly and separate involutes are, therefore, described, in length about a quarter to a third of the length of the vane, and so spaced that they can be joined by a smooth curve or a common tangent. This is best done by drawing the inlet involute on the paper and the outlet involute on a piece of tracing paper, which can be tried in different positions around the impeller periphery.

Fig. 16 shows the geometrical construction of a circular arc to cut the circles of the impeller eye and outer periphery at the given angles of $\phi°$ and $\theta°$ respectively, starting from the inlet edge of the vane. Fig. 17 shows a similar construction, starting from a set point on the outer circle. Fig. 18 shows the method of setting out involute tips. In all three impellers the rotation is left-handed, and the diagrams have been drawn for an inlet angle of 20° and an outlet angle of 30°.

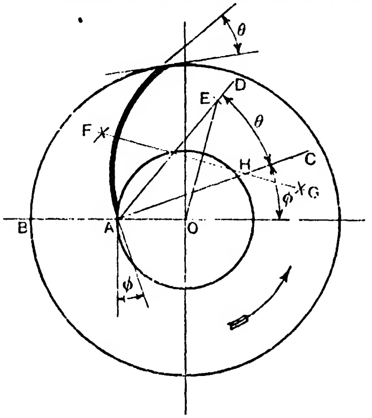

FIG. 16.—Construction for circular arc vane (I).

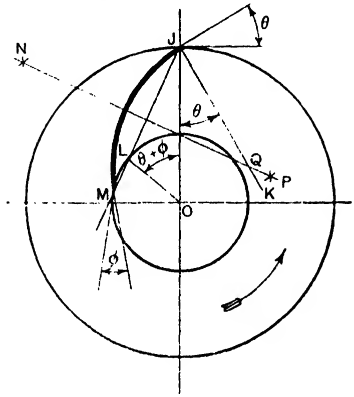

FIG. 17.—Construction for circular arc vane (II).

Fig. 16.—OA = radius at inlet to vane, OB = radius at outlet from vane. From A draw AC, making the angle ϕ with OA, and AD making the angle θ with AC, so that $\angle$ DAO = $(\phi + \theta)°$. On AD set off AE = OB, and join E – O. Draw FG bisecting EO at right angles and cutting

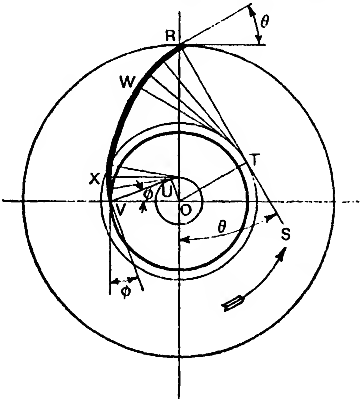

FIG. 18.—Construction for involute vane.

AC at the point H. With centre H an arc described with radius HA will cut the inner and outer circles at the required angles ϕ and θ respectively.

Fig. 17.—From any point J on the outer periphery draw JK, making the angle θ with the radius OJ.

From the centre O draw the radius OL of the inner ('eye') circle, making the angle $(\varphi + \theta)$ with OJ. Join J — L and produce to cut the inner circle again at M. Bisect JM at right angles by the line NP, produced to cut JK at Q. The point Q is then the centre from which to describe an arc with radius QJ, making the required angles with the inner and outer circles.

Fig. 18.—An involute is the curve traced by the end of a cord as it unwinds from a cylinder, represented by the base circle.

Let V be the tip of the vane at inlet, and R the tip at outlet. Draw VU making the angle ϕ with the radius OV; and OU making a right angle with VU. Then OU is the radius of the base circle of the inlet involute.

On a piece of tracing paper placed over the diagram draw RS, making the angle θ with the radius OR, and OT making a right angle with RS at T. Then OT is the radius of the base circle of the outlet involute.

Having described the involutes, rotate the tracing paper about the centre O until a position is found in which the two involutes can be joined by a smooth curve or a common tangent to complete the vane. It is advisable to draw two adjacent vanes, as the form of the connecting curve may require adjustment to correct the sectional area of the passage between them.

The involutes are described by drawing a succession of tangents to the base circles, gradually increasing in length in proportion to the amount of the imaginary cord unwound, in the case of the inlet; decreasing as the cord winds on to the cylinder in the case of the outlet involute. For all but the largest impellers the amount of the variation can be judged by eye with sufficient accuracy; or if the scale permits, a base circle of thick cardboard or wood may be used, and a thin wire unwound from it, a pencil at the end of the wire tracing the curve. It is necessary to use wire for accurate work, as string stretches too easily.

There is little to choose in efficiency between the circular arc vane and the involute. The involute gives a longer vane, and consequently longer water passages and a slightly higher surface friction loss; but as the tips of successive vanes are parallel the flow is usually more even, with less tendency to internal eddy-making. The circular arc is more easily drawn and perhaps makes for more accurate pattern-making.

Number of Impeller Vanes.

There is no definite rule for the number of vanes, which may vary from 2, 3 or 4 in dredging pumps to 12 in turbine pumps, and 14 or even more in special pumps for low and medium lifts. If too few are fitted eddy-making behind the vanes impairs the efficiency, and cavitation may take place at high speeds, leading to impeller corrosion; a large number of vanes increases the loss by fluid friction in the passages, but gives better guidance to the water. Eight or ten is usually the largest number of complete vanes that the circumference at the eye will permit without undue constriction of the passages; if more than eight are provided, alternate vanes are reduced in length by about one-third and do not reach the eye. Six is the least number for pumping water with reasonable efficiency. The designer's judgment must decide in any particular case; guided, if need be, by estimates of the losses with different arrangements.

Example of Centrifugal Pump Design.

Required—To design a centrifugal pump to deliver 2,100 gals. per min. against a total head, including friction, of 45 ft.

Allowing 5 per cent. for leakage past clearances, the impeller must handle $2{,}100 \times 1 \cdot 05 = 2{,}200$ gals. per min. Pump branches 12 ins. bore will pass 2,200 gals. per min. at 7·15 ft. per sec. The duty is one for a low-lift pump running at 600 to 750 r.p.m.; 720 r.p.m. may be adopted, being a standard speed for a.c. motors on a 50-cycles supply. The specific speed (see p. 819) can now be calculated.

$$\text{Specific speed } N_s = \frac{720\sqrt{2{,}200}}{45^{\frac{5}{4}}}$$

From the table (p. 821) $45^{\frac{5}{4}} = 17 \cdot 38$; and $\sqrt{2{,}200} = 46 \cdot 9$

$$\text{Therefore } N_s = \frac{720 \times 46 \cdot 9}{17 \cdot 38} = 1{,}943$$

From fig. 14, at $N_s = 1{,}943$ an efficiency of about 85 per cent. may be obtained with a double-inlet impeller of type *C*, but 80 per cent. will be assumed, to leave a margin for contingencies.

Water h.p. $= \dfrac{2{,}200 \text{ (g.p.m.)} \times 10 \text{ (lbs. weight per gal.)} \times 45 \text{ (ft.)}}{33{,}000} = 30.$ At 80 per cent. efficiency, the power required at the pump coupling $= \dfrac{30}{0 \cdot 8} = 37 \cdot 5$ b.h.p.; calling for a motor rated at, say, 45 h.p.

As $v^2 = 2gH$, and the head H = 45 ft., the required peripheral speed of the impeller $= 8\sqrt{45}$ = 53·5 ft. per sec. At 720 r.p.m., the revolutions per sec. = 12, and the circumference in ins. $= \frac{53 \cdot 5 \times 12}{12} = 53 \cdot 5$ ins., and the diameter $= \frac{53 \cdot 5}{\pi}$, say 17 ins. The quantity in cu. ft. per sec. $= \frac{2{,}200}{6 \cdot 23 \times 60} = 5 \cdot 9$ cu. ft. The quantity entering through each eye of the impeller = 2·95 cu. ft. per sec. Velocity through the suction branch = 7·15 ft. per sec., which may be increased at the vane inlets to a radial velocity of 9 ft. per sec. Clear area required through eye, therefore, $= \frac{2 \cdot 95}{9} = 0 \cdot 328$ sq. ft. Diameter of boss required will be about 3½ ins. = 0·067 sq. ft.; and total area of eye, therefore, = 0·328 + 0·067 = 0·395 sq. ft. = 57 sq. ins., and diameter of eye = 8½ ins. Neglecting vane thickness,

$$\text{Axial width of impeller passages at eye} = \frac{\text{Cu. ft. per sec.}}{\text{Circumf. in ft.} \times \text{rad. vel.}} = \frac{2 \cdot 95}{\frac{\pi \times 8 \cdot 5}{12} \times 9}$$
$$= 1 \cdot 77 \text{ ins., say } 1\tfrac{3}{4} \text{ ins.}$$

At outlet a radial velocity of 8 ft. per sec. may be assumed and, again neglecting vane thickness, vane width at tip $= \frac{\text{Cu. ft. per sec.}}{\text{Circ. ft.} \times \text{Rad. vel.}} = \frac{5 \cdot 9}{\frac{53 \cdot 5}{12} \times 8} \times 12 = 1 \cdot 98$ ins., say 2 ins.

Taking the area at throat = 0·4 of the area of delivery branch, throat area = 113 × 0·4 = 45 sq. ins., giving a velocity = 18 ft. per sec. From the dimensions now available the velocity diagrams (see fig. 15) can be drawn to find the angles at inlet and outlet. As the eye diameter of 8½ ins. is half the extreme diameter, peripheral velocity at inlet = 26·75 ft. per sec., and the inlet angle ϕ may be found by calculation, as $\tan \phi = \frac{\text{radial vel.}}{\text{periph. vel.}} = \frac{9}{26 \cdot 75} = 0 \cdot 336$. As tan 18° = 0·325, the inlet angle may be made 18°. Similarly, for the outlet angle θ, $\tan \theta = \frac{\text{radial velocity}}{\text{velocity of whirl}} = \frac{8}{(53 \cdot 5 - 18 \cdot 0)} = \frac{8}{35 \cdot 5} = 0 \cdot 2254$; and as tan 13° = 0·2309, the outlet angle may be made 13°. (The assumption that the peripheral velocity $v = \sqrt{2gH}$ is not applied to every type; v may vary from $1 \cdot 2\sqrt{2gH}$ in a slow-running, large, low-lift pump to $0 \cdot 8\sqrt{2gH}$ in a high-lift turbine pump.)

Erection of Centrifugal Pumps

Centrifugal pumps of large size are mounted directly upon a foundation block of concrete, but pumps below about 18 ins. (the dimension being the bore of the delivery branch) usually stand on a cast-iron bedplate, which also supports the driving motor. Pump and motor are positioned on the bedplate by fitted dowel pins, and secured by holding-down bolts. The dowels should not be fitted until the bedplate is finally bolted down to the foundation, lest any small irregularity in the foundation should distort the casting and necessitate re-alignment of the pump with the motor. A flexible coupling of the pin type is almost invariably fitted between pump and motor (generally, though not always, with the pins bolted to the pump half and free in the motor half, as in fig. 7), but the flexibility of this type of coupling is limited to a small axial movement, permitting the motor armature and the pump impeller to centre themselves when running. A pin coupling of this type will not correct faulty alignment; therefore care must be taken in erecting to ensure that the two halves of the coupling are truly centred, and their faces exactly parallel. A single-inlet impeller moves axially towards the suction end when running, and the spindle should be moved to the limit in that direction before lining-up; a double-inlet impeller should be set centrally, as it is designed to be in axial balance when running. The motor armature should also be set centrally, or the motor run 'light' and the axial running position noted. The clearance between the coupling faces may range from ⅛ in. on a 1-in. diameter shaft to ⅜ in. on a 6-in. shaft.

Packing for Centrifugal Pump Spindles.

Soft packing only must be used in centrifugal pump stuffing boxes, especially if the spindle is fitted with bronze sleeves. Hard packings, or packings containing tallow (which sets hard in cold water) or flax, though suitable for reciprocating pumps, cause rapid wear of rotating spindles. Soft cotton impregnated with graphite is preferable, four or five rings being the usual quantity. The lantern ring, for introducing the sealing water, is placed between two packing rings and opposite to the sealing water connection; its axial width should allow of ½ in. movement along the shaft without risk of obstructing the connection. The glands should be bored $\frac{1}{16}$ in. larger than the spindle, and only lightly screwed up. A small seepage of water should be always visible, as an indication that the packing is not being run dry, and, on the suction side, that air is not leaking into the pump. Unless the water pumped is perfectly clear, a separate clean supply should be arranged for sealing.

Wherever a gland can be avoided by completely enclosing the end of the spindle, this should be done. In turbine pumps with hydraulic balancers the delivery end gland is outside the balance chamber, and is not subjected to the full delivery head.

Metallic packings for centrifugal pump use have been developed with some success, and grease lubrication is also used in some turbine pumps. Where grease is employed it is necessary to provide means to apply sufficient pressure to the grease injected to overcome any hydraulic pressure acting on the bearing from within the pump.

Starting a Centrifugal Pump.

If the pump is being run for the first time, see that the bearings are filled with oil to the level of the underside of the spindle.

A centrifugal pump must never be started or run unless it is full of water.

To Prime the Pump.—If a suction foot-valve is fitted, open all aircocks, close the delivery valve, and fill the pump with water until there is an overflow at the aircocks. Then shut the aircocks and start the pump. When nearly the full normal delivery pressure is reached (to be seen by the delivery gauge, if fitted, or judged by the ammeter reading, in a motor-driven set) open the delivery valve *slowly*.

Small pumps are usually fitted with a funnel for priming; or, if there is a standing head of water in the rising main, they can be primed by opening the delivery sluice valve slightly and allowing water to run back. The delivery sluice and non-return valves of large or high-pressure sets are fitted with by-pass valves for this purpose.

A pump should not be run for more than a few minutes with the delivery entirely closed. If no water is passing the energy of the driving motor is dissipated in churning and heating the small amount of water in the pump casing, causing a rapid temperature rise, with risk of seizure and serious damage. Close the delivery valve immediately before stopping the pump.

Foot-valves cannot be relied upon to remain perfectly tight, and a very slight air leak at a joint may, therefore, allow the suction pipe to empty when the pump is standing. If a suction sluice valve is not fitted, or has been left open, the water in the casing may also drain away, and the pump must be primed again when restarting. If a suction vacuum gauge is fitted the gauge cock should be closed before priming and kept shut until the pump is delivering freely, to protect the mechanism of the gauge against damage by the static pressure in the pump.

Pumps located below supply level, *i.e.*, with a positive head on the inlet instead of a vertical suction lift, are primed by opening the suction sluice valve, which should be fully opened before the pump is started.

A centrifugal pump will sometimes pick up water for a few moments, as shown by the delivery gauge, and then 'lose the suction.' This is evidence of air in the suction and may be due to the formation of a vortex in the water in which the suction is submerged. If the pipe is open-ended, the addition of a bell-mouth or a strainer may effect a cure; or, in an emergency, holes may be drilled in the pipe, as far as possible below water-level, to reduce the inlet velocity at the extreme end.

Rotary Pumps.

Although the centrifugal pump is rotary in the mechanical sense, the term 'rotary pump' denotes in general usage a positive-displacement pump containing a rotor, and forcing the pumped liquid by direct pressure constantly in the one direction of flow. Of the eight diagrams, figs. 19 (a) to (h), six show true rotary pumps according to this definition; the first, fig. 19 (a), and the last, fig. 19 (h), conform to the definition in part only. The wing pump, fig. 19 (a), although the action of the handle is semi-rotary and there is no reversal of flow, is in principle a reciprocating pump of the valve-in-bucket type arranged to work in a circular casing.

The swashplate pump, fig. 19 (h), is also a reciprocating pump, though often classed as a rotary pump. The cylinders are formed in a rotatable barrel keyed to the spindle. The swashplate fits loosely on the spindle and is prevented from turning in the casing. As the cylinder block is revolved by the spindle, the inclination of the swashplate causes the pistons to reciprocate; suction and delivery taking place through ports in the end plate which put each cylinder in turn in alternate communication with the suction and delivery branches. By mounting the swashplate on trunnions so that the angle of inclination can be altered at will, the stroke of the pistons can be varied, thus giving a variable delivery at a constant speed of rotation.

A rotary pump contains two essential elements, namely, the rotating part through which the power is applied to cause displacement of the fluid; and an abutment against which the liquid is pressed to produce outward delivery. The pumps are classified: (*a*) according as the abutment is fixed or movable; and (*b*) in sub-divisions under these main headings according to the method of causing displacement. In the language of patents the rotating element is a 'piston,' although its motion may be solely rotary. Some categories in the complete classification have little significance, most rotary pumps in commercial production being modifications of the six general types illustrated in figs. 19 (b) to 19 (g). There are also various devices such as a roller compressing a flexible tube coiled within a cylindrical casing, to expel the contents; rotating tubes, such as the Frenier slime pump, formed in a spiral like a snail's shell and discharging through the trunnions; etc.

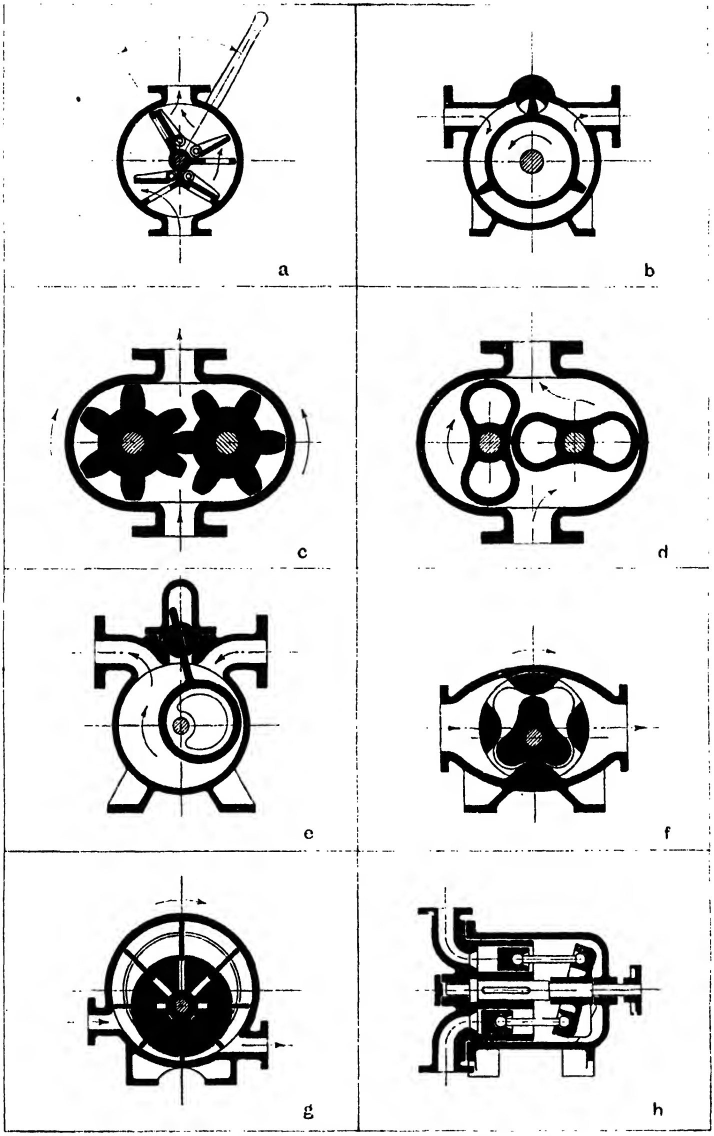

FIG. 19.—Typical rotary pumps.

CLASSIFICATION OF ROTARY PUMPS.

(Based upon Patent Office Classification.)

Fixed Abutment.

1. Oscillating vanes.
2. Piston with axially-sliding vanes.
3. Piston with hinged vanes.
4. Piston with outwardly-sliding vanes.
5. Piston with roller vanes.
6. Piston with rotary vanes.
7. Piston with sliding-rocking vanes.

Movable Abutment.

8. Abutments sliding axially.
9. Abutments sliding outwardly.
10. Geared vane type (intermittent abutments).
11. Gear wheel type (in principle allied to the Roots Blower).
12. Hinged or rocking abutments.
13. Internally-meshing rotors with n and $n + 1$ teeth or lobes.
14. Roots Blower type.
15. Rotary abutments (other than types 3, 6, and 7).
16. Sliding-rocking abutments.

Fig. 19 (b) shows a pump of the rotary abutment type (15). By external gearing the abutment is revolved as many times as there are vanes, for each revolution of the rotor. The displacement is the volume swept by the vanes in a given time between the points of inlet and delivery. The gear-wheel pump (11), shown in fig. 19 (c), has no external gearing; the drive being through the spindle of one gear wheel, which drives the other. The centres may be adjustable for distance apart, as some backlash between the teeth is necessary for smooth working, but too much allows leakage. (For a discussion of gear pump design, see *The Engineer*, Feb. 15, 1929.) Fig. 19 (d) shows one of the many forms of the Roots Blower type (14), this variety being invented by J. T. Wilkin in 1892. The lobes are formed with two epicycloids and two hypocycloids, the surfaces of which work together with uniform angular velocity, and are connected by external gearing.

Fig. 19 (e) is a pump with sliding-rocking abutment (16); if the vane were arranged to slide radially in a slot in the casing, with a hinge or pivot attachment to the eccentric 'piston,' the pump would become a fixed abutment type of class (7). Fig. 19 (f) shows a pump of n and $n + 1$ lobes gearing internally, the n rotor driving the $n + 1$ rotor. Fig. 19 (g), with radially-sliding abutments moving in slots under the guidance of an eccentric ring or groove in the casing, resembles in principle some radial reciprocating pumps in which the plungers are similarly constrained to move into and out of the cylinders, supply and delivery being through ports and passages in the shaft (*e.g.*, the Hele-Shaw variable gear pump and motor, *Proc. Inst. Mech. Eng.*, 1921). Air pumps of the type of fig. 19 (g) are sometimes used as exhausters for priming large centrifugal pumps. To prevent undue tip leakage in rotary vane pumps, a loose inner sleeve may be provided in the casing, free to revolve under the frictional drag of the vanes, and the viscosity of the liquid, and fully ported to leave ample suction and delivery area, however the sleeve may rest.

Air Lift Pumping.

The air lift is a method of raising liquid by injecting air into the lower end of an open pipe immersed in the liquid to be pumped. It is most commonly employed in raising water from boreholes, and this application is here described as typical of the system, but the air-lift is also extensively used to raise oil from deep bores, and has a further sphere of usefulness in the pumping of corrosive chemicals. The action is illustrated in the diagram, fig. 20, and is explained as follows:

If an open-ended pipe is placed in a borehole containing a standing head of water, the water-level stands in the pipe at the same height as in the bore, the water columns inside and outside the pipe being in equilibrium. If air is injected into the foot of the pipe or rising main so as thoroughly to aerate the water contained in it the state of equilibrium is destroyed, as the density of the mixture of air and water is less than that of water alone. The column of aerated water inside the pipe being no longer heavy enough to balance the corresponding height of water outside, and the foot of the pipe being open to the well, more water flows into the foot of the pipe, under the pressure of the standing head, and pushes the aerated column upwards in the endeavour to restore equilibrium. If the pipe be continued upwards indefinitely, a state of balance will again be reached when the weights of the two columns regain equality and the air will merely rise through the aerated column and escape to the atmosphere, no further flow taking place. The aim of the designer is, therefore, so to proportion the total lift to the submergence as to ensure that the aerated column shall reach the height at which delivery is desired, thus maintaining a continuous flow, and to do so with the minimum expenditure of power.

ADVANTAGES AND DISADVANTAGES OF THE AIR LIFT SYSTEM.

The advantages of the system are:

1. There are no moving parts in the borehole.

2. Provided that the yield is available, a large quantity of water can be pumped from a smaller bore than would suffice for any other apparatus for deep-well pumping.

3. It is not essential that the bore should be absolutely straight, nor that it should be truly vertical.

4. The presence of sand or other foreign matter in the water, so detrimental to mechanical deep-well pumps, has practically no effect upon the air lift.

5. The surface machinery, comprising the compressor and its prime mover, air receiver, tanks, etc., can be placed wherever is most convenient, possibly at some distance from the well head.

6. Duplicate compressing plant can be installed to provide a complete stand-by equipment without increasing the amount of apparatus to be accommodated in the well.

7. A number of boreholes can be pumped simultaneously from a central compressing station.

8. Water may be pumped at varying rates, subject to the maximum yield of the well, with only slight variation in efficiency.

9. Flow does not cease suddenly as the result of only a small fall in the pumping level.

The main disadvantage is the low overall efficiency of the system. Although under favourable conditions an efficiency of 50 per cent. can be obtained, 33 per cent. is considered a good result, if the water must be raised through any great height. Under unfavourable conditions the efficiency may fall as low as 15 per cent. or even 10 per cent. If hot liquid is being lifted, the apparent efficiency may reach 100 per cent., as the air is further expanded by heat abstracted from the liquid.

The bore must be carried to a greater depth for an air lift than for other deep-well pumps; and if a considerable drop in water-level takes place, may need further deepening to ensure proper submergence for the air ejector, or the efficiency will suffer a serious reduction. The capital cost of further deepening, however, is often more than counterbalanced by the saving in running costs resulting from corrected submergence.

Data Required for Designing:

The following particulars must be considered in designing an air-lift plant.

1. The yield of the well.
2. The standing water-level in the well.
3. The working or pumping level (below the standing level by an amount depending on the rate at which water flows in to replace the water pumped out).
4. The total height through which the water must be raised.
5. The correct submergence for the air-lift tubes.
6. The sizes of the rising main and air main.
7. The working pressure to which the air must be compressed.
8. The correct amount required of free air per min.
9. The h.p. required.

The diagram, fig. 20, shows the various measurements required and the terms used.

A = Total vertical height through which the mixture of air and water passes.
B = Correct submergence when running at full capacity.
C = Submergence at starting.
D = Total vertical lift = elevation + static head + drop.
E = Static head, or depth of standing water-level below surface.
F = 'Drop,' or distance between standing water-level and pumping water-level.
G = Elevation above the surface.

There are so many variable quantities associated with the air lift that no definite rule can be laid down or formula devised to cover all contingencies, and it is necessary that this should be clearly understood. To mention only a few of the factors which affect an air lift plant in greater or lesser degree, the height above sea-level, smoothness of air and water pipes, temperature of the water, amount of submergence, the ratio of air to water, and the sizes of air main and rising main have each an important bearing upon the final results. The following information, representing the practice of one of the leading British firms specialising in this type of plant, provides a guide to the design and general operation of air lifts. An installation designed on the lines given will give satisfactory performance, although for the best possible efficiency, as stated above, consideration should also be given to a number of other points peculiar to the particular case.

Volume of Air Required.

The volume of air required for any given quantity of water depends mainly, though not wholly, upon the height to which the water is to be raised. Several formulæ have been evolved to express the volume, and the following (by Rix and Abrams) will be found to give good results under normal working conditions, assuming correct submergence.

$$V_a = \frac{h}{C \log \frac{H + 34}{34}}$$

where V_a = cub. ft. of free air per gal. of water;
h = total lift in ft.;
H = submergence in ft. when working;
C = a constant.

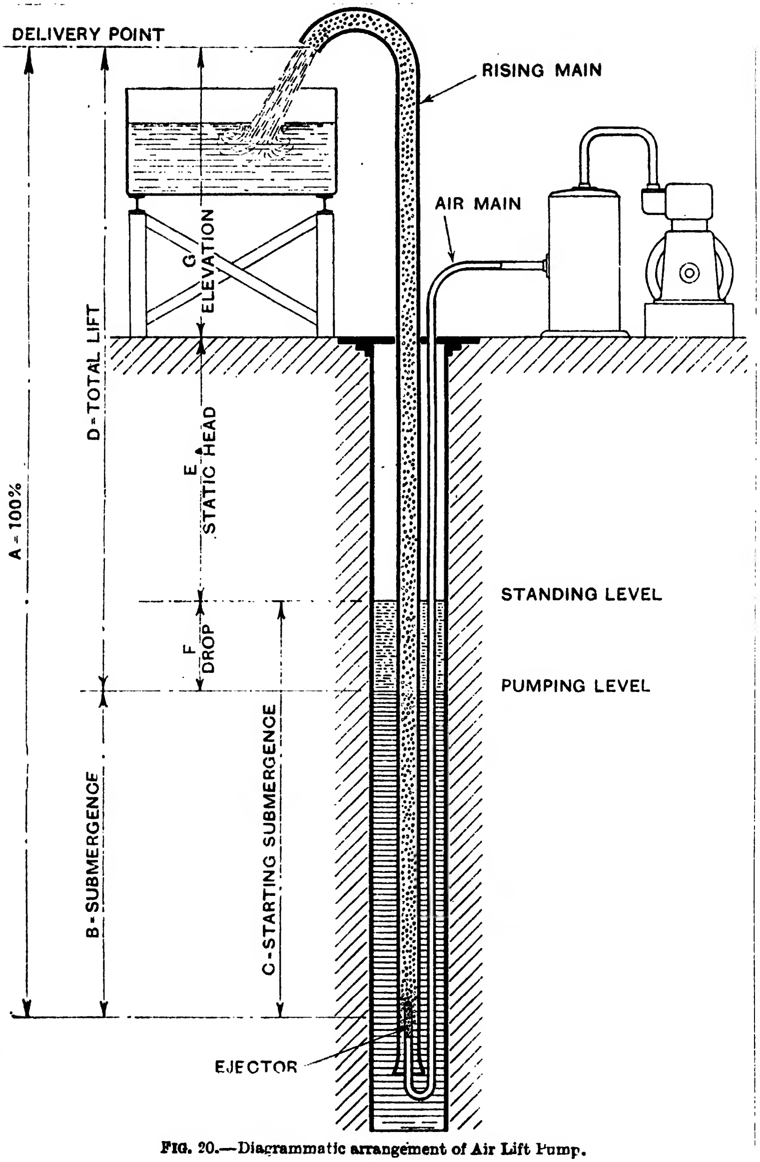

FIG. 20.—Diagrammatic arrangement of Air Lift Pump.

The values of C for various lifts can be taken as follows:

From 30 ft. to 60 ft.	. . .	C = 204
" 60 " " 200 "	. . .	C = 194
" 200 " " 500 "	. . .	C = 180
" 500 " " 650 "	. . .	C = 153
" 650 " " 750 "	. . .	C = 130

(It should be noted that the Rix and Abrams formula is sometimes quoted from American sources in the form

$$V_a = 0 \cdot 8 \frac{h}{C \log \frac{H + 34}{34}};$$

the coefficient 0·8 giving a reduction in the volume of air corresponding to the smaller capacity of the United States gallon. Slightly different values may also be found for the constant C, but those given above represent good practice and can be employed with certainty as being well proved.)

Submergences.

The submergence, *i.e.*, the depth to the air ejector below the pumping water-level, depends chiefly upon the total lift, although it may be varied considerably to suit local conditions. The following proportions will ensure a good efficiency.

Lift in ft. . .	50	75	100	150	200	250	300
Ratio $\frac{\text{Submergence}}{\text{Lift}}$	$\frac{2 \cdot 5}{1}$	$\frac{2 \cdot 0}{1}$	$\frac{1 \cdot 6}{1}$	$\frac{1 \cdot 5}{1}$	$\frac{1 \cdot 3}{1}$	$\frac{1 \cdot 2}{1}$	$\frac{1 \cdot 0}{1}$

Size of Rising Main.

The two main points to consider in selecting the best diameter of rising main are (1) the volume of water, and (2) the height to which it is to be raised. If the diameter of the main is too large the combined velocity of the air and water will be insufficient and slippage will take place, *i.e.*, the air will rise through the water, causing loss of efficiency. On the other hand, too small a diameter involves a high velocity of flow, causing undue friction and consequent *loss*. In actual practice the combined velocity of flow is never constant, but gradually increases as the mixture rises, due to the increase in air volume, which in turn results from the decrease in air pressure as the emulsion travels upwards. For any given size of pipe, therefore, the velocity is at its minimum at the point of injection, and at the maximum at the point of discharge. At the point of injection the velocity may range between 5 ft. and 12 ft. per sec.; and at the point of discharge may be allowed to vary between the limits of 12 ft. and 45 ft. per sec. without detriment. The higher the lift the greater the discharge velocity, and the greater the difference between the initial and final velocities. When the total lift is large it is sometimes advisable to use two or even three sizes of rising main, increasing the diameter at the higher levels so as to restrict the velocity of flow in the upper part of the main; the theoretical ideal of a uniform taper from bottom to top being obviously impracticable on economic grounds. For practical purposes the nearest standard size of pipe should be selected which will give rates of flow between the above limits; the result being checked by the formula:

$$\text{Velocity in ft. per sec.} = \frac{Q}{A};$$

where Q = the combined volume of air and water at any point in cub. ft. per sec.;
A = cross-sectional area of the pipe in sq. ft.

At the point of injection the volume of the compressed air must, of course, be taken, the pressure in lbs. per sq. in. being governed by the depth of the ejector below the pumping water-level.

Size of Air Main.

Various formulæ have been devised to give the most suitable size of air tubing; the one here given is known as Johnson's formula, and can be recommended from experience.

$$P_1^2 - P_2^2 = \frac{\cdot 0006\, q^2 L}{d^5}$$

where L = length of air pipe in ft.;
d = bore of air pipe in ins.;
P_1 = initial air pressure in lbs. per sq. in. (absolute;
P_2 = final air pressure in lbs. per sq. in. (absolute);
q = cub. ft. of free air per min.

Transposing to obtain d, the formula becomes:

$$d = \sqrt[5]{\frac{\cdot 0006\, q^2 L}{P_1^2 - P_2^2}}$$

The length of air main taken must include that above the surface and, therefore, does not necessarily correspond to the length of the rising main, even approximately. It is very important that the air main should be of adequate diameter, to avoid undue friction; the friction is converted into heat, which in turn increases the volume of the air, leading in a vicious circle to a still higher velocity and more friction. A proportion of 1 to 6·24 between the cross-sectional areas of the air main and rising main may be kept in mind as a safe guide.

The air main may be fitted inside the rising main, in which case the same sectional area of air main can be adopted (the slight difference in cooling effect being neglected), and the diameter of the rising main proportionately increased. Taking the given ratio of 1 to 6·24, the diameter of the rising main will then be

$$3 \cdot 04 \sqrt{(\text{area of air main})};$$

the nearest standard size being taken in practice, as in the former case.

H.P. Consumed.

For normal lifts up to about 250 ft. it will be found that single-stage compressors are usually equal to all requirements; but if the working pressure is likely to exceed 110 lbs. per sq. in., it is advisable to use a two-stage or three-stage machine. The following table gives the approximate b.h.p. required by a well-designed compressor to deliver 1 cu. ft. of free air per min. (referred to as 'F.A.D.' = Free Air Delivered) at the gauge pressures indicated.

Gauge Pressure. Lbs. per sq. in.	Brake H.P.	Gauge Pressure. Lbs. per sq. in.	Brake H.P.
10	0·047	55	0·167
15	0·066	60	0·177
20	0·082	65	0·185
25	0·098	70	0·193
30	0·110	80	0·210
35	0·124	90	0·225
40	0·136	100	0·238
45	0·146	110	0·250
50	0·157	120	0·264

Arrangements of Pipes.

It is usual to fit the air main outside the rising main; the internal arrangement of air pipe being adopted only when space is very limited. Many types of ejector nozzle have been evolved and used from time to time, but in actual practice there is not a great deal to choose between them. It is desirable that the stream of air should be broken up as much as possible in the form of small bubbles, to prevent slippage. Nozzles of the perforated type are usually drilled with holes ¼ in. diameter. Probably the best method of introducing the air pipe is also the simplest, *i.e.*, to lead it below the foot of the rising main and up the centre, as shown in the diagram (fig. 20). The ejector should be located not less than 3 ft. from the foot of the main.

The delivery may be taken up the side of the collecting tank with a plain bend, as in the diagram; or may be led straight through the bottom of the tank, continued to a height of 2 or 3 ft. above the tank water-level, and surmounted by an umbrella-shaped deflector. The support of the rising main and air main at the well-head need only be of the simplest description; a plain, cast-iron suspension flange being quite sufficient.

Arrangement of Compressing Plant.

Wherever possible the compressor should be installed in a cool, clean room, and a generous space allowed around it for oiling and cleaning. The air intake should be placed outside the engine room to avoid a heated or dusty atmosphere; bearing in mind that a rise of 5° F. in the temperature of the intake air will cause a decrease of 1 per cent. in the volumetric efficiency of the compressor.

An air receiver of ample size should always be interposed between the compressor and the well, and provided with safety valve, pressure gauge, and a blow-off cock. On no account should a stop-valve be fitted in the air line between compressor and receiver. The function of the receiver is to damp out the pulsations and (perhaps more important) to act as a trap for oil and moisture carried over from the compressor. The blow-off cock should be used daily to draw off the trapped oil.

A greater pressure is required at starting than for maintaining flow, as at first the rising main contains solid water which the air pressure tends to force, unaerated, out of the delivery. The initial discharge is usually unbroken, and is followed by a highly aerated section. This alternation of water and air may be repeated a few times before steady flow begins. The correct discharge is an opaque emulsion, which clears almost instantly when drawn off in a glass.

TYPICAL AIR LIFT PLANTS. (*Le Grand, Sutcliff & Gell, Ltd.*)

	Gasworks.	Laundry.	Paper Mill.	Cement Works.	Brickworks.
Borehole diameter	10½ ins.	8¾ ins.	7½ ins.	12 ins.	10½ ins.
Borehole depth	520 ft.	500 ft.	898 ft.	380 ft.	250 ft.
Rising main diameter	4 and 3½ ins.	4 and 3½ ins.	2½ ins.	3 ins.	4 ins.
Size of air main	2 ins.	2 ins.	1 in.	1½ ins.	1 in.
Standing water level (below surface)	200 ft.	207 ft.	Overflow	149 ft.	44½ ft.
Pumping water level	282 ft.	299 ft.	96 ft.	169 ft.	55 ft.
Total lift	302 ft.	334 ft.	116 ft.	174 ft.	62 ft.
Size of compressor	9 ins. × 6 ins.	9 ins. × 6 ins.	6 ins. × 4½ ins.	6 ins. × 6 ins.	6 ins. × 7 ins.
Compressor capacity in cu. ft. min.	300	315	50	75	68
Power of driving motor (electric)	80 h.p.	80 h.p.	14 h.p.	22 h.p.	14 h.p.
Depth to ejector	511 ft.	487 ft.	300 ft.	343 ft.	164½ ft.
Submergence	229 ft.	188 ft.	204 ft.	174 ft.	102½ ft.
Yield in gals. per hr.	8,000	6,000	4,000	5,000	11,000

THE PULSOMETER STEAM PUMP.

The Pulsometer stands in a class by itself. It may be described as a development of the Savery pump for the suction is effected by the condensation of steam in a chamber and the water is lifted by the pressure of steam acting directly upon its surface. By the employment of two chambers the delivery is made continuous and completely automatic, and by careful attention to design a good efficiency is attained. The action of the Pulsometer is fully described in Descriptive Section XIX, Part III.

THE HYDRAULIC RAM.

The water-hammer effect due to the momentum of a long column of moving water in a pipe, noticed in connection with reciprocating pumps, is turned to advantage in the hydraulic ram, invented by Joseph Montgolfier and sometimes called by his name, which utilises the momentum of a relatively large flow, under a small head, to raise a smaller quantity against a great head. Although not usually classed as a pump it may be regarded as one if the moving column in the supply or 'drive' pipe be considered as the plunger which, actuated by its own acquired energy, forces the water in the chest up the delivery pipe. The action can be followed from the sectional diagram, fig. 21, on p. 832. Water from the source of supply, usually a small stream, is led with a moderate fall through the drive pipe into the branch marked 'Injection,' and at low velocities can flow freely to waste through the 'outer valve' or 'pulse valve,' which is of the mushroom type, arranged to open downward by its own weight. When the flow reaches a sufficient velocity the pulse valve is lifted sharply against its seat, the momentum of the moving water column, thus suddenly arrested, causing an instant pressure rise in the chest, forcing open the 'inner valve,' and driving water up the delivery pipe until equilibrium is restored. The delivery valve closes with the cessation of upward flow and the pulse valve falls again to the open position, allowing the cycle to repeat. An air vessel on the delivery prevents damage by shock. Both valves are provided with means to adjust the lift, and the flow into the drive pipe is regulated at the source by a sluice. The ram can be stopped by raising the pulse valve against its seat by means of the projecting spindle, when the head of water in the drive pipe and chest will hold it there; and it can be restarted by pressing down the spindle to force the valve off its seat and allow flow to be resumed. A snifting valve is fitted to admit cushioning air in replacement of that absorbed by the water.

A polluted supply can be used to raise clean water from another source by means of the modification shown in fig. 22, in which the dirty water actuates a plunger in a cylinder, the other side of the plunger being in contact with the clean water to be pumped. When the drive water carries leaves or other matter in suspension a screening chamber must be provided at the inlet to the drive pipe.

Rams can be made to work with a fall as low as 18 ins., but as a rule 5 ft. is the practical minimum. With standard types the delivery head may be as high as 200 ft., beyond which figure a specially strengthened apparatus will probably be necessary. The working fall and rate

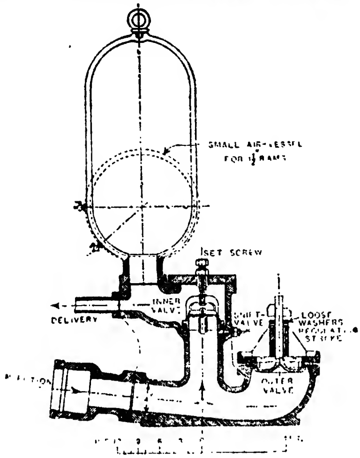

FIG. 21.

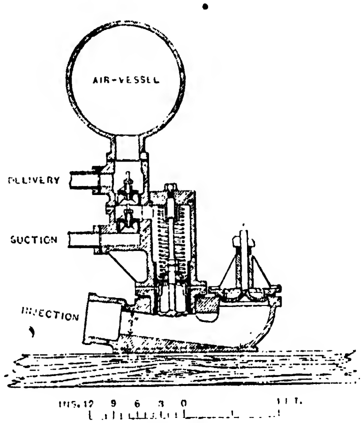

FIG. 22.

of flow determine the limiting conditions of discharge in a given case, the efficiency of the ram as a water-raising machine depending on the ratio of vertical height of delivery to vertical fall of driving supply. If W = available drive water in galls. per minute, H = vertical fall in ft., h = vertical height of delivery in ft., and Q = galls. raised per minute

$$Q = \frac{W \times H}{h} \text{ (efficiency per cent.)}$$

Typical efficiencies for various ratios of h/H are as follows :—

h/H.	2	3	4	5	6	7	8	9	10	12	15	18	20	25
Effy.	·85	·85	·80	·75	·75	·70	·65	·65	·60	·60	·55	·45	·40	·40

If the length of the delivery pipe is unusually long the efficiency will be less than is given in the table. The efficiency of a dirty-water ram of the type shown in fig. 22 is also less, owing to internal losses, and ranges from 35 to 56 per cent. By increasing the stroke of the pulse valve a greater quantity is delivered, but at a reduced efficiency, and *vice versa*. If the largest possible delivery is required with a limited supply of water a large ram should be used, with a short stroke of valve, but if water is plentiful it is cheaper to use a small ram with a long stroke. Ball valves are not advisable. The delivery valve should have 1 sq. in. of area per gallon to be delivered per minute. The area of the annulus between the edge of the pulse valve and the wall of the chest should be rather less than the area of passage through the valve, to impart a high speed of flow past the edge and a sharp closing action. This feature is shown in the half-section of the valve and chest, in fig. 23. The length of the drive pipe may be from 9 ins. to 12 ins. for every foot of height through which the water is to be raised, using a longer pipe in proportion if the fall is very low, and a shorter pipe if the fall is high; but it is generally preferable to have the pipe too short rather than too long, to reduce frictional losses.

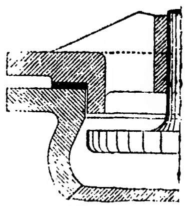

FIG. 23.

The number of beats may be as high as 200 per minute, but usually ranges between 40 and 60 per minute. If the available supply is small the stream should be gauged before selecting the

size of ram to be fitted. This can be done, with very small flows, by damming the stream and measuring into a bucket or tank the quantity of water flowing in a given time through a pipe inserted in the dam. For larger flows a weir-board should be used. The ram can be placed above ground, provided sufficient fall is obtainable, but for protection against frost is better housed in a brick-lined pit constructed in the bank of the stream and suitably covered, the floor of the pit being at such a level that the waste water drain has a fall of about 1 in 100. The ram must be truly level on its foundation.

Defective working may be due to fouling in the drive pipe, which should be periodically cleaned by drawing a scraper through it; to leaky joints in the drive pipe; to insufficient air in the air-vessel, or to a choked snifting valve; or to a fall in level of the supply, allowing air to be carried into the drive pipe.

For further notes on hydraulic rams reference may be made to a paper by E. W. Anderson in the *Proc. I.Mech.E.*, 1922, from which figs. 21 to 23 are reproduced. Some typical operating results are given in the subjoined table.

TABLE OF RESULTS OF OPERATIONS OF HYDRAULIC RAMS.

Number of Strokes.	Height of Fall.	Height of Elevation.	Water Expended.	Water Raised.	Useful Effect.
Min.	Feet.	Feet.	Cubic Feet.	Cubic Feet.	
66	10·06	26·3	1·71	·543	·9
50	9·93	38·6	1·93	·421	·85
36	6·05	38·6	1·43	·169	·75
31	5·06	38·6	1·29	·113	·67
15	3·22	38·6	1·98	·058	·35
10	1·97	38·6	1·58	·014	·18
–	22·8	196·8	·38	·029	·67

See also Descriptive Section XIX, Part III.

Clarke Chapman & Co. Ltd.
Chas. S Madan & Co. Ltd.
Pulsometer Engineering Co. Ltd.
G. & J. Weir Ltd.

SECTION XIX

PART IV

WATER TURBINES.

(By P. W. Seewer, D.E. (Zurich), M.I.C.E.)

The fundamental division is as in the case of steam and other turbines, between impulse and reaction types.

Impulse Turbines.

In impulse turbines the pressure energy of the water is transformed entirely into kinetic energy in the stationary parts of the turbine, *i.e.* the nozzles, the flow through the rotating parts is under atmospheric pressure and the forces on the rotating parts are due entirely to changes in the direction of the flow of the water, the arithmetical value of the relative velocity remaining unchanged, except for slight losses due to friction, etc. The absolute velocity of the water leaving the rotating parts is reduced as nearly to zero as possible, so that all the kinetic energy of the water is given up.

The forms of impulse turbine that have survived and are now manufactured are Pelton, and Turgo wheels for smaller power. Pelton turbines have been built to give up to 30,000 h.p. for a single wheel and jet, operating under a static head of 718 m., or 2,350 ft.

Reaction Turbines.

In reaction turbines the pressure energy of the water is only partly transformed into kinetic energy in the stationary parts of the turbine, *i.e.* the guide apparatus, and the flow through the rotating parts takes place under a varying pressure, the passages remaining full. The forces on the rotating parts are due both to changes in pressure and in the direction and velocity of flow of the water. Both the pressure and the absolute velocity of the water are reduced as the water gives up its energy to the wheel.

The surviving forms of reaction turbines now being manufactured are Francis turbines, sometimes called American or mixed-flow turbines, and of late various forms of propeller turbines both with fixed and movable blades. The latter are known as Kaplan turbines.

The Relations between Power, Speed, Head, etc.

Before it is possible to compare the various types of turbines operating under very different heads and at very different speeds, etc., it is necessary to be able to reduce all these variables to a common value.

In all turbines, both impulse and reaction, the flow through the turbine takes place at a velocity which is some definite fraction of the free spouting velocity, $\sqrt{2gH}$, where H is the head. That is, the velocity of the water, and hence, the quantity of water, Q, passed by a given turbine under different heads, varies as $H^{\frac{1}{2}}$.

At the same time the energy per unit volume of water varies directly as the head, so that the power input varies as $H^{\frac{3}{2}}$

The losses of head in any turbine vary as the squares of the velocities concerned, *i.e.* as $(H^{\frac{1}{2}})^2$ or as H. The quantity of water passing varies as $H^{\frac{1}{2}}$, and it follows that the amount of the hydraulic losses vary as the product of these, *i.e.* as $H^{\frac{1}{2}} \times H$, or $H^{\frac{3}{2}}$.

Since the power input and the losses both vary as $H^{\frac{3}{2}}$, it follows that the hydraulic efficiency is substantially independent of the head, and the power output, N, of any given turbine is the difference between input and losses and will vary as $H^{\frac{3}{2}}$.

In the same way the peripheral velocity of any rotating part bears a definite relation to the free spouting velocity, and hence will vary, as $H^{\frac{1}{2}}$. The speed, n, of any given turbine therefore will vary as $H^{\frac{1}{2}}$.

For homologous turbines of different sizes under the same head, since the velocity of the water will be the same proportion of the spouting velocity, the discharge will vary as the square of the linear dimensions, *i.e.* as D^2. The power will also, therefore, vary as D^2.

The peripheral velocity of the wheel will be some definite fraction of the spouting velocity, that is, the speed in r.p.m. will vary inversely on the linear dimensions, *i.e.* as $1/D$.

For homologous turbines under various heads, therefore, we have the following relations:

The quantity discharged, Q, varies as $H^{\frac{1}{2}} \times D^2$.

The power developed, N, varies as $Q \times H$.

$$H^{\frac{3}{2}} \times D^2.$$

The speed, r.p.m., n, varies as $H^{\frac{1}{2}} \times 1/D$.

Specific Speed.

The fundamental basis of comparison between all forms of turbines is a figure known as the Specific Speed, which has a constant value for every turbine of a homologous series. It may be defined as the speed at which the turbine would run (at its designed efficiency) under unit head when reduced in size so as to produce unit power under that head.

If metric units are taken for power and head the specific speed is in metric units, and if h.p. and feet are taken the specific speed is in English units. The specific speed, as found in metric units, has to be divided by 4·446 to reduce it to English units.

The following figures are also sometimes used:

Unit speed n_1 = speed of actual wheel under unit head.

$$= \frac{n}{\sqrt{H}}.$$

Unit power N_1 = power of actual wheel under unit head.

$$= \frac{N}{H\sqrt{H}}.$$

Unit quantity Q_1 = quantity passed by actual wheel under unit head.

$$= \frac{Q}{\sqrt{H}}.$$

Then specific speed, $n_s = n_1\sqrt{N_1}$

$$= \frac{n\sqrt{N}}{H^{\frac{5}{4}}}.$$

Where n = speed of wheel in r.p.m.
N = output of wheel;
Q = quantity passed by wheel;
H = head on wheel.

Choice of Type of Turbine.

When a given scheme is being considered in general the practical and economic output and speed are only variable within limits. From these, and the head, the specific speed of the proposed turbine can be found, and hence its type settled. Whether the type thus arrived at is possible or not, can be checked by the turbine makers, who have curves showing the highest specific speed that can safely be employed at a given head, and with given suction conditions, etc. The modern very high specific speed types have been developed in order that the speed of large units at very low heads may be as high as possible, and the cost of the generators and all appurtenances kept down, so as to make the schemes attractive from the financial standpoint.

The ranges of specific speeds covered by the various types of turbines are as follows :

n_s (Metric Units).	Type of Turbines.	
Up to 15	Single jet	Normal type impulse wheel, proportioned for best efficiency.
15 to 21	2 jet	
21 „ 26	3 „	
26 „ 30	4 „	
15 to 21	Single jet	High-speed modern type. Impulse wheels.
21 „ 30	2 jet	
30 „ 36	3 „	
36 „ 42	4 „	
50 to 70	Very low speed	Francis.
80 „ 100	Low speed	
100 „ 200	Medium speed	
200 „ 400	High speed	
400 to 600	Propeller type.	Fixed vanes.
600 „ 800 and above	Propeller type.	Kaplan types with movable vanes.

It will be seen that a gap exists between the highest specific speed of the normal type of Pelton turbine and the lowest specific speed of the normal type of Francis turbine. Such machines as the Turgo Wheel are intended to bridge this gap, but as yet no really satisfactory solution has been found for large powers.

PELTON TURBINES.

These consist of a jet, or jets, striking a series of buckets mounted on a disc. The buckets consist of two ellipsoids meeting in a central splitter edge, and have a portion removed on the outside so that the jet always strikes this splitter edge on one or other of the buckets, and thus meets the buckets without shock losses at entrance. The water is divided equally by the splitter edge, and diverted through approximately 180° by the bucket, so that it leaves in a direction directly opposed to the motion of the wheel. The action is shown diagrammatically below :

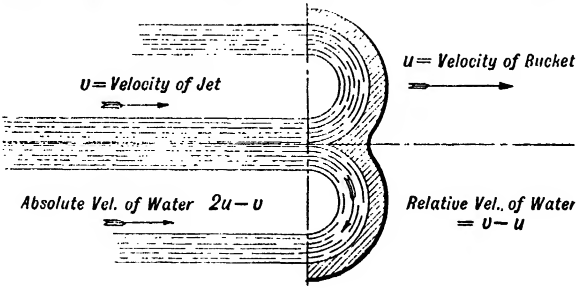

FIG. 1.

If the velocity of the jet is v and the velocity of the bucket u, the relative velocity of water to bucket is $v - u$.

The absolute velocity of the water leaving the bucket, measured in the direction of the movement of the bucket, is $u - (v - u)$, or $2u - v$. By making $u = \frac{1}{2}v$, this absolute velocity of the leaving water is reduced to zero and all the energy of the water is given to the wheel. This means that the peripheral velocity of the buckets, measured on the circle tangent to the jet, should be half that of the jet, and this is the figure usually adopted in practice. This imaginary circle is usually known as the pitch circle of the wheel.

The general proportions of a Pelton turbine may be taken as follows

D = diameter of pitch circle;
d = diameter of jet.

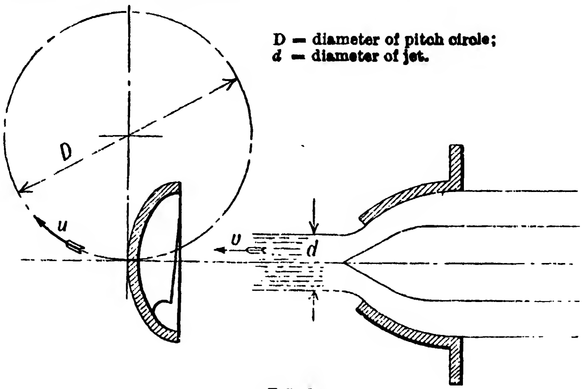

FIG. 2.

Velocity of jet	$v = 0{\cdot}95$ to $0{\cdot}98\sqrt{2gH}$.
Peripheral velocity of wheel (measured on pitch circle) . . .	u = approximately $\frac{1}{2}v$.
	$= 0{\cdot}44$ to $0{\cdot}48\sqrt{2gH}$.
Speed in r.p.m.	$n = \frac{60u}{\pi D} = 26$ to $29\frac{\sqrt{2gH}}{\pi D}$
The quantity of water	$Q = \frac{\pi d^2 v}{4} = 0{\cdot}24$ to $0{\cdot}245\ \pi d^2\sqrt{2gH}$.

For best efficiency the ratio D/d should lie between 14 to 1 and 10 to 1, but wheels have been built with this ratio up to 30 to 1, in exceptional cases.

Values of this ratio below 10 to 1 need very careful design, successful wheels have been built with this ratio down to 7·5 to 1, with only a small sacrifice of efficiency.

For large machines, under good conditions, efficiencies as high as 90 per cent. have been achieved.

The efficiency of a well-designed machine is constant over a large range of load and varies by only 1 per cent. to 3 per cent. between half-load and full load.

For horizontal-shaft machines, either one or two jets per wheel is common, and three and four jets have been used. Vertical-shaft machines, having 3 and 4 jets per wheel, have been built with complete success.

For multi-jet machines it is necessary to make a careful study of the paths of the water relative to the wheel in order to ensure that the jets do not interfere with one another. The angles between the jet of a multijet machine should usually lie between 60° to 90°.

BUCKETS.

The proportions of a typical bucket are shown below:

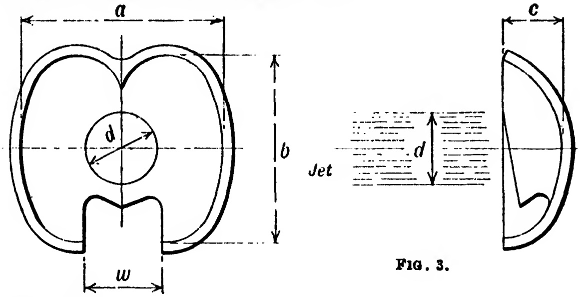

FIG. 3.

$a = 3 \cdot 0$ to $3 \cdot 5d$.
$b = 0 \cdot 8 \times a$.
$c = 0 \cdot 27 \times a$.
$w = 1 \cdot 2 \times d$.

The exact shape of the bucket varies considerably for various makers, and much study and many experiments are necessary to arrive at the best form of bucket for any given case.

NOZZLES.

The section of all modern nozzles is circular and the size of jet is regulated by a spear operated either by hand or by an oil- or water-pressure servomotor.

Considerable experiment is necessary to arrive at the best form of nozzle and spear to give a true steamline jet, particularly at the higher heads. The discharge co-efficient measured on the minimum area at the mouth of the nozzle usually lies between 0·75 and 0·82 when the spear is fully open, and increases gradually to about 0·95 as the spear approaches the closed position. It is most important that the area of the water passage should decrease steadily towards the outlet.

Typical proportions are shown below:

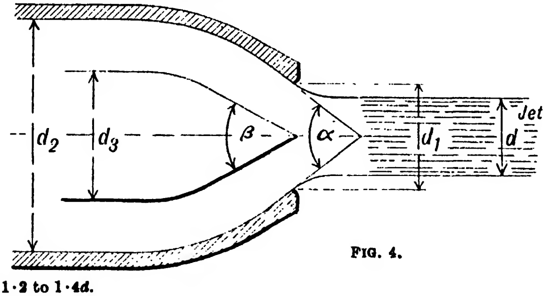

FIG. 4.

$d_1 = 1 \cdot 2$ to $1 \cdot 4d$.
$d_2 = 3 \cdot 0$ to $4 \cdot 0d$
$d_3 = 1 \cdot 25$ to $1 \cdot 5d$.
$\alpha = 60°$ to $90°$.
$\beta = 40°$ to $60°$.

REGULATION.

Unless the pipe-line is very short compared to the head, direct regulation by means of the spear will cause severe water hammer when large loads are thrown off suddenly. To prevent this the spear is usually arranged to move so that the velocity of the water in the pipe is changed only very slowly, and special devices are used to cut the water off from the wheel when load is thrown off until the quantity of water flowing down the pipe is correct for the new load.

This diversion of the water can be achieved in three ways:

(1) *Relief Valve Device.*—An auxiliary nozzle is arranged to open simultaneously as the main nozzle closes, and then this nozzle is closed very slowly under the control of a dashpot.

(2) *Deflector Device.*—A deflector is arranged in front of the nozzle which moves to cut off the jet from the wheel until the spear, which moves slowly under the control of a dashpot, has reached the position corresponding to the new jet. The link gear between the deflector and spear is so arranged that the deflector just clears the jet when the spear reaches its final position.

(3) *Diffusing Device.*—Small, inclined blades are made to protrude from the spear, and cause the jet to whirl and break up into spray, until the spear, moving slowly under the control of a dashpot, reaches the position corresponding to the new jet. These blades lie just flush with the spear in the final position.

One or other of the two last is the usual European practice.

In a well-designed system the governor can be arranged to operate a deflector or spear, and relief nozzle in 1·8 to 2 secs., and a Seewer governor device in ½ to 1 sec.

The speed rise will depend on the flywheel effect of the set, but the following figures are usual ;

Load thrown off. Per cent. of full load.	Momentary speed rise above normal steady speed after drop in load in per cent. of normal speed.	
	Deflector or spear and relief spear.	Diffuser.
100 per cent.	15 per cent. to 25 per cent.	5 per cent. to 10 per cent.
75 „	12 „ „ 19 „	4 „ „ 8 „
50 „	8 „ „ 13 „	3 „ „ 6 „
25 „	4 „ „ 6 „	1 „ „ 2 „

For load thrown on the conditions are dictated entirely by the design of the pipe-line, since any attempt to cause the water column in the pipe to accelerate suddenly will cause so great a drop of head that no greater power is available at the jet, and dangerous pressure waves are set up.

Unless the conditions of operation are studied very carefully in the design of the pipe-line it will be impossible to throw load on the set quickly, and any attempt to adjust the governor to enable this to be done will only result in hunting and dangerous surges in the pipe-line.

Reaction Turbines.

All reaction turbines consist fundamentally of four main sections, *i.e.* casing (open or closed), guide apparatus, runner wheel and suction tube. The proportions and arrangement of these differ considerably for various conditions of speed and head. The two extreme cases of a very low specific speed Francis turbine and a high specific speed propeller turbine are shown in fig. 5.

It will be seen that there is a ring of guide vanes surrounding the runner, from which the water discharges through the runner into a tapering tube known as the draft tube or suction tube.

The guide vanes are arranged to swivel to control the amount of water reaching the runner, either by links interconnecting them directly or by levers on their spindles and outside the water stream. The latter method is more expensive, but is generally adopted in modern machines.

By the use of the draft tube the turbine can be set above the level of the tail water without loss of head due to its elevation, and at the same time a large part of the kinetic energy of the water leaving the runner can be recovered by giving the draft tube a suitable taper. In low specific speed turbines this recovery is not of great importance, but in high specific speed turbines as much as 25 per cent. of the energy of the water is recovered in the draft tube, and its design and efficiency is consequently of great importance.

The shaft of a reaction turbine may be horizontal or vertical with an improvement of 1 per cent. or 2 per cent. in the efficiency for the vertical shaft type—especially for high specific speeds—due to the absence of a suction bend near the runner.

Type of Turbine Setting.

The method by which the water reaches the turbine proper determines the type of setting.

(1) *Open Flume.*—In the case of low heads the turbine is made without a casing and is merely set at the bottom or side of an open chamber as shown in fig. 6.

(2) *Cased Turbines.*—When the head increases, and an open flume setting is no longer possible, or when an open flume setting is inconvenient for local reasons, the turbine is encased, and the water led to it under pressure by a duct or pipe.

(*a*) *Cylindrical.*—Typical arrangements are shown in fig. 7.

As will be seen, the turbine is entirely enclosed. The pipe bringing the water to the casing may be on the side, or on the end.

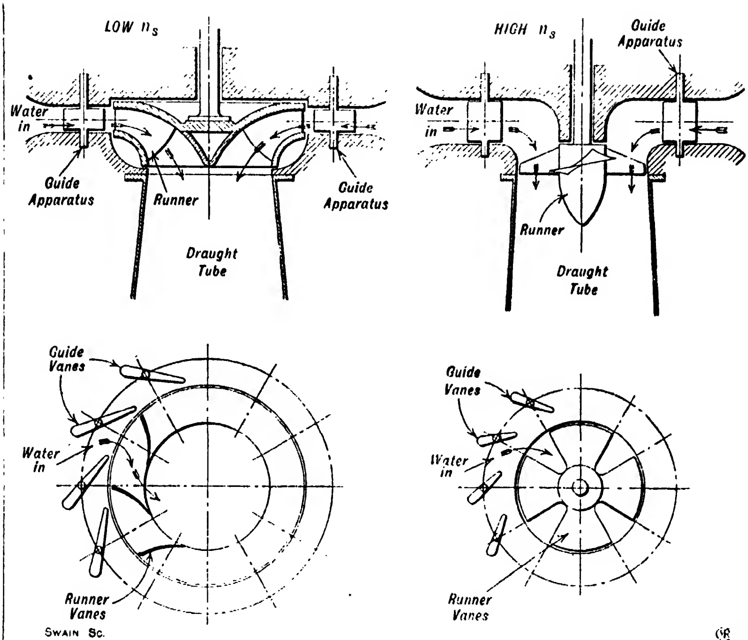

FIG. 5.

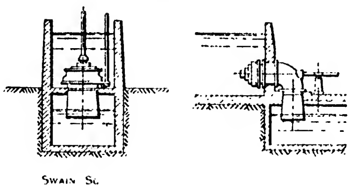

FIG. 6.

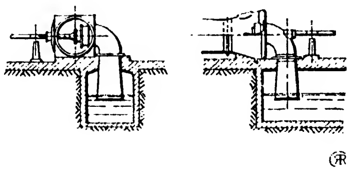

FIG. 7.

(b) *Spiral*.—This is the general form adopted for modern machines, the shape of the casing being such that the water velocity varies throughout according to the rules of free vortex flow, *i.e.* the velocity is inversely proportional to the radius. The casing may be of cast iron, cast steel, plate steel welded or riveted, or may merely be formed in concrete. When possible the casing should be designed so that the water is gradually and constantly accelerated up to the point where it meets the runner. A typical spiral turbine is shown below.

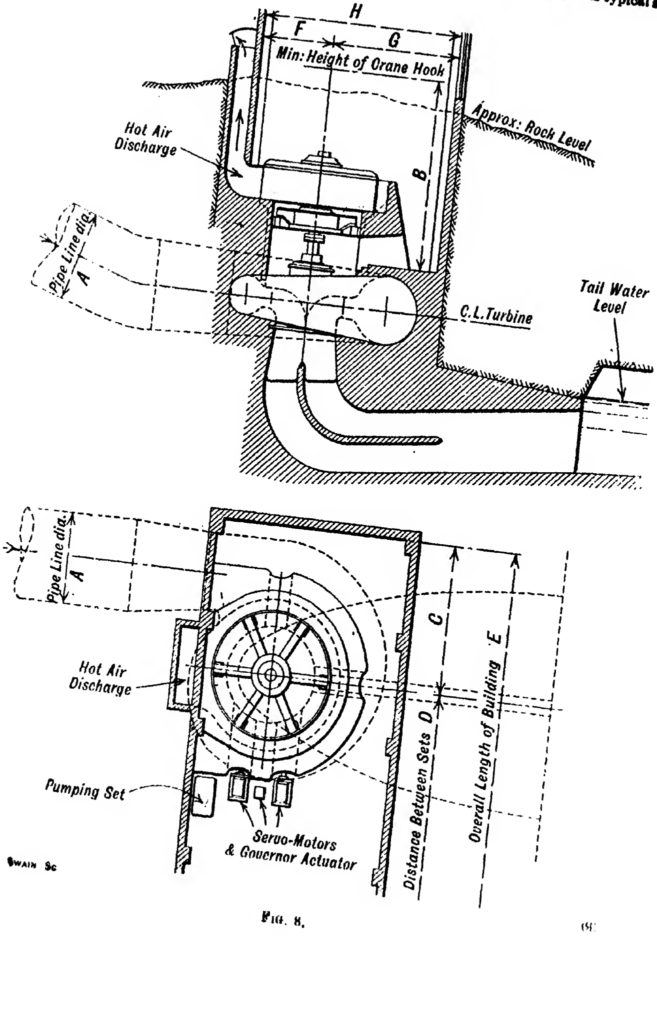

FIG. 8.

General Proportions of Reaction Turbines.

The following table, together with the figures attached, will enable the general proportions of runner for any specific speed to be determined. All velocities are given in terms of the spouting velocity, $\sqrt{2gH}$.

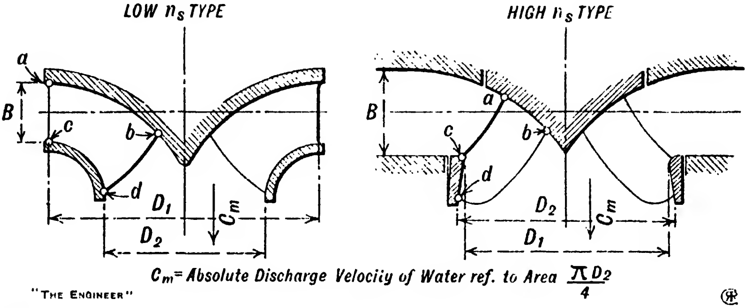

Fig. 9.

Specific Speed in Metric Units.	100	150	200	250	300	350	400
Ratio B/D_1	0·12	0·20	0·27	0·33	0·35	0·35	0·35
Velocity C_m	0·18	0·21	0·23	0·28	0·32	0·34	0·33
Peripheral velocity of point							
(a)	0·70	0·72	0·74	0·76	0·77	0·76	0·72
(b)	0·34	0·33	0·33	0·33	0·33	0·34	0·35
(c)	0·70	0·73	0·76	0·82	0·92	1·05	1·22
(d)	0·51	0·66	0·80	0·88	0·98	1·13	1·30

Specific Speed and Efficiency.

While the maximum efficiency of a turbine differs within a few per cent. only whatever the specific speed, the shape of the efficiency curve differs considerably. As the specific speed increases the curve becomes more peaked, and the efficiencies on either side of the optimum decrease. Modern development is tending to increase the range of loads over which the efficiency is high, but if good part load efficiencies are required and a good overload capacity, it is not advisable to install a turbine with a specific speed higher than about 300 (metric). On the other hand, the higher specific speed turbines are less sensitive to changes in the head under which they are running, and for a large scheme, it is advisable that the turbine makers should be consulted early in its preparation in order that the best speed and size of unit and arrangement of station may be chosen.

Regulation.

The regulation of reaction turbines is achieved by moving the guide vanes to admit more or less water. As in the case of impulse wheels, the speed at which this can be done depends on the pipe line conditions. For long pipe lines the rate at which the pipe flow can safely be decreased is so slow that for good regulation it is necessary to by-pass the water. A relief valve is then provided which opens simultaneously as the turbine guide vanes close, and which closes subsequently slowly under the control of a dashpot. For load thrown on, the pipe line conditions determine the rate of opening of the turbine guide vanes, and consequently the momentary drop of speed of the sets.

It is usually impossible to close the turbine in less than 2 secs., whatever arrangements are made to by-pass the water, and the flywheel effect of the set must be chosen so as to keep the momentary speed fluctuations to a figure consistent with sound commercial and industrial practice for the type of load on the station.

Governors.

Of all the types of governor tried at various times for water turbines only one type has finally survived, this is the automatic oil-pressure operated servomotor type. In general the forces required to move the control gear of turbines are far too large to enable a plain centrifugal ball type of governor to do the work, and one or several relays are consequently always required. Modern governors always consist of some form of centrifugal pendulum which either directly or through relays controls the admission of oil under pressure to one or other side of a cylinder in which moves a piston connected to the control gear of the turbine. In many cases some form of constant pressure, *e.g.* a spring, or piston subjected to water pressure, is used to close the turbine in order that a failure of the oil-pressure supply will not result in a runaway. In order that the turbine may be able to drive a generator in parallel with the other sets and that the hydraulic conditions may be fulfilled, the following adjustments are required on the governor:

(1) *Steady Speed.*—It must be possible to adjust the steady speed of the set at any load, within limits of about 5 to 10 per cent. up and down.

(2) *Permanent Speed Rise.*—In order that two or more sets when running in parallel should be able to share the load, a certain increase in steady speed is needed from the full load to no load, usually 2 or 3 per cent. This increase in the steady speed of the set from full load to no load should be adjustable from zero to approximately 5 per cent. of the normal steady speed. In some special cases a negative speed rise, *i.e.* a drop in speed from full load to no load is advisable. In any station the machine with the smallest permanent speed rise will carry the largest proportion of the load changes.

(3) *Closing and Opening Times.*—These must be capable of accurate, independent adjustment in order to get the closest possible regulation without exceeding the permissible pressure rise and pressure drop figures in the pipe line or ducts.

(4) *Temporary Statism.*—In order to prevent hunting, the governor should be arranged to approach its final position asymptotically. This implies some form of non-rigid return-motion gear, which tends temporarily to bring the valve controlling the movement of the main servomotor piston to its neutral position as soon as the gates begin to move, thus damping the oscillations of the gates about their final position. The degree of this damping should be adjustable in order that the governor should be as sensitive as possible without causing hunting.

Governor Power.

There are many formulæ for the governor power required to control any given turbine. For all large machines, the final choice of governor is made by calculating the water and friction forces acting on the moving regulating parts in every position. A large margin, of the order of 100 to 200 per cent. of the force thus calculated, is always provided in order that the acceleration of the moving masses may be rapid.

The governor power may be calculated approximately by the following formula:

$$P = K \frac{N}{\sqrt{H}}$$

when P = power of governor in KgM;
N = output of turbine in h.p.;
H = head on turbine in metres;
and K is a constant depending on the turbine.

Values of K are approximately as follows:

Impulse Turbines.	Seewer governing device only	K = 1·0.
„ „	Deflector only . .	K = 2·0.
„ „	Spear only	K = 2·0.
Reaction turbine.	High head	K = 2·5.
„ „	Low head	K = 3·5.

Small changes in the design of the spear in the case of impulse wheels, or of the guide vanes in reaction turbines, can change the power required for regulation by 100 per cent. or more, and the makers should always be consulted in any given case.

Momentary Speed Changes.

Load Off.—To calculate the momentary speed rise of a set when load is thrown off it is necessary to know the flywheel effect. This is generally measured in metric units as kilogrammes × (diameter in metres)² and written GD². In English units it is measured as lbs. × (radius in ft.)² and written WR². It is arrived at by summing the products of weight × (diameter)², or weight × (radius)² for all parts of the set.

A flywheel effect of 1KgM² GD² is equivalent to 5·92 lbs. ft.² WR².

The speed rise may be calculated as follows:
Let

N_1 = original load in h.p.;
N_2 = final load in h.p.;
n_1 = original steady speed;
n_2 = maximum momentary speed;
t = closing time of turbine in seconds;
GD^2 = flywheel effect in KgM^2.

Then

$$\left(\frac{n_2}{n_1}\right)^2 - 1 = \frac{\left(\frac{520}{n_1}\right)^2 \times (N_1 - N_2) \times t}{GD^2}$$

Since the governor will be set to give some permanent speed rise, when the load is steady at N_2 the set will run at some new steady speed, n_3 differing slightly from n_1.

The percentage speed rise quoted in regulation guarantees is the percentage rise in speed above the *final* steady speed, and is, therefore,

$$\frac{n_2 - n_3}{n_1} \times 100.$$

In actual cases there are two corrections to be taken into account. These are:

(1) The drop in efficiency as the set speeds up. This tends to reduce the momentary speed rise.

(2) The pressure rise in the pipe line when the water quantity passing is decreased. This tends to increase the momentary speed rise.

For preliminary estimates these corrections are usually assumed to cancel out, but before giving any final regulation guarantees a careful study is made, and in extreme cases a speed-time curve is plotted, and all factors taken into account by arithmetical integration, using the time and instantaneous torque to find the angular acceleration at each successive instant.

Load On.—For the change of speed when load is thrown on the formula given above for load off becomes:

$$1 - \left(\frac{n_2}{n_1}\right)^2 = \frac{\left(\frac{520}{n_1}\right)^2 \times (N_2 - N_1) \times t}{GD^2}$$

and if n_4 be the new steady speed, the percentage speed drop is given by $\frac{n_4 - n_2}{n_1} \times 100$. In general, however, the correction factors become so large that the results found from the formula above are not sufficiently close to the actual results in the field, and can only be regarded as approximate. If a true figure is required it is necessary to plot a speed-time curve, finding the value of the torque and hence the instantaneous angular acceleration at successive intervals, and obtaining the speed by arithmetical integration.

Pipe Lines and Accessories.

Except in the simplest possible cases, a water power station consists of much more than the turbines, generators and governors. In fact, in large schemes the cost of these is only 10 to 20 per cent. of the total cost of the scheme.

The modern scheme will usually consist of an intake headrace, or a canal or tunnel, leading the water to an open forebay, from which the pressure pipes feed the turbines.

Sluice Gates are necessary at the canal or tunnel intake, and either sluice gates or valves at the head of the pipes, and in many cases at the power-house.

Strainer Racks should be provided at the pipe intakes to prevent injurious floating and suspended matter from damaging or choking the turbines. The spacing of the bars in the racks should be such that anything which will pass between them will also pass through the turbines without jamming.

Pipe Lines should be designed with an ample factor of safety to provide for accidental water hammer effects, in addition to the normal designed pressure rises. A point that must not be overlooked is that a pressure rise is propagated as a wave from the downstream end, and has a value which is a proportion of the static head at that end. The intensity of the pressure wave diminishes as it approaches the free, upstream-water surface, but the pressure attained at any point is not merely proportional to the static head at that point. If a pipe line runs for some distance at a small slope and then dips steeply, while the static pressure in the upper part is always small, the pressure waves will still have nearly their initial intensity at the top of the steep slope, and so may raise the pressure in the pipe to several times its static value.

Valves are necessary in most cases just upstream of the turbines themselves. They form a vital safety factor in the event of damage to the plant, and are required in many cases for dismantling purposes. In addition it is inconvenient to have to empty the pipe line every time it is necessary to inspect the turbines. When more than one turbine is fed from one pipe a valve upstream of the turbine is essential. For low heads either butterfly or sluice valves are used, for higher heads sluice valves, or some form of balanced stream-line valve, such as the Larner-Johnson or the English Electric type. The valves may be arranged for either hand, electric or hydraulic operation, and it is necessary to ensure that it is impossible to close them so fast as to cause water hammer in the pipes. The closing characteristic should be such that the valve slows down as it approaches its closed position, as it is the proportional rate of closing, *i.e.* the percentage reduction of quantity, that decides the consequent pressure rise.

Expansion Joints must be provided between the pipe anchorages to deal with expansion and contraction due to temperature changes. It should be remembered that an empty pipe exposed to the direct rays of the sun will reach a temperature far above that of the surrounding air, and unless the pipes are buried this effect must be carefully considered.

Anchor Blocks must be provided to take the hydraulic thrusts and the weight of the pipe, and the pipe must be sufficiently solidly anchored to prevent any loads being transmitted to the turbines. It is usual to provide feet and flanges on the pipes, and to cast round them large concrete blocks which are capable of resisting by their weight alone the force applied to them. When the foundations are solid rock it is possible to obtain good anchorage for the pipes without the use of heavy anchor blocks.

Pressure Changes in the Pipe Line.

Since so much in the design of water turbines depends on the pressure conditions in the pipe line, a theory that enables these to be calculated for any changes in the flow is very necessary. Until recently no such theory existed, and the theories that were advanced led to results little in accordance with the experimental facts. No approximate methods of calculating the pressure rises or drops, due to changes of the velocity of flow, are sufficient or satisfactory.

A theory due to an Italian engineer, Sig. L. Alliévi, has rendered it possible to calculate with a high degree of accuracy what will be the result of any change in the conditions of flow in a pipe line, and for information on this subject his work should be consulted.

It is most important in any scheme that the turbine makers be consulted before the final layout of the pipe line is settled, or the hydraulic conditions may subsequently make good regulation under commercial loads difficult or impossible.

SECTION XX

PART I

BALL AND ROLLER BEARINGS

(Specially Contributed.)

BEARING PRESSURES AND DIMENSIONS

(Contributed by H. W. Swift, D.Sc., M.I.Mech.E.)

(pp. 849–870)

PART II

LUBRICATING OILS AND LUBRICATION (pp. 871–881)

(Contributed by Harold Moore, M.Sc.Tech., F.C.S., and E. Stokoe, M.Inst.Pet.)

PART III

VISCOSITY AND VISCOUS FLOW OF OILS (pp. 883–888)

(Contributed by Harold Moore M.Sc.Tech., F.C.S., and E. Stokoe M.Inst.Pet.)

PART IV

SHAFTING AND SHAFTS (pp. 889–894)

(Contributed by H. W. Swift, D.Sc., M.I.Mech.E.)

PART V

SPRINGS (pp. 895–901)

(Contributed by R. G. Batson, M.I.C.E.)


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SECTION XX

PART I

BALL AND ROLLER BEARINGS

Ball and roller bearings are made in a number of types and designs that consist primarily of two race rings, between which are interposed the rolling elements, these latter being usually held in a form of cage or separator.

The rolling elements, which are often made from direct hardening steel, may consist of balls, parallel rollers, taper rollers or barrel-shaped rollers. Some tapered rollers are made also of case-hardening steels of the chrome, nickel-molybdenum, and nickel-chrome types selected according to operating conditions.

The balls employed may be regarded as perfect spheres that are manufactured to a tolerance of plus or minus ·0001″, although to ensure more even distribution of the load over the balls in any one bearing, the balls should be graded to a considerably finer tolerance.

Parallel rollers are made either from the solid or from flat steel wound spirally, similar to a helical spring; the latter, known as spiral rollers, have the merit of slight flexibility.

With parallel rollers, whilst in the majority of types the length is equal to the diameter, rollers where the length exceeds the diameter are not infrequently employed, whilst in other types, the length may be less than the diameter.

This type of roller is made to the closest limits of accuracy for parallelism, and the diameter should not vary by more than ·0001″.

The ends of the rollers are generally flat and parallel, although some types have rounded ends, whilst in other cases the ends are recessed, or may carry a projection to suit the type of cage employed.

A common tolerance for the length of rollers is $\pm$·0002″, although where the length considerably exceeds the diameter, this may be greater.

Taper rollers, as the name implies, are made to a predetermined taper, the ends of the roller being ground parallel, or in some instances, they have a slight radius. Others are shouldered at the ends for the purpose of guiding the rollers in the bearing.

Spherical or barrel-shaped rollers vary from the type where a perfect symmetrical barrel is employed, to a type where the maximum diameter of the barrel is not coincident with the centre line of the barrel.

Race rings are, in the majority of instances, made with parallel fitting seatings, *i.e.* bore and outside diameter, whilst the path on which the rolling elements run usually conforms to the shape of these members.

The rings are made either from direct hardening steel, or from case-hardening material, and the hardness obtained with either material should be approximately 600 Brinel, whilst with case-hardened material, the depth of the casing should be about 2 m/m. and can extend to as much as 6 m/m. in the case of the largest size tapered roller bearings used for rolling mills.

The cages, or retainers, can be made from steel, brass or bronze, machined from the solid bar or tube, or pressed from sheet steel or brass. Bakelite or duralumin are sometimes used for high-speed bearings where reduction of weight is of vital importance. Cages can be made to locate on the rolling elements, on the shoulders of the inner ring, or in the bore of the outer ring; balance and concentricity being essential features in cage construction. The cages in some roller bearings in principle consist of two side plates separated by rods or rivets. These side plates can have holes or projections to locate the ends of the rollers, or in the case of hollow rollers, the rods or rivets may pass through the roller.

Standard ball and roller bearings are, in the majority of instances, made to internationally recognised sizes (see tolerances in Table I on p. 856), but the number and size of balls and rollers, etc., varies in accordance with the practice of different manufacturers.

Bearing capacities are directly governed by the number and size of the rolling elements, and it is generally the practice to fit the largest number of balls or rollers having the maximum diameter, at the same time retaining a section through the race rings that will adequately withstand the abuse to which they may be subjected when being fitted.

The closest limits of accuracy are adopted for track diameters; the resultant degree of accuracy may be as close as ·0001″ and where bearings are manufactured on a production basis, this limit is generally obtained by grading the track diameters of the inner and outer rings and pairing these together.

BALL JOURNAL BEARINGS.

The most common form of bearing under this heading is the single-row rigid type ball journal bearing (see fig. 1).

The balls run in formed tracks, the radius of the track being slightly larger than that of the balls.

Two methods are available for inserting the balls between the race rings. By placing the inner ring eccentric to the outer ring, the balls can be inserted as shown at fig. 2. The race rings are then brought to their correct position, and the balls evenly spaced between the two rings, the space between the balls being subsequently filled by the cage.

The alternative method is to employ a filling slot, or notch, that is formed in the side of the races, through which the balls are inserted.

The filling slot should not be as deep as the tracks (this in some measure prevents the balls contacting with the edge of the filling slot when the bearing is under load), mechanical means being adopted to temporarily distort the rings to permit the balls being inserted. This enables a greater number of balls to be used than can be obtained by the former method of assembly, and theoretically, this results in a greater load carrying capacity for the bearing.

Owing to the irregularity in the section of the race rings that is caused by the filling slot, the advantage gained by the additional number of balls is sometimes lost, through the distortion of the tracking surfaces that is likely to occur against the filling slot. Further, should the bearing be submitted to thrust duties, there is always the danger of the balls fouling the filling slot, and therefore bearings of this design are only desirable where journal loads are present.

For general purposes, the non-filling slot type is usually adopted.

After insertion of the balls, the cage, which is generally in two halves, is inserted, and secured in position by rivets or other means.

A variation of the above type of bearing is that where no cage is employed, the races being crowded with balls, and a filling slot used for the insertion of the balls. This type is usually employed with very light sectioned race rings, and for lightly loaded applications.

The non-filling slot type, whilst being primarily intended for journal loads, can also be used for combined journal and thrust loads, or thrust loads only. The proportion of thrust to journal loads that these bearings will carry varies with the different makes of bearings, and individual manufacturers should be consulted. It can be assumed that the thrust rating would vary in accordance with the diametrical freedom or slackness in the bearing. This allows end movement, and therefore a bearing with the maximum degree of end play would give the maximum angle of contact between the balls and tracks, and would have the maximum thrust rating, which would be at least equal to its radial rating. (It should be noted that theoretically all the balls are under load when thrust is applied, whereas only a proportion of the balls are loaded when the bearing is subject to journal duty only.)

A variation of the single-row bearing is the double-row rigid type of journal bearing shown at fig. 3. In effect, this is two bearings of the single-row type running side by side. Filling slots are usually employed to assemble the balls in the bearing.

Although theoretically this bearing should have a 100 per cent. greater load-carrying capacity than the single-row bearing of equal size, variations between track diameters due to manufacturing difficulties reduce the capacity somewhat, and the bearing can therefore only be rated as having approximately 75 per cent. greater journal capacity than a single-row ball bearing.

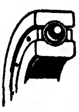


FIG. 1.

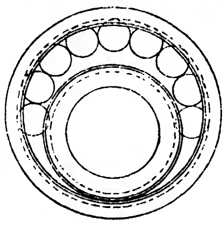


FIG. 2.


FIG. 3.

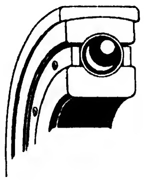


FIG. 4.

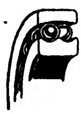


FIG. 5.

Where misalignment is likely to be caused between two bearings, this is catered for with the bearings previously described by spherically forming the outside diameter of the bearing, and fitting this into a corresponding spherical seat (see fig. 4). This increases the outside diameter of the complete unit by approximately twice the thickness of the section of the outer ring.

Another form of self-aligning bearing, which is made interchangeable with the overall dimensions of single-row ball journal bearings, is illustrated at fig. 5. The track in the outer ring is ground to a sphere, the centre of which coincides with the centre of the bearing bore. Two distinct tracks are formed in the inner ring, and two rows of balls are employed. It should be noted, however, that although two rows of balls are used, when the bearing is of the same overall dimensions as a single-row ball bearing, the balls are of a smaller diameter. This, combined with the smaller area of contact between the balls and the track in the outer ring, results in the load-carrying capacity being no greater than the single-row type. The bearings will withstand a proportion of thrust loading, although it is preferred that before adopting this type for combined journal and thrust loads, reference should be made to the manufacturer.

Self-aligning bearings are not intended to accommodate misalignment, which results in constant oscillation in the bearing, and no bearing will give efficient service under these conditions. Where, however, it is essential to cater for this, then double-row self-aligning bearings, as shown at fig. 5, should be adopted.

The single-row angular contact bearing shown at fig. 6 is a bearing specially manufactured to withstand thrust, or a combination of journal and thrust loads. In design this varies from the single-row rigid type bearing by the use of a one-piece cage, a greater number of balls, and that the track is cut away on one side to permit assembly of the balls and cage. The cage being in one piece allows the maximum number of balls to be inserted, and by providing for the balls to contact at an angle with the tracks, the maximum thrust capacity is obtained.

In providing for the angular contact of the balls, it is essential that an initial slackness in the bearing is allowed. The bearings, however, are intended to be mounted so that the thrust load is of sufficient proportion that the balls are held constantly in their correct tracking position, or alternatively, means of adjustment must be adopted to ensure this. These bearings will carry any proportion of journal and thrust loads, or thrust loads only which are equal to or greater than the rated journal load carrying capacity of the bearings, and are particularly suitable for thrust loads at high speeds.

In common with single-row ball journal bearings, angular contact bearings are also made in the double-row series. Fig. 7 shows the simplest form, where a one-piece inner ring carries the two rows of balls. The outer ring is divided into two pieces, each carrying a separate track, and each row of balls is held in a one-piece cage. These are made so that when mounted, and the outer rings are clamped together, correct tracking of the balls is ensured. It is therefore essential that the two outer rings should be clamped.

A similar bearing, but having a solid outer as well as a solid inner ring, is shown at fig. 8. In this type the balls again track at an angle. The bearings can be assembled by means of a filling slot, or can be of the notchless variety. Where filling slots are employed, these are so arranged that the slot is shallow in the outer ring and deeper in the inner ring, to prevent the balls fouling these points. The bearings can be used for pure journal loads as well as combined journal and thrust loading.

Bearings shown at figs. 7 and 8, whilst having an increased load-carrying capacity over single-row ball bearings, do not have a corresponding increase in the thrust rating. When the bearing is subject to thrust loads, one row of balls becomes more heavily loaded, whilst the other row is relieved of load.

A further design of angular contact bearing is shown at fig. 9. Four distinct tracks are provided, two in the outer, and two in the inner rings. A one-piece cage is employed, thus permitting the use of the maximum number of balls. It is essential for assembling the bearing that either the inner or outer rings are in two parts, which are subsequently held together by means provided in the method of mounting the bearing. This bearing is used for combined journal and thrust loads, or thrust in both or either directions. In order to ensure two point tracking, it should only be used under those conditions where the thrust is always of higher value than the journal load. In this case a slight axial freedom in the bearing will ensure clearance between the balls and the tracks opposite the points of contact. This, in common with all angular contact bearings, when used as a double thrust unit, is capable of dealing with much higher speeds than the standard type of thrust bearing, owing to the fact that the balls are restrained by the tracks against centrifugal loading.

Roller Journal Bearings.

When loads are greater than ball bearings will withstand, roller bearings are to be preferred, the load-carrying capacity of the roller bearing varying from 50 to 100 per cent. (according to speed) greater than the corresponding size of ball bearing.

The most common types are shown at figs. 10 to 22.

Fig. 10 illustrates the solid roller type, where the rollers have a length equal to the diameter, The rollers are separated in the bearing by a cage which also retains the rollers in position on the inner ring when the outer ring is removed. Guiding shoulders, which must be square with the tracks, are provided in the inner ring.

In some cases it is an advantage for the guiding shoulders to be in the outer ring, and fig. 11 illustrates this type.

Other designs of roller bearings employing a similar type of roller are fig. 12, a double-row roller bearing, fig. 13 a roller bearing where the length of the roller exceeds the diameter, and fig. 14 a bearing made with multiple rows of rollers.

Similarly to single-row ball journal bearings, roller journals can be fitted in spherical seatings to provide for self-alignment of the bearing (see fig. 15).

A spiral roller bearing is shown at fig. 16. The hollow spiral roller permits of slight flexibility, and by the use of left- and right-hand spiral rollers throughout each bearing, the tendency for the rollers to skew is overcome.

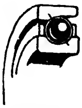


FIG. 6.

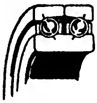


FIG. 7.

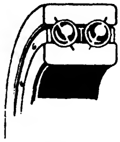


FIG. 8.


FIG. 9.

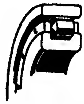


FIG. 10.


FIG. 11.

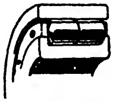
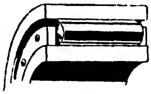


FIG. 12.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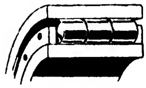


FIG. 14.

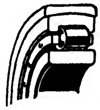


FIG. 15.

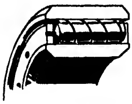


FIG. 16.

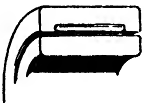


FIG. 17.

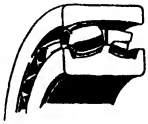


FIG. 18.

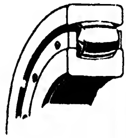


FIG. 19.

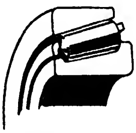


FIG. 20.

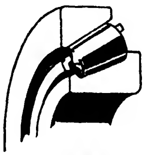
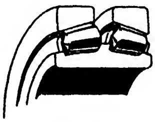


FIG. 21.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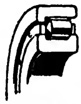
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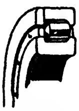
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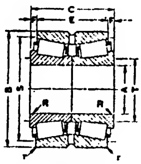


FIG. 25.

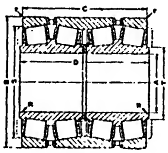


FIG. 26.

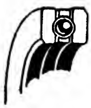


FIG. 27.

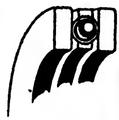
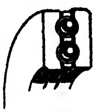


FIG. 28.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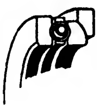


FIG. 30.

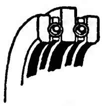


FIG. 31.

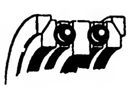


FIG. 32.

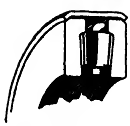
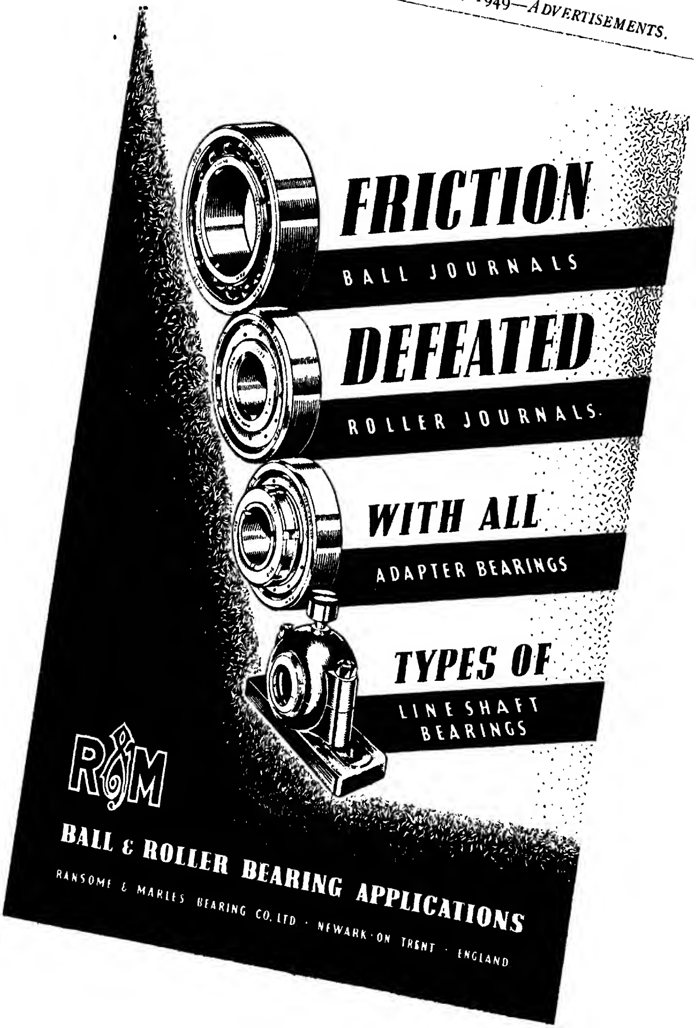
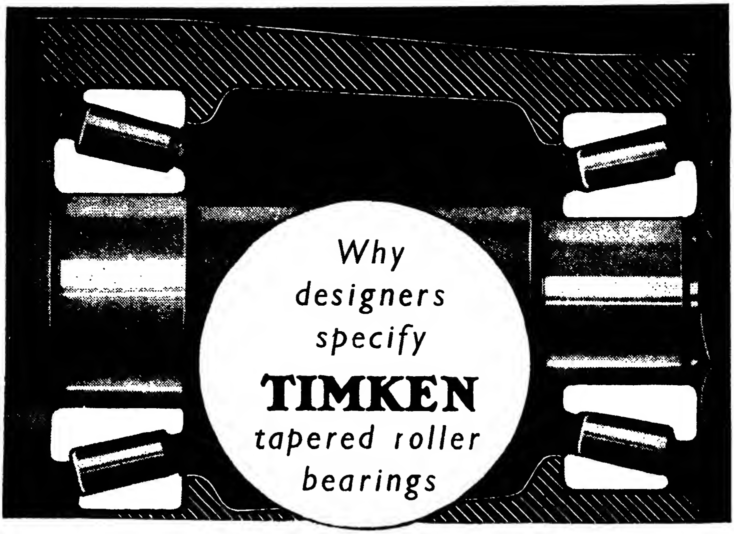


FIG. 33.


FRICTION
BALL JOURNALS
DEFEATED
ROLLER JOURNALS.
WITH ALL
ADAPTER BEARINGS
TYPES OF
LINE SHAFT BEARINGS
R&M
BALL & ROLLER BEARING APPLICATIONS
RANSOME & MARLES BEARING CO. LTD · NEWARK-ON TRENT · ENGLAND



Because <u>two</u> bearings do the whole trick!

There is no simpler way of meeting all the loads on a shaft or rotating part, than by mounting it on two Timken tapered roller bearings. Heavy radial loads, or heavy thrust loads, or a combination of both, are all taken care of by a pair of bearings, and even if the thrust load is unduly high, nevertheless two bearings will still suffice.

Furthermore, the bearings, having long line contact and a perfect rolling geometry, continue to give accurate location of the shaft for an extremely long life.

TIMKEN

tapered roller bearings

Regd.
Trade Mark:
Timken.

BRITISH TIMKEN LTD., Birmingham, and Duston, Northampton. Telephone : East 1321. Telegrams : 'Britimken, Birmingham.' Associated Company : Fischer Bearings Co. Ltd., Wolverhampton, makers of FBC ball and roller bearings and transmission equipment.

Needle roller bearings are another form of parallel roller bearing, and consist of rollers where the length exceeds the diameter by many times, *e.g.* rollers 2 mm. in diameter may have a length varying from 10 mm. to 40 mm. The rollers are crowded between the two tracks as shown at fig. 17, no separating cage being employed. Whilst these can be used for journal loads only, their sphere of utility is limited, and they cannot be regarded as superseding other types of roller bearings. Their most useful function is for applications where diameters are restricted, and speeds moderate, and for oscillatory motions.

Barrel-shaped rollers, when employed in bearings as shown at fig. 18, provide a roller bearing that is capable of withstanding heavy journal loads and thrust loads, whilst at the same time the spherical track and roller formation will permit of misalignment.

Another form of barrel-shaped roller bearing capable of correcting misalignment is illustrated at fig. 19.

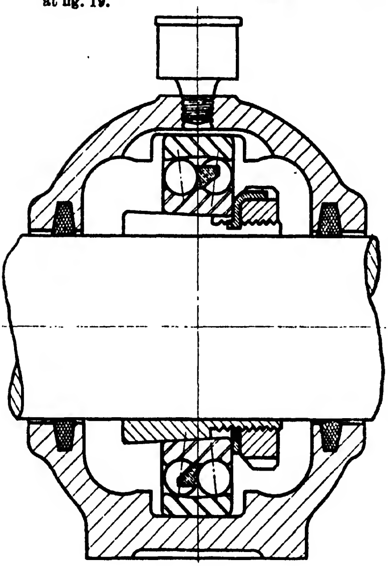


FIG. 34. C 18

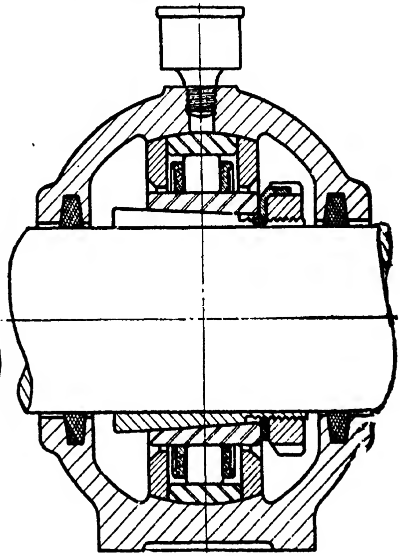


FIG. 35. C.16

Another well-known form of roller bearing is the tapered roller type which because of its full length line contact of rollers and races and true rolling properties is particularly well adapted to withstand combined radial and thrust loads. There are two common types, one, as in fig. 20, which has a normal angle for the rollers, and two, as at fig. 21, where the rollers run at a steeper angle. The latter is intended for duty where the thrust loads are considerably in excess of the journal load. Unless the thrust load is always of sufficient magnitude to keep the rollers and tracks in correct relation, these bearings are not mounted singly. Two bearings are usually employed, and such a combination will resist loads from any direction. Means of adjustment is provided in the application to ensure that the rollers are contacting with the tracks.

Taper roller bearings of the double-row type are also made, as shown at fig. 22, and other combinations of two and four-row bearings are also made, as shown in figs. 25 and 26.

Other forms of roller bearings are shown in figs. 23 and 24. The bearings are of the solid roller type, and in these, location or thrust loads can be resisted between the shoulders, or on the loose side plates provided in, or on, the inner and outer rings. The use of these bearings for heavy axial loading is to be deprecated, particularly at high speeds. Under thrust loads the rollers are rubbing on the lips, which is, of course, contrary to the principle of an anti-friction bearing; it being also appreciated that the efficiency of such a scheme is dependent upon adequate lubrication between the roller ends and the lips.

Fig. 37 shows the split type of roller bearing. The advantage of such a bearing is obvious, particularly on lineshafting, where pulleys and couplings have to be removed before the ordinary types of ball and roller bearings can be fitted. It is generally accepted that bearing efficiency is greatly dependent upon the retention of an unbroken ball or roller track, and therefore the use of split type roller bearings is limited to specific applications.

Thrust Bearings.

Ball thrust bearings usually consist of a single row of balls held in a cage which run in suitably formed tracks, see fig. 27. For light duties, however, it is permissible to use flat track washers as illustrated at fig. 28, whilst for heavy duty two rows of balls are frequently employed, as illustrated at fig. 29.

These bearings can be provided with a self-aligning feature consisting of spherically ground race rings, which fit in correspondingly ground washers, as shown at fig. 30.

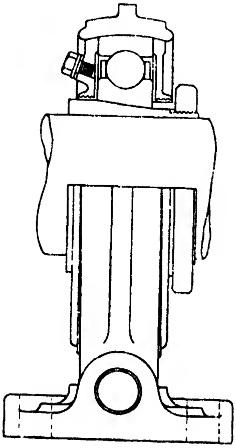


Fig. 36.

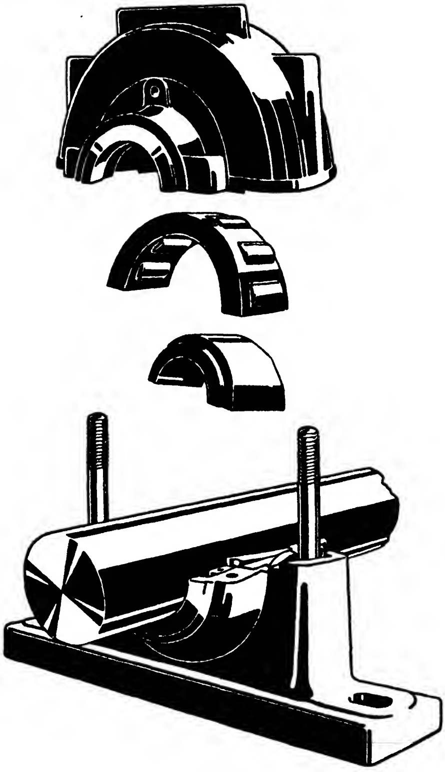


Fig. 37.

Where thrust bearings are required for thrust in alternate directions, then double thrust bearings, as shown at figs. 31 and 32, are essential, the latter showing one form of self-aligning double thrust bearing.

Taper roller bearings are also manufactured for thrust duties, and a section of one of these is shown at fig. 33.

Lineshaft Bearings.

Bearings for lineshafting can be of various designs, and are either of ball or roller construction. Examples of these are shown at figs. 34 to 37. These are usually mounted on a split screwed taper sleeve that can be secured to ordinary lineshafting, and are obtainable in the form of plummer blocks, hangers, side pedestals, etc.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TELEGRAMS "BALBEARING BIRMINGHAM" TELEPHONE ERDINGTON 1196 1197
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And at FERRYBRIDGE, YORKSHIRE.

Bearing Capacities.

The investigations carried out by Goodman, Stribeck, Hertz, and other pioneers, have formed the basis of later investigation and research carried out by various bearing manufacturers.

The original Goodman and Stribeck formulæ are only printed here in order that the user may compare the load-carrying capacities where the data required is available.

• The load ratings published by the various manufacturers varies widely, notwithstanding the fact that in each case the bearing may be fitted with the same number and size of balls and rollers.

Quality is fairly consistent amongst the principal makers, and obviously in actual fact similar bearings of competitive make have approximately the same load carrying capacity. The variations are due to the different makers' ratings being based on differing factors of safety and life. The only safe course, therefore, is to work to the load ratings published by the makers, and to adopt the life and safety factors that they advise.

Let P = the maximum permissible working load in pounds.
d = the diameter of the ball in inches.
D = the diameter of the ball path in inches.
n = the number of revolutions per minute of the shaft.
k = a constant.
c = a constant.
m = the number of balls in the bearing.
x = a co-efficient depending upon the revolutions per minute of the shaft.

Professor Stribeck's formula for radial ball bearings :—

$$P = 12 \cdot 8\, x,\, md^2$$

Where x has the following values :—

n	1	10	150	300	500	1,000	1,200
x	48·4	36·3	29·04	24·2	20·9	15·62	14·96
n	1,500	2,000	2,500	3,000	4,000	5,000	10.000
x	13·64	12·1	10·56	9·68	8·58	7·7	5·06

Professor Goodman's formula is :—

$$P = \frac{kmd^2}{nD + cd}$$

Where k and c have the following values :—

Type of Bearing.		k	c
Journal	Hollow races (best quality)	2,500,000	2,000
Thrust	Flat washers (best quality)	500,000	200
	Hollow races (best quality)	1,250,000	200

Methods of Mounting Journal Bearings.

The closest degree of accuracy is essential in the machining of shafts, housings and other components used in the mounting of ball and roller bearings. This is particularly the case where the working conditions are exacting, and especially where the speeds are high. Where alignment cannot be assured, as when two independent housings are employed, then self-aligning bearings or housings should be adopted.

Shaft seats on which the bearings fit, and the housing bores, should be machined to a good fine finish and should preferably be ground.

Abutment faces on shafts, or in housings against which the bearing rings are clamped, must be machined square with the shaft or housing bore. (Out of squareness results in stresses being imposed on the bearings, and is a frequent source of bearing failure.)

The diameter of the abutment faces should project well past the radius or chamfer on the bearing, as otherwise a tilting effect may be imparted to the bearing ring. The depth of face should be at least twice the bearing radius or chamfer.

The revolving race of a journal bearing should be an interference fit on its seat, the amount of interference varying with the duty to which the bearing is to be subjected.

Small lightly loaded bearings can be made a tight push fit, whereas with bearings subject to very heavy or shock loads, the interference may be as high as ·002 in.

It is advisable to consult the particular maker before adopting any tolerance for shafts or housings, but as a guide, Table I gives suitable tolerances in English sizes, which will ensure the correct fit of the bearing on the shaft, or in the housing under different conditions. English sizes

TABLE I.—TOLERANCES FOR SHAFTS AND HOUSINGS FOR BALL AND PARALLEL ROLLER BEARINGS.

Shaft Limits. T = Interference.
C = Clearance.

Nominal Shaft Diameter.	Limits in Bearing Bore.	Average Fit for Normal Duty. Shaft Rotating.		Average Fit for Normal Duty. Shaft Stationary.		Average Fit for Heavy Duty. Shaft Rotating.	
		Suggested Shaft Limits.	Fit obtained between Shaft and Bore.	Suggested Shaft Limits.	Fit obtained between Shaft and Bore.	Suggested Shaft Limits.	Fit obtained between Shaft and Bore.
In.	In.	In.	In.	In.	In.	In.	In.
Up to 3 .	+ ·0002 − ·0003	+ ·0000 + ·0005	·0002 C ·0008 T	− ·0005 − ·0000	·0007 C ·0003 T	+ ·0006 + ·0011	·0004 T ·0014 T
3 to 6 . .	+ ·0002 − ·0003	+ ·0002 + ·0007	·0000 C ·0010 T	− ·0007 − ·0002	·0009 C ·0001 T	+ ·0008 + ·0013	·0006 T ·0016 T
6 to 9 .	+ ·0002 − ·0005	+ ·0004 + ·0011	·0002 T ·0016 T	− ·0011 − ·0004	·0013 C ·0001 T	+ ·0010 + ·0017	·0008 T ·0022 T

Housing Limits. T = Interference.
O = Clearance.

Nominal Housing Bore.	Limit on Outside Diameter of Bearing.	Fit for Average Duty. Average Speeds, Solid Housings.		Fit for Light Duty. Low Speeds, Split Housings.	
		Housing Bore Limits.	Fit obtained between Bearing and Housing Bore.	Housing Bore Limits.	Fit obtained between Bearing and Housing Bore.
In.	In.	In.	In.	In.	In.
Under 2 . . .	− ·0003 − ·0008	+ ·0001 − ·0004	·0001 T ·0009 C	+ ·0008 − ·0004	·0001 T ·0016 C
2 to 3	− ·0005 − ·0010	− ·0001 − ·0006	·0001 T ·0009 C	+ ·0009 − ·0006	·0001 T ·0019 C
3 to 5	− ·0008 − ·0013	− ·0003 − ·0010	·0002 T ·0010 C	+ ·0008 − ·0010	·0002 T ·0021 C
5 to 7 . . .	− ·0013 − ·0018	− ·0006 − ·0013	·0000 T ·0012 C	+ ·0007 − ·0013	·0000 C ·0025 C
Above 7 . . .	− ·0013 − ·0018	− ·0003 − ·0013	·0000 T ·0015 C	+ ·0012 − ·0013	·0000 C ·0030 C

The above Housing Limits represent the usual conditions when the outer ring of the bearing is stationary. If the outer ring is the rotating member, the fit should be about ·0005 in. tighter.

The bearing limits shown are the standard limits used by most British manufacturers. For metric or other bearings having different limits, the shaft and housing limits must be modified to give the fit shown in the table.

only are shown, but the column which gives the fit obtained between the shaft and the bore or the housing, should be used as a guide for shaft or housing limits in those cases where the bearing tolerances vary from the figures given in the Table; this ruling being also applicable to metric size bearings.

In determining the amount of interference, care must be taken to ensure that the bearing is not initially loaded due to the expansion of the race rings.

The principal manufacturers segregate most types of journal bearings into three grades of internal fit, which are marked on the bearing faces in the form of one, two or three dots, or small circles, indicating the tightest, medium and slackest fits respectively. This internal fit relates to the clearance between the rolling elements and the bearing tracks, and it can range from nothing to several thousandths part of an inch. The advantage of this method of grading is that it enables the engineer to select a bearing fit that will be suitable for the particular conditions in service. For instance, where a heavy interference fit on a shaft is necessary due to the working conditions, the expansion of the bearing ring may initially load the bearing. The aim should be that the grade of 'fit of the bearing' selected should be such that there is the minimum degree of slackness when the bearing is assembled on the shaft and/or in the housing, due consideration being given to such conditions as variation in temperature, which may cause expansion of one ring but not of the other ring, such as is likely to occur on some electric motor applications, where the shaft is subjected to heat that can only be dissipated through the bearing. It is also preferable that the revolving ring should be secured against endwise movement by a nut or other similar locking device. When this is not adopted an increase in the interference fit is necessary.

The stationary ring, except when the bearing is required to accommodate thrust loads, should not be secured against lateral movement, and whilst being a good fit in the housing, it should be free enough to take up its correct position relative to the inner ring.

Where thrust loads are imposed, then the stationary ring as well as the revolving ring must be secured against lateral movement, one bearing only on each shaft being mounted in this manner, except in cases where two bearings are adjusted against one another, as occurs with angular contact bearings, or where two bearings are mounted to take thrust in opposite directions.

When it is impossible to use shoulders for abutments on the shaft, or applications where the duty is not excessive, such as lineshafting, the use of taper sleeves is permissible, and fig. 38 shows this design as made by the leading manufacturers.

The use of setscrews or similar devices to prevent the rotation of the race rings is to be deprecated, as they are likely to cause distortion, and unless extreme care is exercised, they will readily shear or fritter away.

Tapered Roller Bearings.

In general the revolving member should be press-fitted into place, and by reason of the tapered construction tighter interference fits can be safely used, as the internal clearance in the bearing is not affected, this latter being separately determined by the adjustment device.

Methods of Mounting Thrust Bearings.

When mounting thrust bearings it is essential to keep the supporting shoulders square with the axis of the shaft, and these should be of sufficient diameter to afford support to the ball path. The revolving and the stationary race should be made an easy fit to ensure that there is no possibility of causing eccentricity between the tracks in the two washers between which the balls are revolving.

Provision for Thrust or Location Duty.

Means must, of course, always be provided for locating the shaft and withstanding such thrust loads as may be present. This can be accomplished by means of a double thrust bearing, or alternatively, by two single thrust bearings, whilst journal bearings carry all the radial load.

In the average application, however, the ball or roller bearings carrying the journal loads have, in addition, to withstand the thrust load in both directions. The general practice is that one bearing only on each shaft or spindle should be so located, whilst the remaining bearings on the same shaft or spindle carry journal loads only. The reason for this is that any variation in the shaft length, which may be caused through the rise or fall in temperature, will not impose an artificial load on the bearings, such as would occur if two bearings were each mounted to resist thrust loads.

The common method of locating the bearing in the housing is by means of shoulders formed in the housing, or by the end caps, whilst on the shaft a locknut, which in turn is secured by some locking device, holds the bearing firmly against the shoulder on the shaft.

As an alternative, spring rings can be used for location or light thrust duties. At fig. 39, grooves are turned in the housings into which the spring rings fit, and the bearing is located between these rings. Fig. 40 shows a bearing that is now common with some manufacturers, in which the spring ring fits in a groove in the outer ring of the bearing. This held in the manner shown in the illustration, resists light thrust duties in one direction, the major thrust in the application shown being taken on the end cap of the housing. A spring ring is also shown for locating the inner ring on the shaft.

Where the combined journal and thrust loads are too heavy for one bearing to sustain, a single bearing of the ball journal type can be used for thrust duties only. The bearing is mounted so that it is relieved of all journal loads, by being clear of the housing on its outside diameter and may be the ordinary single-row ball journal bearing, or a duplex angular contact bearing as shown at fig. 41.

In order to keep the bearing clear of the housing, the bore can be made larger than the bearing diameter. Some manufacturers supply bearings that are undersize so that they may be fitted to housings that are bored to suit the journal bearing.

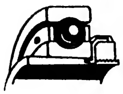


FIG. 38.

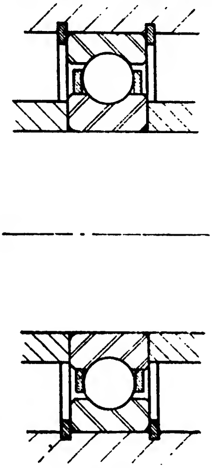


FIG. 39.

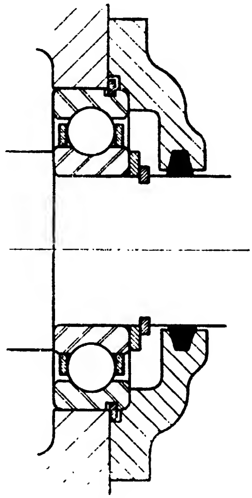


FIG. 40.

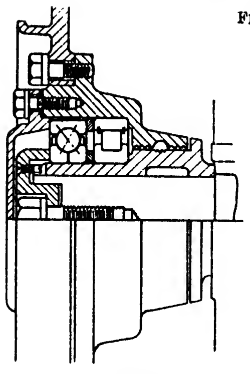


FIG. 41.

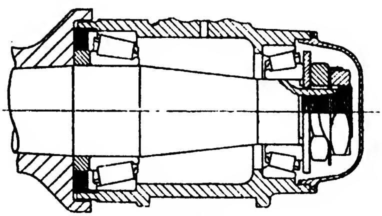


FIG. 42.

Exceptions to the principle of using one bearing only for locating the shaft occur when angular contact bearings or taper roller bearings are mounted to take thrust in opposite directions, in the manner illustrated in fig. 42. In these instances the overall dimensions between the abutting faces should be governed so that the bearings are not heavily initially loaded, or where adjustable abutments, such as packing shims or screwed abutments are used, similar care must be exercised to prevent overloading the bearings.

It is permissible in instances where two ball journal bearings are mounted on one shaft for each bearing to take thrust in alternate directions. Such an application is shown at fig. 43, and it is preferable, when this arrangement is adopted, and the application will permit of the feature, for clearance to be left between the face of the bearing and the locating face of approximately 0·010 in., rather than the bearing should be initially loaded.

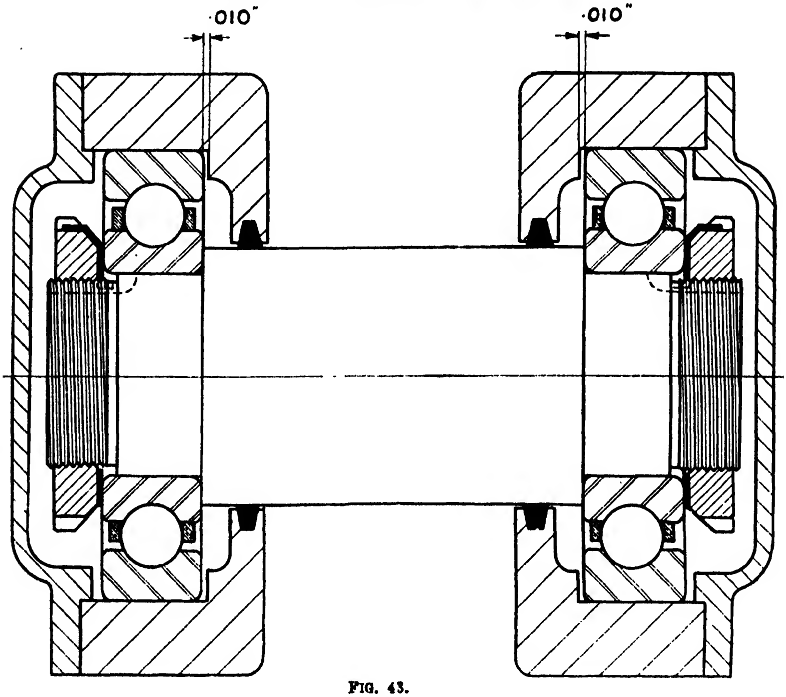


FIG. 43.

HOUSING DESIGN.

The arrangement of the housing should adequately protect the bearing from the possible intrusion of moisture, grit or abrasive particles, and at the same time be grease- or oil-tight, in accordance with the form of lubricant used.

The method of sealing must vary with the conditions under which the bearings will be working. Figs. 44 to 51 show several alternative sealing arrangements.

Felt washers are commonly used for average speeds and conditions, as shown at figs. 44 to 46.

Special seals of various designs, generally consisting of spring loaded leather or composite washers, can be fitted as an alternative to the felt washer. These can be used for protection purposes, or alternatively, as an oil or grease retaining device.

Labyrinth washers of various designs are very effective, and are a necessity for high speeds and oil lubrication; typical examples are shown at figs. 47 to 49. Fig. 51 illustrates a vertical mounting, and in such case, the housing arrangement must ensure that the lubricant cannot fall away from the bearing, leaving the latter dry.

Under wet conditions it is advisable to fit a flinger to the shaft as shown at fig. 48. This prevents percolation of moisture along the shaft.

A number of bearings are now also available fitted with grease retaining devices as integral parts of the bearing. The bearings are packed with grease when they leave the manufacturers' works, and receive no further lubricant during their life. Before using a bearing of this type it is essential to consider whether the type of seal will ensure protection, and whether the amount of lubricant present in the bearing is adequate to deal with the conditions in service, and give the anticipated life required from the bearing.

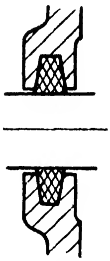


FIG. 44.

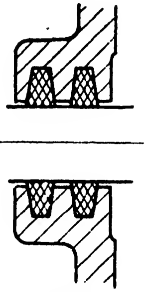


FIG. 45.

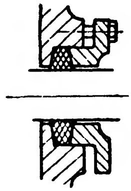


FIG. 46.

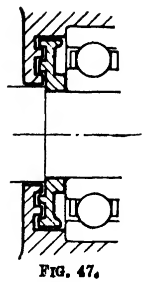


FIG. 47.

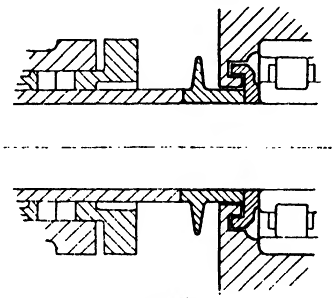


FIG. 48.

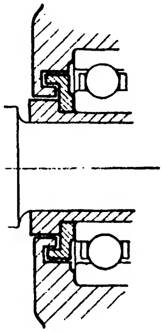


FIG. 49.

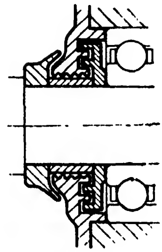


FIG. 50.

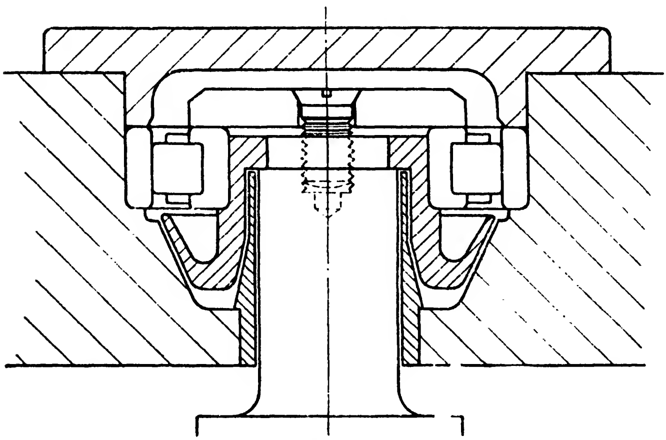


FIG. 51.

Fig. 52 shows a shielded bearing of this type, fitted with pressed side plates, the bores of which are a fine running clearance on the inner ring of the bearing. Fig. 53 shows a similar bearing in which felt washers held in side plates are a running fit on the shoulder of the inner ring.

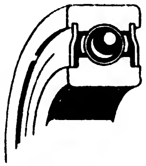


FIG. 52.

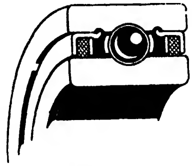
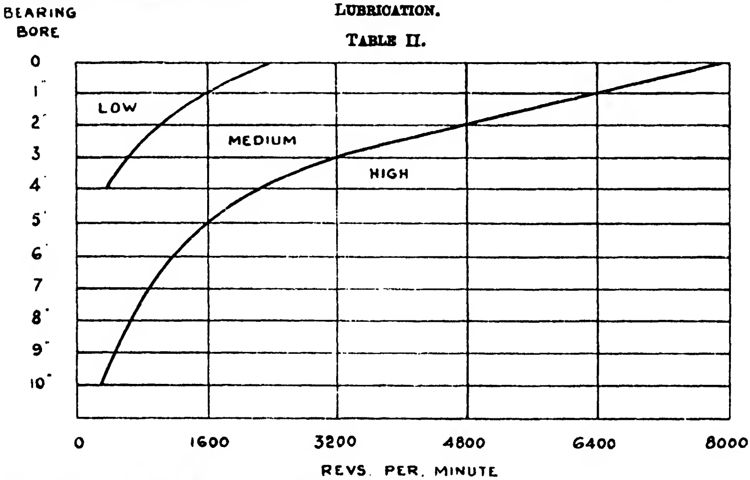


FIG. 53.

It should be noted that there are many variations obtainable in this design of bearing, all of which adopt a similar principle.

LUBRICATION.

TABLE II.



Temperature below 10° *F.*

Low .	. Low Temperature Mineral Oil.
Medium	. Low Temperature Mineral Oil.
High .	. Low Temperature High Speed Mineral Oil.

Temperature 10° *F. to* 32° *F.*

Low .	. Low Temperature Mineral Grease or Mineral Oil.
Medium	. Mineral Oil.
High .	. High Speed Mineral Oil.

Temperature 32° *F. to* 150° *F.*

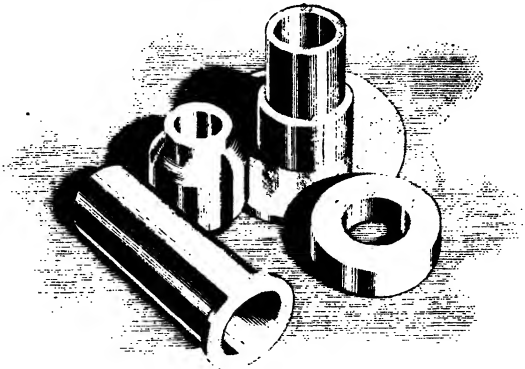
Low .	. Mineral Grease or Mineral Oil.
Medium	. High Speed High Grade Soda Base Grease.
High .	. High Speed Mineral Oil.

Temperature 150° *F. to* 280° *F.*

Low .	. High Melting Point Mineral Grease or Mineral Oil.
Medium	. High Speed High Melting Point Soda Base Grease or High Speed Mineral Oil.
High .	. High Speed Mineral Oil.

Temperature above 280° *F.*

Low .	. Mineral Oil.
Medium	. High Speed Mineral Oil.
High .	. High Speed Mineral Oil.



THESE SELF-LUBRICATING BEARINGS PROVIDE A CONSTANT OIL FILM

Compo bronze bearings start with a big advantage for they have a natural asset in their make up. They are die pressed from powdered metals and have a porous structure producing myriads of minute reservoirs which are charged with lubricating oil. This ensures a constant oil film on the shaft, increasing or decreasing according to shaft speed and bearing temperature.

COMPO

oil retaining

BEARINGS

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AND LEAD BRONZE
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OILITE
SELF-LUBRICATING
BRONZE BEARINGS
OILITE
Self Lubricating
BRONZE BEARINGS
—their
application
to
industry
THE MANGANESE BRONZE & BRASS CO. LTD
HANDFORD WORKS, IPSWICH TELEPHONE IPSWICH 2127 TELEGRAMS BRONZE IPSWICH

F. 863

Lubrication is required to reduce frictional resistance between the cage and the unit on which it is located, *i.e.* the race rings or the rolling elements; and to prevent corrosion.

Under high-speed conditions reduction of cage friction is most important, and the service obtained from either ball or roller bearings is entirely dependent on the correct lubrication.

Grease is the most convenient medium. This should have a mineral base, as animal or vegetable fats become rancid and develop free acid. The majority of mineral greases of either lime or soda soap are suitable for all average conditions. Soda base greases have a high melting point, and are more suitable for high speeds.

Many greases have a tendency to separate out at high speeds or high temperature, leaving a harmful deposit. Under such conditions it is essential that a suitable grade of grease is selected.

Housings should not be tightly packed with grease, especially when speeds are high, as the churning of the lubricant is liable to develop a high working temperature, particularly if the design of the housing does not permit of adequate radiation.

The renewal of lubricant depends entirely on the working conditions and the design of the housing. Recharging about every three months should be ample for normal conditions.

Oil lubrication involves the use of a better protection against the loss of lubricant in the housing than does grease. At the same time oil is more reliable than grease; it is essential for higher speeds, and a graph giving a rough indication of the conditions where oil is advisable is shown at Table II.

A good quality mineral oil is necessary, the specification depending upon the speed and general conditions. For temperatures from 250 to 300° F. and above, oil is necessary, in fact, the majority of greases fail to give service where the temperature exceeds 250° F.

The method of feeding the oil varies according to the working conditions. A drip or wick feed is ideal, or alternatively, a splash feed, in which the bearings are working in an oil mist.

An oil bath can be adopted, but at ultra high speeds this will generate a high temperature.

Where a wick or other oil circulating system is used, the feed should preferably be taken right through the bearing and a suitable drain provided.

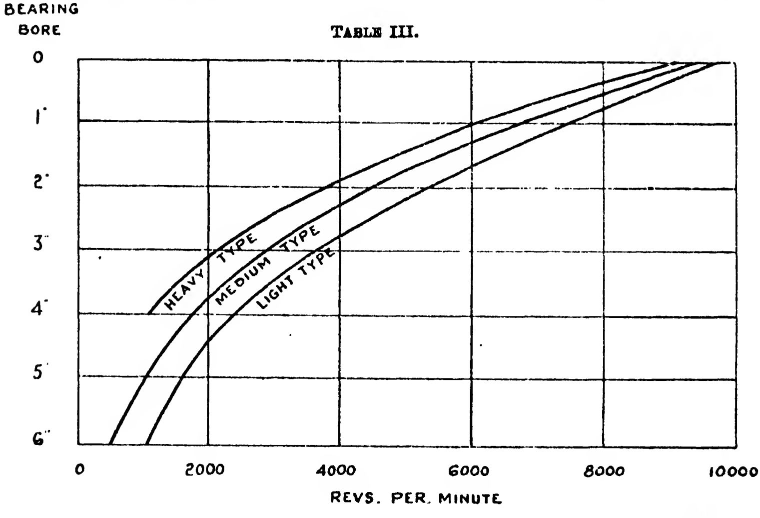
Graphited oils should be used with discretion, and then only when they are compounded from a pure colloidal graphite.

Bearings for High Speeds.

Under ultra high-speed conditions the whole bearing arrangement requires careful and special consideration; the type of bearing, the method of mounting and the lubrication are points of importance.

As a guide as to what may be considered ultra high speeds, Table III gives an indication. Experience is the only safe guide, and where such experience is lacking, the user should take the matter up with the manufacturer.

TABLE III.



Most manufacturers supply precision bearings for very high speeds. These are made to a closer limit of internal accuracy than standard bearings, and are sometimes fitted with special cages which are frequently made from bakelite or duralumin.

High Temperatures.

The limiting temperature for ball and roller bearings is approximately 300° F. This will vary upon the material used in the manufacture of the bearing, and the temperature at which this material is tempered. When the temperature approximates to 300° F., it is advisable to consult individual manufacturers regarding same.

Preloading Ball and Roller Bearings.

Preloading, *i.e.* applying an initial load on a bearing before the load from the application is imposed, is frequently employed on many applications, such as certain machine tools. This preloading gives rigidity to the shaft that is otherwise unobtainable, and in some instances it is adopted as a means of reducing noise.

Ball and roller bearings can in this case be supplied minus any clearance between the balls or rollers and the tracks, or alternatively, the preloading can be carried out by springs or other adjustable features on the application.

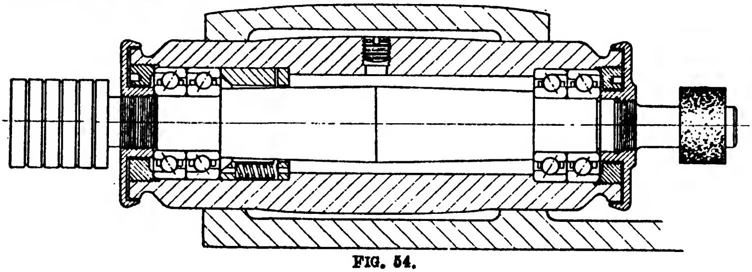


FIG. 54.

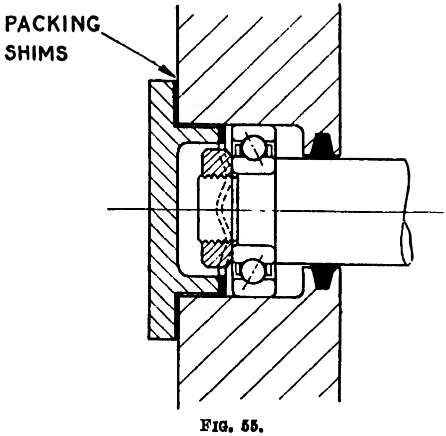


FIG. 55.

Fig. 55 shows an arrangement using a corrugated spring washer, whilst a number of small springs are used in the case of the grinding spindle illustrated at fig. 54. In both instances it will be seen that the springs exert a pressure which must be of sufficient magnitude to ensure that the full circle of balls are in constant contact with the tracks, and not only in contact at the heaviest loaded point.

Another method of achieving this object would be to use an adjustable abutment, although the difficulty arises here of determining the amount of preloading that is being exerted.

Fig. 57 A and B, shows two angular contact bearings mounted back to back, which are made so that when the inner and outer rings have been clamped up laterally, there is a degree of pre-

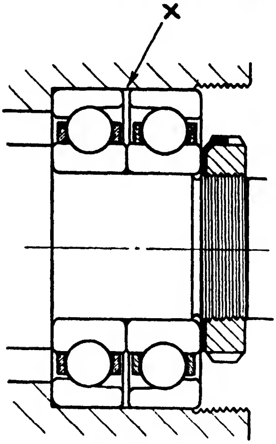


FIG. 56.

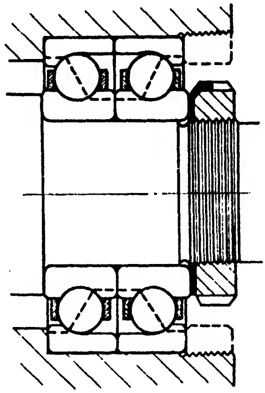


FIG. 57.

loading present in these bearings. At fig. 56 is shown the two bearings before being secured laterally. Note the clearance between the outer rings X; this space varying according to the

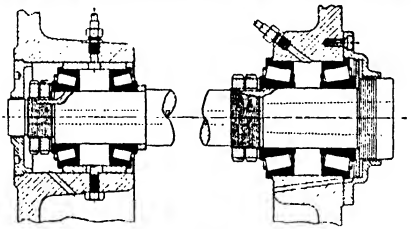


FIG. 58.

amount of preloading required. Bearings mounted in this manner are specially made and paired up with flush outer faces. After clamping the outer rings these abut each other, and the dotted line through the balls at fig. 57 shows how the preloading is obtained.

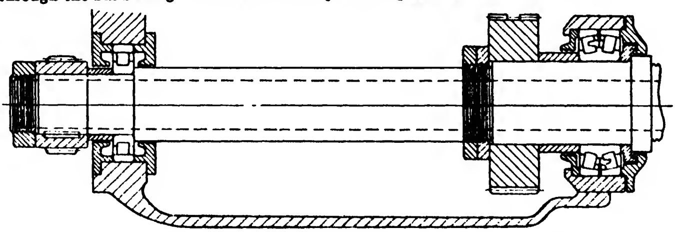


FIG. 59.

Figs. 58 and 59 illustrate respectively the application of a taper roller bearing and a barrel-shaped roller bearing, where the outer rings are made in two halves to allow for initial preloading in a similar manner to that graphically illustrated with the ball bearings at figs. 56 and 57.

BEARING PRESSURES AND DIMENSIONS.

(Contributed by H. W. Swift, D.Sc., M.I.Mech.E.)

Bearings generally may be divided into two classes: A. Bearings on which the load is heavy, the speed low or the rubbing surfaces so shaped that a lubricating film cannot be maintained. B. Bearings in which the rubbing speed is so high in relation to the pressure and the surfaces so shaped that a film of lubricant is maintained by the rubbing motion.

Class A.—These bearings operate under conditions of 'sparse' lubrication in which intermittent metallic contact occurs with resulting abrasion and heat dissipation. Such bearings should be designed to prevent seizure or mechanical failure of the metal and to limit the rise of temperature. For this purpose the pressure per unit of projected area should always be kept below the value at which there is danger of failure, and for higher rubbing speeds the pressure should be reduced in order to limit the intensity of heat dissipation. Limiting pressures for shaft bearings are shown approximately in Table IV.

TABLE IV.—PERMISSIBLE PRESSURE FOR SHAFT BEARINGS.

Journal.	Bearing.	Pressure. Lb./sq. in.
Mild steel	Cast iron	300
,, ,,	Bronze	500
,, ,,	White metal	600
Medium steel	Bronze	800
Hard steel	,,	1,200
,, ,,	Hard steel	2,000
Cast iron	White metal	200

On the assumption that the heat dissipation per unit area determines the temperature rise, and that the effective coefficient of friction is constant (with sparse lubrication) the general basis of design for journal and other rotary bearings will be: $p = \frac{K}{DN}$ where p is the pressure per square inch of projected area, D the journal diameter in inches, N the speed in r.p.m. and K a constant depending on the application and on the materials employed.

Representative values of K employed for various applications are shown in Table V.

TABLE V.—BEARINGS WITH SPARSE LUBRICATION.

Values of $K = pDN$.

Application.		K.
Crankpins:	Locomotive	1,250,000
	Marine engine	250,000
	Vertical engine	350,000
	Horizontal engine	150,000
	Small engines	60,000
Main bearings:	Horizontal engines	70,000
	Vertical engines	200,000
	Electric generators	50,000
Axles:	Locomotive	800,000
	Carriage and wagon	500,000
Pivots		300,000
Collar bearings		60,000

In bearings of this class the least axial width B is determined by the load W and permissible pressure p: $B = \frac{W}{pD}$. Although bearings of the types shown in the table are commonly designed on the basis of constant pDN which presupposes sparse lubrication, in many cases it is possible to attain conditions of film lubrication and to apply the methods for Class B bearings. A rough idea as to whether film conditions can be expected may be obtained from formulæ of American origin defining the pressure below which firm formation is reasonably certain :—

$$p < \frac{5}{2}\sqrt{DN} \text{ when } DN < 2{,}500$$

$$p < 10\sqrt[3]{DN} \text{ when } DN > 2{,}500$$

Class B.—In these bearings a pressure film is maintained and metallic contact does not occur except at starting and stopping. The friction is therefore much less than in Class A bearings and pressures may be higher for a prescribed temperature rise. Moreover, the permissible intensity of pressure does not vary inversely with the rubbing speed, but increases with it. The principles of film lubrication are sufficiently well understood to form a rational basis for design procedure.

Clearance Bearing Design for Film Lubrication.—The lowest frictional coefficient for a clearance bearing and the greatest film thickness are obtained when the clearance is so chosen in relation to the load and speed that the working eccentricity ratio is about 0·5.

For a given total load, speed and oil the frictional losses are least when the developed bearing surface is approximately square : for a half-bearing $\frac{B}{D} = \frac{3}{2}$, for a 120° bearing $\frac{B}{D} = 1$. Moreover the friction becomes less the smaller the permissible film thickness. But film thickness increases as the width B of the bearing is increased. Hence the optimum bearing is one with a 'square' bearing surface provided the corresponding film thickness is sufficient. If this thickness is insufficient a wider bearing is required.

The least permissible film thickness depends largely on the limits of accuracy with which a specified clearance can be produced. At low speeds the smallest clearance capable of being produced with a tolerance of $\pm$ 30 per cent. is desirable, but at high speeds a greater clearance is required to facilitate heat disposal. A reasonable clearance formula conforming generally to modern practice is :—

$$d_1 = \frac{\sqrt{D_1}}{1{,}000}\left(1 + \frac{\sqrt{N}}{20}\right) > \frac{3}{1{,}000}\sqrt{D_1}$$

where d_1 is the clearance in inches, D_1 the diameter in inches and N the speed in r.p.m.

If this clearance formula is accepted the procedure in bearing design is as follows :—

1. For the given load W lb. and speed N r.p.m. Compute the 'critical diameter' below which the square bearing gives inadequate film thickness :—

$$D_c = 1{\cdot}39\, W^{\frac{1}{3}}\left(\frac{1}{\sqrt{N}} + \frac{1}{20}\right)^{\frac{2}{3}}$$

2. If the specified diameter $D_1 > D_c$ then make—

   (*a*) the width $B = \frac{3}{2} D_1$ for 180° bearing.
   $B = D_1$ „ 120° „

3. If the specified diameter $D_1 < D_c$ then make—

   (*a*) the width $B = \frac{3}{2} D_c$ for 180° bearing.
   $B = D_c$ „ 120° „

   (*b*) the clearance :—

$$d_1 = \frac{\sqrt{D_1}}{1{,}000}\left(1 + \frac{\sqrt{N}}{20}\right) < \frac{3}{2{,}000}\sqrt{D_1}$$

Curves showing specific diameters and clearances will be found in the *Proc.I.Mech.E.*, 1935, pp. 421, 424. For convenience of designers alignment charts drawn up in accordance with the above procedure are shown in figs. 60 and 61. From fig. 60 the critical journal diameter D_c for a given load and speed is found by drawing a single line as explained on the figure. If this diameter is employed the corresponding clearance is obtained by another line on the same figure. If the chosen diameter $D_1 > D_c$ then the appropriate clearance is found by drawing two lines on fig. 61.

The procedure outlined above is suitable for bearings with constant load. Cyclic variations of load on a bearing increase its capacity as based on the maximum load to an extent which may be judged from the tables of bearing pressures given in Table IV.

The ratio of bearing width to diameter $\frac{B}{D}$ is usually made greater at high speeds to increase the area available for heat disposal. On the other hand the modern tendency is to reduce the value of $\frac{B}{D}$ for bearings of class B especially where the journal is subject to considerable bending action. In aeroplane engines values of $\frac{B}{D}$ as low as $\frac{1}{2}$ have been employed.

Representative values of bearing pressures and proportions for various applications are given in Table VI.

TABLE VI.

Main Bearings. p	Main Bearings. $\frac{B}{D}$	Crank Pins. p	Crank Pins. $\frac{B}{D}$	Application.
400	1¼	600	1¼	Marine engine
250	2	1,500	1	Locomotive engine
300	2	1,000	1	Horizontal engine
250	2½	500	1	Vertical engine
500	1½	1,500	1	Oil engine
200	2	400	1	Compressors
300	1½	1,200	1¼	Gas and petrol engine
50	2½	—	—	Steam turbines
50	2½	—	—	Electrical machines
250	2	—	—	Shafting, heavy, brass or babbit
100	2	—	—	Shafting, medium
50	2½	—	—	Shafting, light, C.I. steps
2,500	3	—	—	Presses and rolling-mills
200	1½	—	—	Machine tools.
250	1–1½	—	—	Locomotive axle
350	2	—	—	Railway, carriage, wagon, tender

Crosshead pins p = 1,000–1,500 lb./sq. in.*

$\frac{B}{D}$ = 1¼

Crosshead slides p = 40 lb./sq. in.

Eccentric straps p = 80 lb./sq. in.

The following particulars of representative modern bearings have been collected by H. L. Haslegrave (*Proc. I.Mech.E.*, 1935, pp. 456–473):—

TABLE VII.

Machine.	Bearing.	p Lb./in.^{2}	Rubbing Speed. Ft./sec.	Clearance $1{,}000\frac{d}{D}$	Bearing Angle.	$\frac{B}{D}$
Steam turbine . .	Main . .	150	170	¾–3	120	1–1½
„ engine . .	„ . .	300	11	1	180	1–2
„ „ . .	Crankpin . .	550	10	1	360	1–2
„ „ . .	Crosshead .	1,200	0·5	1	360	1–2
Electrical machines .	Industrial .	130	25	1–3	360	1¼–2½
—	Induction motors	130	25	½–1	360 (relieved)	2½
—	Turbo-driven .	130	75	1–3	360 (relieved)	1
Machine tools . .	General . .	200	9–18	¾–2	100	1½–2
—	Lathe spindles .	150	10	1–¾	360	¾–1¼
I.C. engines . .	Main . .	500	12	1–1½	140	1–1½
Oil . . .	Crankpin . .	1,400	12	1–1½	360	1
Gas and Petrol .	Main . .	300	15	1–1½	110–140	1
—	Crankpin . .	1,200	12	1–1½	360	1
Gears . . .	—	120–200	4–67	¾–2	140	1½–2½

* In American locomotive practice double this pressure is employed.

Supply of Lubricant.—In bearings of Class B it is necessary to provide an adequate supply of oil to replenish the pressure film and to allow for heat disposal. This oil is distributed by axial

CRITICAL DIAMETER AND CLEARANCE.

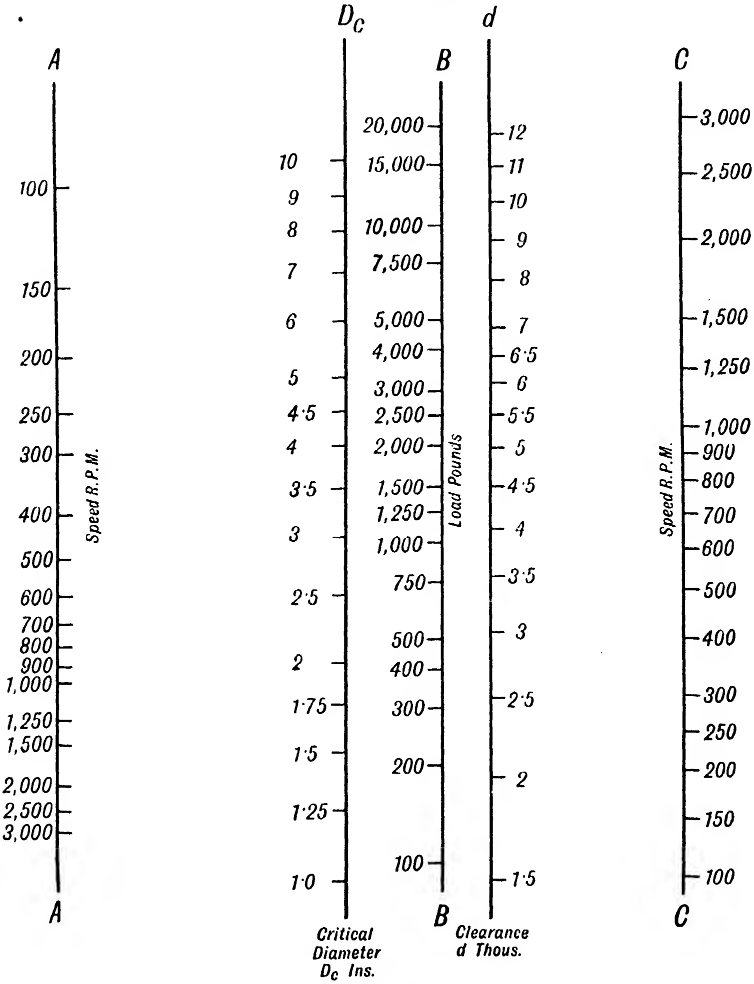


FIG. 60.

Join *Speed* on A to *Load* on B, cutting D_c in *Critical Diameter*.
Join *Speed* on C to *Critical Diameter* D_c, cutting d in *Clearance*.

groove along the bearing, but no grooves should be allowed nearer than about 60° to the loaded region of the bearing. In bearings of Class A an adequate supply of oil is required at higher

speeds for purposes of heat disposal, but grease is frequently more effective lubricant at low speeds and high pressures. Since the circulation of the lubricant by traction is not assured and

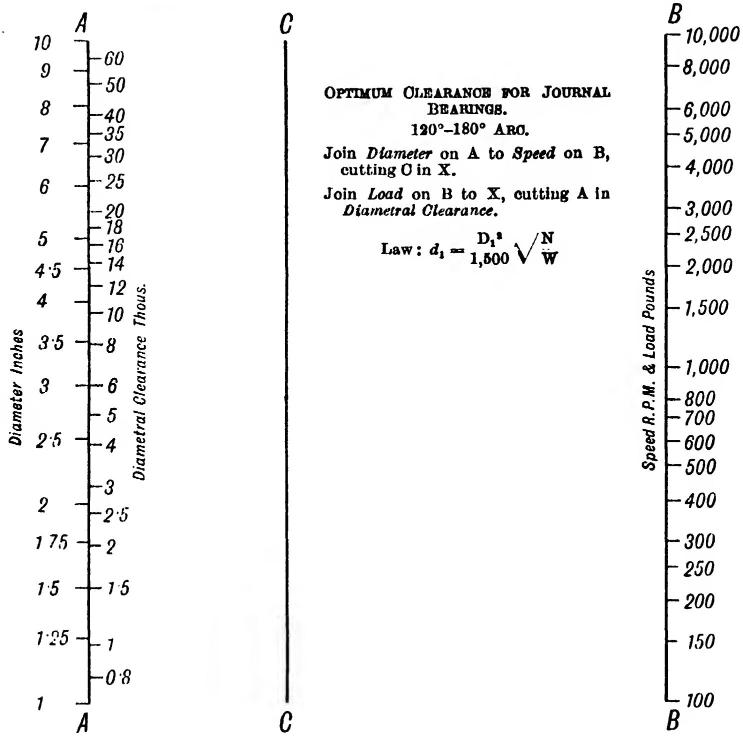


FIG. 61.

no true pressure film is attainable, the lubricant may properly be conveyed by grooves quite close to the region of load.

Of the ordinary automatic oil-circulating devices the collar is by far the most profuse in its supply. The oil-ring is sensitive to journal diameter and speed ; its efficiency is improved if grooves are turned in its inner surface.

See also Descriptive Section XX, Part I.

Bound Brook Bearings (G.B.), Ltd.
British Timken, Ltd.
Hoffmann Manufacturing Co., Ltd.
Manganese Bronze & Brass Co., Ltd.
Ransome & Marles Bearing Co., Ltd.
Skefko Ball Bearing Co., Ltd.

SECTION XX

PART II

LUBRICATING OILS AND LUBRICATION.

LUBRICATING OILS.

(Contributed by Harold Moore, M.Sc.Tech., F.Inst. Pet., F.C.S., A.I.M.E., and E. Stokoe, M.Inst. Pet.)

At the present time most machinery is lubricated either with mineral oils or with mixtures of mineral oils with a small proportion of either animal or vegetable oil. The mineral oils used are almost entirely derived from petroleum, but small quantities of low viscosity oils, as for example spindle oils, are prepared from oil shale, while certain tar and lignite products are used for the manufacture of greases.

Apart from the small proportion of lubricants produced from shale oil, tar and lignite, lubricating oils may be divided into three classes, (1) straight run or unblended mineral oils which may be distillates or residues obtained by the distillation of petroleum, (2) blended mineral oils which are mineral lubricants prepared by the mixture of two or more different varieties of mineral oil, and (3) compounded oils which consist of mineral oils ' compounded ' or blended with animal or vegetable oils. It may be noted that the term ' blending ' is usually employed to denote the mixture of two or more mineral lubricating oils, while the term ' compounding ' implies the mixing of mineral oils with oils of animal or vegetable origin. Reference should also be made to ' doped ' lubricating oils, which may be oils of any of the types previously mentioned to which have been added special products or compounds in order to improve certain characteristics.

A wide range of lubricants may be prepared from crude petroleum, and a brief summary of the various types of petroleum lubricating oils is given below. Reference is also made to commercial terms commonly used to describe certain grades.

As previously mentioned, lubricants are prepared from suitable grades of crude petroleum by fractional distillation, and formerly oils of low and medium viscosity were usually recovered as distillates, while oils of high viscosity were prepared by distilling off a large portion of the crude petroleum, the residue being utilised as a lubricating oil. In recent years, however, largely as a result of improved technique in distillation under reduced pressure, many oils of high viscosity are now obtained as distillates, although they are still frequently described by the terms formerly used to describe residual lubricants.

The petroleum distillates used for lubrication are generally refined either by treatment with sulphuric acid or by filtration through Fuller's Earth, bauxite or clay, frequently both processes are used. Oils of high viscosity, which were formerly obtained as petroleum residues after removal of the lighter oils, are also treated in a similar manner. Many such oils, however, contain paraffin wax and as the refining treatment previously mentioned has little or no effect on the wax content, additional treatment was found advisable in order to improve the setting point of wax containing lubricants. Earlier processes consisted in diluting the oil with petroleum spirit, the wax being then precipitated from the solution by cooling. After removal of the wax in

PROPERTIES OF TYPICAL LUBRICATING OILS.

Description.	Specific Gravity at 15° C.	Closed Flash Point ° F.	Viscosity Redwood in Seconds. 70° F.	100° F.	140° F.	200° F.	Coke per cent. Air Board Specification.	Sludge Test BESA Specification.	Acidity 100 gr. Oil Neutralises Mgr. KOH.	Sulphur Content by Bomb Method. Per Cent.	Oxidation Test (Air Board) Specification after blowing for 12 hrs. at 200° C. Viscosity Redwood. 100° F.	Viscosity Redwood. 200° F.	Coke Air Board Specification.	Acidity 100 grs. Oil Neutralises KOH.	Viscosity Redwood in seconds oil-heated for 45 hrs. at 150° C. without air passing through 70° F.	100° F.	140° F.	200° F.	Proportional values before and after oxidation. Coke per Cent. After/Before.	Viscosity Redwood at 100° F. After/Before.	Viscosity Redwood at 200° F. After/Before.
Russian Pale Oil .	·909	384	1180	387	132	56	·19	0·72	6·0	0·17	880	70	1·09	140·0	1310	412	141	56	5·7	2·3	1·25
Californian Red .	·928	354	1235	398	120	52	·21	1·13	18·0	1·32	2660	158	4·69	212·0	—	—	—	—	22·3	6·7	3·0
Californian Pale .	·928	355	1194	336	125	53	·15	2·15	18·0	0·66	2370	128	4·28	220·0	—	—	—	—	28·5	6·1	2·4
South American Red	·930	348	1130	354	123	52	·87	1·78	12·0	1·76	Very viscous.	429	8·01	140·0	—	—	—	—	9·2	Probably over 20.	8·25
Mixed base Red Oil (considered to be mixture of Mid. Continent and Penna base).	·910	425	1314	434	150	59	·51	0·82	15·0	0·58	1010	95	3·50	140·0	—	—	—	—	6·9	2·3	1·6
Pa. Long Residuum	·879	427	1595	612	211	74	·55	0·35	9·0	0·10	900	112	2·94	140·0	—	—	—	—	5·4	1·5	1·5
Russian Cylinder Oil	·912	417	2610	760	220	72	·27	0·59	6·0	0·15	1210	88	1·55	110·0	2870	806	231	73	5·7	1·6	1·2
Texas Red Oil .	·939	355	1440	418	127	53	·46	1·24	28·1	0·58	1320	99	2·68	234·5	—	—	—	—	5·8	3·2	1·9

suitable centrifuges the petroleum spirit was removed leaving a residual lubricating oil of improved setting point. Oils treated by these and similar processes were known as ' bright stocks' as distinct from ' filtered cylinder oils ' which were refined petroleum residues which had not been subjected to the dewaxing process. In more recent years various dewaxing processes have been developed and owing to the general adoption of such processes high quality oils of low cold test are more common than was the case some years ago.

It should further be noted that treatment with sulphuric acid and Fuller's Earth, clay or other adsorbent medium, does not greatly change the general characteristics of the finished lubricants, which formerly were mainly dependent upon the nature of the crude oil from which they had been produced. In recent years, however, various solvent refining processes have been developed, by means of which it is possible to prepare oils of Pennsylvanian type from mid-Continent or similar medium quality crude oils. These processes perform what is in effect the separation of the high grade constituents from those of lower grade, the yield rather than the quality of the finished lubricant being determined by the grade of crude oil from which they are produced.

Distillate oils of medium viscosity are frequently described by reference to their Saybolt viscosity at 100° F., qualified in many cases by reference to their colour, 300 Pale, 750 Red and so on, referring to oils having these Saybolt viscosities at 100° F. Oils of low viscosity, however, are frequently classified according to their A.P.I. or specific gravity, 36 Pale Neutral and 875 Pale, for example, indicating lubricating oils having these A.P.I. and specific gravities respectively. The term Neutral indicated an oil which had been refined by treatment with Fuller's Earth or similar adsorbent medium without having been treated with sulphuric acid.

' Dark steam refined cylinder oils ' were petroleum residues of high flash point which had not received refining treatment other than a possible treatment to reduce their hard asphalt content. Such oils are frequently classified in accordance with their open flash points or fire points, common grades being 600° F. Fire, 630° F., 600° F. Flash and 640° F. Flash. These and similar oils are distinguished from ' filtered cylinder oils,' which are oils which have been treated with refining earth.

Oils used for the lubrication of machinery may consist of unblended oils, of mixtures of two or more oils of similar or varying type, or of mineral lubricating oils compounded with fatty oils of animal or vegetable origin.

Laboratory Tests for Lubricating Oils.

Various laboratory tests have been devised in order to examine certain properties of lubricating oils. The methods usually employed in Great Britain are described in ' Standard Methods for testing Petroleum and its Products,' published by the Institute of Petroleum; standard methods are also described in certain B.S.I. Specifications for petroleum products. It should nevertheless be noted that the majority of the tests commonly employed in the petroleum industry give only an indirect indication of the suitability of a given oil for any particular purpose and considerable experience is therefore necessary in order to interpret the results of laboratory tests, such interpretation being based to a considerable extent on a knowledge of the properties of oils which have previously been found suitable for the particular type of machinery under consideration.

These reservations being borne in mind the significance of some of the tests commonly employed is indicated below :—

The specific gravity of a lubricating oil has little, if any, direct bearing on the lubricating properties, but is nevertheless useful as giving an indication of the base of the oil.

The open flash point is determined by heating the oil in an open cup at a uniform rate, the temperature being observed, whilst a small flame is periodically brought near the surface of the oil, the open flash point being that temperature at which a small sheet of flame or flash flickers over the surface of the oil and goes out almost immediately. If the oil is heated to a higher temperature after the observation of the open flash point, a temperature will eventually be reached at which sufficient inflammable vapour is generated to maintain a flame on the surface of the oil for a few seconds. This temperature is known as the burning point or fire test.

Special instruments have been devised for the determination of flash points in which the oil is enclosed in a small cylindrical vessel so that the vapour is shielded from the effects of air currents. Results obtained by the use of such instruments are called closed flash points. The instrument specified by the Institute of Petroleum and most frequently employed in Great Britain and many European countries is the Pensky Marten apparatus, in France the Luchaire instrument is commonly employed.

The closed flash point test is of importance as a means for detecting whether lubricating oils have become contaminated with inflammable products of low boiling point. While it cannot be stated that flash point alone is an accurate guide to probable evaporation losses in service, it may nevertheless be stated that high closed and open flash points are desirable in oils used for the lubrication of steam engine cylinders, particularly when temperatures are high. It is usually considered that the employment of low flash point oil for this service is liable to cause high evaporation losses and to increase oil consumption.

The property of viscosity is the resistance to motion or in common terms the thickness of the oil. As the viscosity of lubricating oils falls with increasing temperature, the temperature at which the test was performed should be stated when giving the results of a viscosity determination. For convenience of comparison it is usual to determine viscosities at certain fixed temperatures commonly employed in the trade in various countries. In Great Britain viscosities are usually determined at 70° F., 100° F., 140° F., 200° F., and 250° F. In the United States the temperatures 100° F., 130° F., 210° F., and 212° F., are usual, and on the Continent viscosities are frequently determined at 20° C., 50° C., and 100° C. The temperatures chosen for determining the viscosity of any given oil naturally would not necessarily include all the temperatures given in the three ranges mentioned. When testing extremely viscous oils, for example, it is frequently not convenient to determine viscosities at the lower temperatures, similarly, extremely fluid oils are occasionally tested at the lower temperatures only. Instruments for measuring viscosity may be divided into two classes, (*a*) the 'arbitrary standard' type, and (*b*) the 'absolute' type. Examples of the first type are the Redwood viscometers commonly used in the United Kingdom, the Saybolt instruments used in the United States of America, both giving readings in seconds, and the Engler instrument giving results in degrees Engler commonly employed on the European Continent. A further example is the Barbey instrument, occasionally employed in France, and measuring 'fluidity,' *i.e.* increasing results on the Barbey instrument indicate decreasing viscosity and vice versa.

When an oil of a given viscosity is tested at the same temperature by the various 'arbitrary' instruments mentioned above, results are obtained in various different units which do not bear a simple relationship to the true viscosity of the product. In recent years there has been a growing tendency to express viscosity in Absolute units employing, for example, the C.G.S. system. Units in this system have been named the poise, defined in dynes per square centimetre per second, and the centipoise, being one-hundredth part of the poise. Whilst Absolute units have not as yet been universally adopted for commercial purposes, 'Arbitrary readings' are nevertheless being steadily displaced by Absolute units, particularly in specifications. When Absolute viscosity is measured by means of an efflux viscometer, or other apparatus wherein the oil falls through a calibrated tube or orific by gravity, it is necessary to take into account the specific gravity or density of the liquid at the temperature of measurement, and an alternative system of measurement has been suggested wherein this correction is omitted. Results obtained in this manner are termed Kinematic viscosities, the units of measurement being the stoke and centistoke, oils having the same Redwood, Saybolt, or Engler viscosity possess the same Kinematic viscosity in centistokes, the Absolute viscosity in centipoises, however, will vary if the specific gravities of the oils differ, the relationship between Kinematic and Absolute viscosities is given below:—

$$\text{Kinematic viscosity (centistokes) at } t^\circ = \frac{\text{Absolute viscosity (centipoises) at } t^\circ}{\text{Density at } t^\circ}$$

Viscosities in Redwood seconds may be converted to Absolute viscosities in C.G.S. units by means of the following formula:—

$\text{Viscosity in poises} = 0{\cdot}00353 \left(3{\cdot}6 \text{ Red} - \frac{\delta^2}{\text{Red.}}\right)$, approximately, where 'Red' is the Redwood value, and δ is the density.

On pp. 875 and 876 graphs are given showing the viscosity temperature relationships of various oils. The viscosity scales apply to both sides of the charts and they can thus be employed for the inter-conversion of viscosities in various commercial units and also for their conversion into Kinematic viscosity in centistokes. When it is desired to determine the Absolute viscosity the Kinematic viscosity should be multiplied by the density of the oil at the temperatures at which the viscosity is measured.

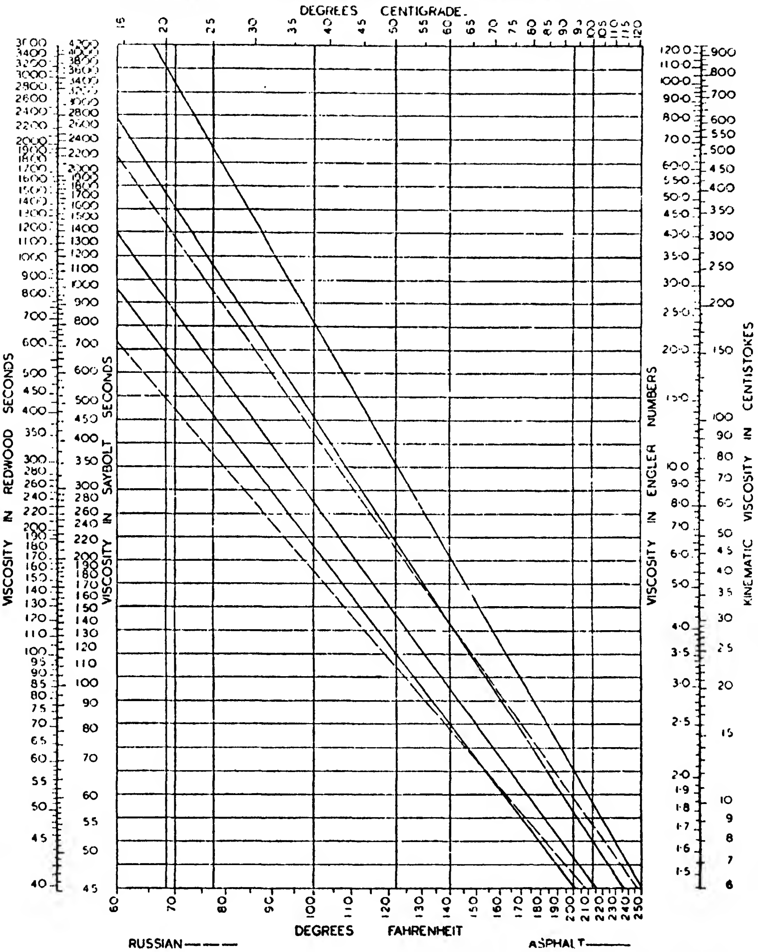
Examination of the graphs previously mentioned shows that mineral oils possessing similar viscosities at any particular temperature may not possess similar viscosities at a higher or at a lower temperature, the rate of fall in viscosity with increase in temperature being dependent upon the chemical nature of the base oil. In general, oils of paraffin base possess flatter viscosity temperature curves than do oils of naphthenic or asphaltic base. The inclination of the temperature viscosity curve is of importance when comparing oils intended for many purposes.

The viscosity of the oil to be used for any particular purpose is largely decided by its viscosity at the normal temperature of operation of the machinery for which it is intended. If this temperature is appreciably higher than normal atmospheric temperatures, it will be noted that the viscosity of the oil, when the machinery is cold, will depend on the viscosity temperature curve of the oil employed. Comparing, therefore, two oils having the same suitable viscosity at working temperature, it will be found that the oil possessing the steeper viscosity temperature curve will be more viscous at atmospheric temperatures and there is thus increased resistance when the machinery is started from cold. This may result in difficult starting and in increased loss of power while the machinery is warming up. Similarly, if for any reason the temperature of the machinery should appreciably exceed the normal operating temperature, the former oil will lose its viscosity to greater extent than the latter, so that oils possessing flat viscosity curves are to be preferred.

In view of the importance of viscosity temperature characteristics, various formulæ have been suggested by means of which it is possible to express the slope of the curve in terms of an arbitrary scale. A scale commonly employed in the industry gives results in terms of viscosity

VISCOSITY-TEMPERATURE CHART.

ASPHALT BASE OILS (CALIFORNIAN AND TEXAS) AND RUSSIAN OILS.



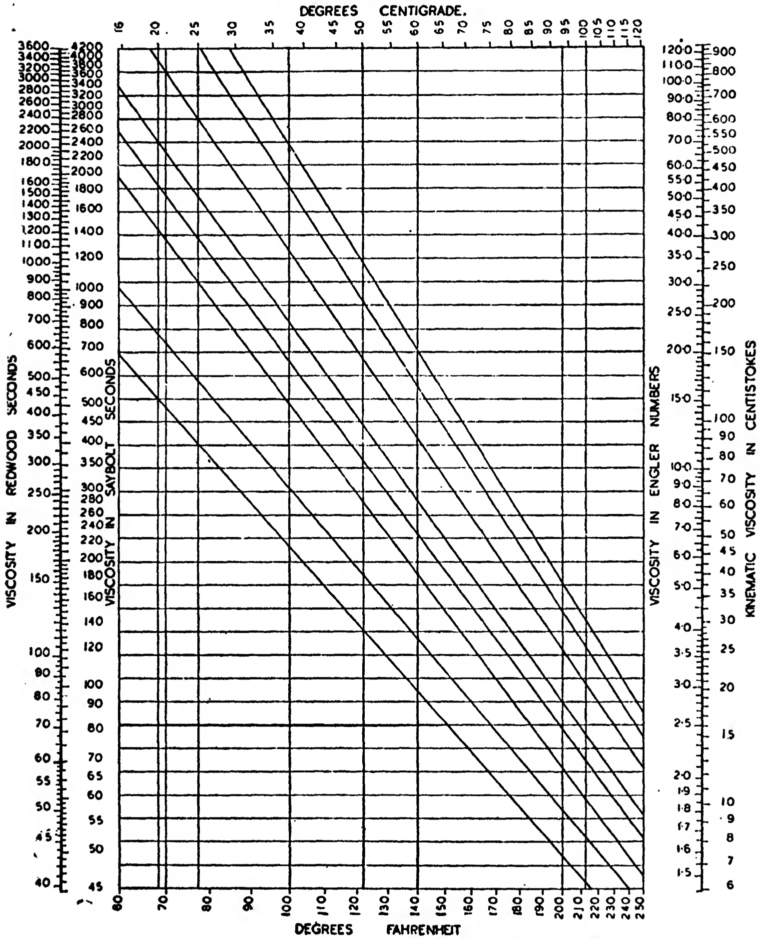
(*'Petroleum Charts and Tables,'* *H. Moore.*)

index. This index is an arbitrary figure derived from a formula based on viscosity at 100° F. and 210° F. The scale is so arranged that oils of Pennsylvanian type possess viscosity indices of approximately 100, while high gravity oils of the Californian type give viscosity indices of approximately 0. Owing to the introduction of various solvent refining processes, however, it is possible to produce oils having viscosity indices considerably superior to 100, while oils of which the viscosity indices are less than 0 are also known.

In addition to the determination of specific gravity, flash or fire point and viscosity, various other tests may be used for the examination of lubricating oils and among these may be mentioned setting point, coke test, demulsibility, saponification value and acidity.

VISCOSITY-TEMPERATURE CHART.

PENNSYLVANIAN OILS.



(*'Petroleum Charts and Tables,' H. Moore.*)

It is desirable that the setting point of a lubricating oil should be lower than the lowest temperatures to which it is likely that the oil will be subjected in service.

Various coke tests have been devised and are frequently taken as being an indication of the tendency to form carbon deposits in internal combustion and other engines. It should be noted, however, that the coke test of an unused lubricating oil is not necessarily a true indication of carbon-forming tendency, the relationship is by no means straightforward and there is a certain diversity of opinion regarding the exact significance of the coke test. For example, many distillate oils of 'asphaltic' origin have extremely low coke tests, but the resistance of such oils to the effects of oxidation is usually inferior to that of high grade paraffin base lubricants which may possess higher coke tests in the unused state. When, therefore, lubricating oils remain in the circuit and are exposed to oxidising conditions the coke test alone cannot be taken as being a true and exact indication of carbon-forming tendency. When, however, conditions are such that the oil may be burnt without being exposed to oxidising conditions for any appreciable period, then the coke test is of greater importance. It may nevertheless be stated, however, that for oils of similar type and base a low coke test is to be desired.

Various demulsibility tests have been devised in order to determine the readiness with which lubricating oils will separate from water, and such tests are of importance, for example, when examining oils required for the lubrication of steam turbines.

While it is not usual to determine the saponification value of pure mineral oils, the test is of interest for the examination of lubricants the origin of which is unknown, as the results give an indication of the presence of animal or vegetable oil in compounded lubricants and if the nature of the fatty oil is known it is possible to estimate the percentage of fatty oil present from a consideration of the saponification value.

Acidity tests may be performed in order to ascertain whether the oils are free from mineral acids and whether they contain excessive quantities of organic acids.

The above-mentioned list is by no means exhaustive and various other tests may be applied according to circumstances. For example, the determination of hard asphalt, ash, water or mechanical impurities may be necessary.

It will be noted that when any or all of the above tests are applied to unused lubricants they give only an indirect indication of their stability under working conditions and tests have therefore been devised in an effort to measure the stability of lubricants by means of laboratory tests. Such tests include the B.S.I. sludge test, which was primarily devised for the examination of electrical transformer oils, and the Air Ministry oxidation test. The B.S.I. sludge test has been included in the B.S.I. specification for turbine oils and certain users employ the test as a measure of the stability of lubricants. There is, however, a certain diversity of opinion regarding the utility of the test for this purpose.

The Air Ministry oxidation test consists in passing air at a fixed rate through a given quantity of oil under specified conditions. The increases in Ramsbottom Coke and Absolute viscosity are measured and these increases are taken as being an indication of stability in internal combustion engines. While it must be admitted that a laboratory test of this nature fails to duplicate the diverse conditions which may exist in practice, nevertheless the results obtained may be considered as being of assistance in default of prolonged engine trials, the performance and interpretation of which are by no means simple.

Notes on Lubricating Oils.

In addition to the various properties mentioned above, considerable research has been conducted in recent years on the subject of 'oiliness.' This is undoubtedly a surface tension effect, and is the property of maintaining a film when subjected to pressure between two plane surfaces. This is a property which cannot be readily measured, but its influence is well known. Certain of the saponifiable oils, that is to say, animal and vegetable oils, are much more satisfactory as regards this property than are mineral oils, and it is frequently observed that when a saponifiable oil or a compounded oil is employed for any particular purpose it is possible to use a product of lower viscosity than if a straight mineral oil were being used for the same purpose. The asphaltic and naphthenic oils are slightly superior to the paraffin base oils as regards this property.

A noteworthy development in recent years is the introduction of what are known as 'Extreme Pressure' lubricants. Many of these consist of straight or compounded mineral oils to which are added special preparations or 'dopes' which have the peculiar property of greatly increasing the film strength of the lubricant. Even should the lubricating film break down, it is claimed that certain additives react with the metal surface under extreme conditions and prevent or delay metal to metal seizure. These 'E.P.' lubricants are primarily intended for use in machiner where the oil film encounters exceptionally high pressures, an example being the lubrication of hypoid gears in the back axles of automobiles and other road vehicles. The pressure between the teeth of gears of this type is extremely high and special lubricants have accordingly been developed for this service. Various machines have been developed in order to measure the film strength of E.P. and other lubricants in the laboratory and among these may be mentioned the Timken, Cornell and Almen testers. Other recent developments include the use of certain metallic compounds in order to improve the resistance of lubricating oils to oxidation. It is also claimed that the addition of chromium compounds reduces cylinder wear. Special products which the writer believes to be polymerised hydrocarbon bodies have been introduced, certain of these being used to lower the setting point, others being employed to increase viscosity and to improve viscosity index.

When drafting specifications for the supply of lubricants for any particular purpose it may be noted that while it is naturally essential to decide the type of oil which will be necessary, a knowledge of the various grades of oil commonly available on the market is also desirable. The preparation of lubricating oil specifications usually requires considerable experience, and detailed notes are therefore not included in this chapter. When drafting specifications note should be taken of the type of base which it is desired that the lubricating oil should possess and for many purposes a specification covering specific gravity, flash point and viscosity is totally insufficient.

Among oils of the same type, base and viscosity, the colour may usually be taken as an indication of the degree to which the oil has been refined, a lighter colour indicating more efficient refining. When comparing the colour of lubricating oils, however, it is very important to bear in mind that distillates have a different appearance to residuum oils. A low viscosity index lubricating oil distillate is frequently pale in colour, whereas an oil of similar viscosity prepared from the finest Pennsylvanian Long Residuum would have a green appearance to reflected light and a dark red colour to transmitted light. Notwithstanding the lighter colour of the distillate oil the second oil of course would be of very much higher quality. It is only when comparing oils of similar type that colour bears any relationship to the degree of refining.

RECOVERY OF USED LUBRICATING OILS.

Used petroleum lubricating oils may contain various bodies which are similar in nature to those present in petroleum crude oils. For example, used oils may contain water, mechanical impurities and hard asphalt; when the oils have been used for the lubrication of internal combustion engines they may also be diluted with motor spirit or fuel oil fractions.

As described in the previous section, various refining methods are employed in order to produce lubricating oil from petroleum crude oil, and if used lubricating oils are submitted to similar or modified operations it is possible to produce a satisfactory lubricant. The quality of the recovered oil may be equal to or even better than that of the original oil before use. The recovery of used oils is, therefore, limited by economic rather than by technical considerations.

The methods which may be employed for the recovery of used lubricating oil include :—

(1) Centrifuges, which may be used to remove water and to reduce ash. Settling tanks may also be employed for the same purpose.

(2) The used oil may be warmed and passed through a suitable filter.

(3) The oil may be treated with sulphuric acid followed by decolourising earth or decolourising earth may be used alone.

(4) Solvent refining methods may be employed but as a rule the quantities available do not warrant the use of such processes.

(5) Distillation with steam or under reduced pressure may be employed.

Various methods are thus available for the recovery of used lubricating oil. Settling tanks or centrifuges are frequently employed for the removal of water and mechanical impurities from oil taken from circulating systems. Various filtering systems have been developed and are used in many installations with satisfactory results. Filters may either be employed separately or, alternatively, may form a portion of the circulating system. The adoption of more complicated systems, as, for example, treatment with sulphuric acid and adsorbent earth, is to a considerable extent dependent on there being sufficient quantities of oil conveniently available in order to justify the expense of such processes.

NOTES ON LUBRICATION.

Stationary and Marine Oil Engines.—The lubricating oil specifications of the D.E.U.A. are given in Table on p. 879, and the grade of oil required for any particular engine will usually be indicated by the makers. In general it may be noted that slow speed, stationary and marine Diesel engines usually require a slightly more viscous oil than do engines running at higher speeds. Also as a general rule two-stroke engines usually employ a slightly more viscous oil than four-stroke engines of similar size and type.

Considerably more viscous oils are employed for the lubrication of paraffin engines as these engines are liable to fuel dilution of the oil in the crankcase. Here again two-stroke paraffin engines usually employ a slightly more viscous oil than four-stroke engines.

Bearing, Shafting and Machinery Oils.—For the lubrication of bearing, shafting and machinery plain mineral oils are usually employed. Compounded oils, however, are sometimes used when circumstances demand.

Low viscosity oils are used for light machinery and for very small high speed shafting, somewhat more viscous oils being used for shafting operating at lower speeds. In general, it may be stated that reduced speed and increased bearing pressures usually call for oils of increased viscosity.

Steam Cylinder Oils.—For the lubrication of steam engine cylinders oils of the steam refined cylinder class are usually employed. Where desirable steam cylinder oils may be compounded, oils of somewhat higher flash point being used when steam temperatures are extremely high.

Gas Engine Oils.—While plain mineral oils may be used for the lubrication of gas engines, compounded oils may be used to advantage, particularly when the engine is run on suction gas.

Turbine Oils.—The B.S.I. specification for turbine oils is given on p. 880. Oils employed for this purpose should possess, among other characteristics, good demulsibility and high resistance to oxidation. It should be borne in mind that turbine lubricating oils usually remain in service for a considerable period of time and they should consequently be as stable as possible. Oils which oxidise readily may give trouble by gumming or forming sludges in the lubricating circuit. In use turbine oils may become contaminated with water and consequently the oils should possess the property of separating readily from water, so that this can be drained off periodically from the lubricating oil circuit.

SPECIFICATIONS FOR LUBRICATING OILS FOR USE ON HEAVY OIL ENGINES.

Issued by the Diesel Engine Users Association.

	'L. 1.'		'M. 1.'		'H. 1.'	
Specific Gravity at 60° F.	Not exceeding	0·882	Not exceeding	0·885	Not exceeding	0·888
Closed Flash Point	Not below	400° F.	Not below	400° F.	Not below	400° F.
Viscosity Redwood (Secs.): at 70° F.	Not exceeding	700	Not exceeding	1,250	Not exceeding	1,800
,, ,, at 200° F.	Not less than	50	Not less than	56	Not less than	68
Cold Test or Pour Point	Not higher than	35° F.	Not higher than	35° F.*	Not higher than	35° F.*
Colour (by transmitted light)	Clear red		Clear red		Clear red	
Demulsibility	2 minutes		3 minutes		4 minutes.	
	'L. 2.'		'M. 2.'		'H. 2.'	
Specific Gravity at 60° F.	Not exceeding	0·915	Not exceeding	0·918	Not exceeding	0·920
Closed Flash Point	Not below	375° F.	Not below	375° F.	Not below	375° F.
Viscosity Redwood (Secs.): at 70° F.	Not exceeding	1,300	—		—	
,, ,, at 140° F.	—		Not exceeding	190	Not exceeding	230
,, ,, at 200° F.	Not less than	52	Not less than	60	Not less than	68
Cold Test or Pour Point	Not higher than	35° F.	Not higher than	35° F.	Not higher than	35° F.
Colour (by transmitted light)	Clear red.		Clear red.		Clear red.	
Demulsibility	4 minutes.		6 minutes		8 minutes.	

The oil shall be a STRAIGHT MINERAL OIL. Method of testing: Institution of Petroleum. No tolerances allowed.

Where oils are required for engines in which there is a liability for water to become admixed with the oil in the crankcase, it is advisable for buyers to make the above stipulation regarding the demulsibility test of oils supplied, otherwise the above demulsification clause can be omitted from the specification.

* Where engines are liable to be exposed to abnormally low temperatures, a lower cold test should be specified.

As a general rule the lighter grades are used for very high speed turbines, slightly more viscous grades being employed for medium speed turbines. When the oil is required to lubricate both turbine bearings and also reduction gears a slightly more viscous grade is usually employed.

EXTRACT FROM BRITISH STANDARD TURBINE OIL SPECIFICATION.

B.S.S. 489—1933.)

Grade.	Viscosity (Redwood No. 1) Seconds. At 21° C. (70° F.).		At 60° C. (140° F.).		At 93·3° C. (200° F.).		Closed Flash Point. Not less than	Cold Test.	Sludge. Not Exceeding.	Volatility. Not Exceeding.	Total Acidity mg. KOH. Not Exceeding.	Demulsification. Not Exceeding.	Total Sulphur. Not Exceeding.
	Max.	Min.	Max.	Min.	Max.	Min.			Per Cent.	Per Cent.		Min.	Per Cent.
Light	450	—	75	60	—	40	180° C. (356° F.)	0° C. (32° F.)	1·5	0·5	10	2	0·3
Medium	680	—	90	75	—	42	180° C. (356° F.)	0° C. (32° F.)	1·5	0·5	10	2	0·3
Heavy	850	—	112	90	—	45	180° C. (356° F.)	+ 2° C. (35·6° F.)	1·5	0·5	10	4	0·3
Extra Heavy	1,300	—	140	112	—	51	180° C. (356° F.)	+ 2° C. (35·6° F.)	1·5	0·5	10	6	0·3

Crankchamber Oils.—Oils required for the lubrication of the crankchambers of high speed steam engines should possess properties which closely resemble those required in turbine oils. Both oils are required for bearing lubricating and should possess good resistance to oxidation. A good demulsibility is also required as crankchamber oils may become mixed with water and the property of separating readily from water is therefore desirable.

Air Compressor Oils.—Oils of high quality possessing good resistance to oxidation are usually employed for the lubrication of air compressors. Slightly compounded oils can frequently be used with advantage but the proportion of fatty oil should be small in order to avoid possible oxidation.

GRAPHITE LUBRICATION.

(E. G. Acheson, Ltd.)

Colloidal graphite as a dispersion in oil or water is used for the lubrication of a variety of equipment. It has largely replaced powdered graphite owing to its higher technical quality, and consequently wider range of application. In graphite lubrication purity of the material used is essential; fineness is also important, and in addition, homogeneity of dispersion and unvarying quality must be considered.

Oil containing colloidal graphite made by the Acheson process is now standardised as a lubricant for the assembly and running-in of petrol, high and low speed Diesels, gas and oil engines, by manufacturers. It is smeared on the working faces before assembly and the engine is then run-in with oil containing a lower percentage of the product. It is also used for running-in reduction gear. In high temperature work colloidal graphite is an excellent lubricant, being used for kiln and annealing car bearings, oven chains, die-casting machines, glass bottle machinery and other applications where plain oil would find it difficult to operate. Because of its value at high temperature it is becoming widely used as a constituent for upper cylinder lubricants, particularly in the motoring field where it reduces cylinder wear and discourages sticking piston rings. As a dispersion in water colloidal graphite is employed as a steam cylinder and pump lubricant, being fed, by means of a special lubricator. Here, again, it is being standaridsed by manufacturers of steam engines and pumps.

When a bearing face is lubricated with colloidal graphite there is formed on it an extremely thin lubricating surface; referred to as a graphoid surface, it is made up of flat particles of graphite, submicroscopic in size, lying parallel to the bearing face. In this position the particles offer the maximum lubrication value of the graphite, and protect the bearing face mechanically from corrosion.

Recent work has shown that colloidal graphite is essentially superior to ordinary powdered graphite, because the particles of the former being finer are able to lie uniformly on a bearing face, whereas particles of ordinary graphite if they are too coarse will lie heterogeneously, offering relatively poor protection against corrosion and less efficient lubrication. The graphoid surface formed by colloidal material cannot be removed by bearing pressures and has a considerable value in lubrication at higher loads, where it provides a second line of protection behind the oil film, so preventing metal pick-up.

There is much published information on the subject of graphite lubrication. The Institution of Automobile Engineers Research Department has shown in a series of tests that oil containing colloidal graphite made by the Acheson process added to the sump oil of a new engine reduces cylinder wear by one-half. The National Physical Laboratory conducted a test on a ball bearing assembly which showed that colloidal graphite in oil is suitable for such lubrication. The properties of colloidal graphite may be summarised by the statement that: it raises the critical temperature of an oil, permits lubrication at higher loads, permits a lower oil feed and reduces starting friction in a bearing. Graphited oils are marketed by various oil companies and can be obtained to any specification required. Penetrating oil containing colloidal graphite is also available; it is a superior penetrant and protects surfaces exposed to weathering, besides providing long-lasting lubrication.

See also Descriptive Section XX Part II.

Graphite Products, Ltd.
Tecalemit, Ltd.
A. C. Wells & Co., Ltd.

SECTION XX

PART III

VISCOSITY AND VISCOUS FLOW OF OILS.

(Contributed by Harold Moore, M.Sc.Tech., F.C.S., and
E. Stokoe, M.Inst.Pet.)

VISCOSITY OF OILS.

Mineral oils of any particular origin may vary greatly in viscosity. For example, liquid petroleum products range from motor spirit to extremely viscous fuel and lubricating oils. Further, the viscosity of any particular oil decreases with increasing temperature, so that when considering pumping problems the temperature of the product must be taken into consideration. The rate of change in viscosity with change of temperature varies for different oils. Approximate computations, however, may be made by means of the graphs given in Part II on pp. 875 and 876. A table showing the properties of typical petroleum lubricating oils is given on p. 872.

CHANGE OF VISCOSITY BY CHANGE OF PRESSURE.

Experiments made by the National Physical Laboratory have shown that at a pressure of 9,000 atmospheres, the viscosity of animal oils was about 4 times as much as at atmospheric pressure, but in the case of mineral oils the increase was as much as 16-fold.

VISCOUS AND TURBULENT FLOW.

(1)

Although there are two kinds of flow, viz. 'stream line' or 'viscous' flow, and 'turbulent' flow, for all practical conditions, heavy oils, in general, follow the law governing viscous flow.

The frictional loss due to purely viscous flow is given by the equation

$$H = 0 \cdot 0066875 \frac{TVP}{d^2}$$

where,
H = loss of head per 100 ft. of piping; P = weight in lbs. per cub. ft.;
T = viscosity measured in secs. by Redwood No. 2 Viscometer; d = diameter of pipe in ins.
V = velocity in ft. per sec.; (*G. & J. Weir, Ltd., Cathcart, Glasgow.*)

If
Red. = Redwood viscosity number;
W = weight in tons per hour of oil passing through pipe
δ = oil density;

then,

$$T = 4 \cdot 895 \frac{\text{Red.}}{\delta}; \quad V = 1 \cdot 83 \frac{W}{d^2 \delta} \text{ ft. per sec.};$$

$$H = 2 \cdot 09 \frac{\text{Red. } V.}{d^2} = 3 \cdot 73 \frac{\text{Red. } W.}{d^4 \delta} \qquad (H. R. K.)$$

Example.—124 tons (W) per hour of oil, Redwood No. 2 viscosity 579 seconds at 20° C., density 0·9495 at 20° C., is to be pumped through a straight length of 9-in. diameter (d) pipe 50 ft. long. What is the frictional loss (H)?

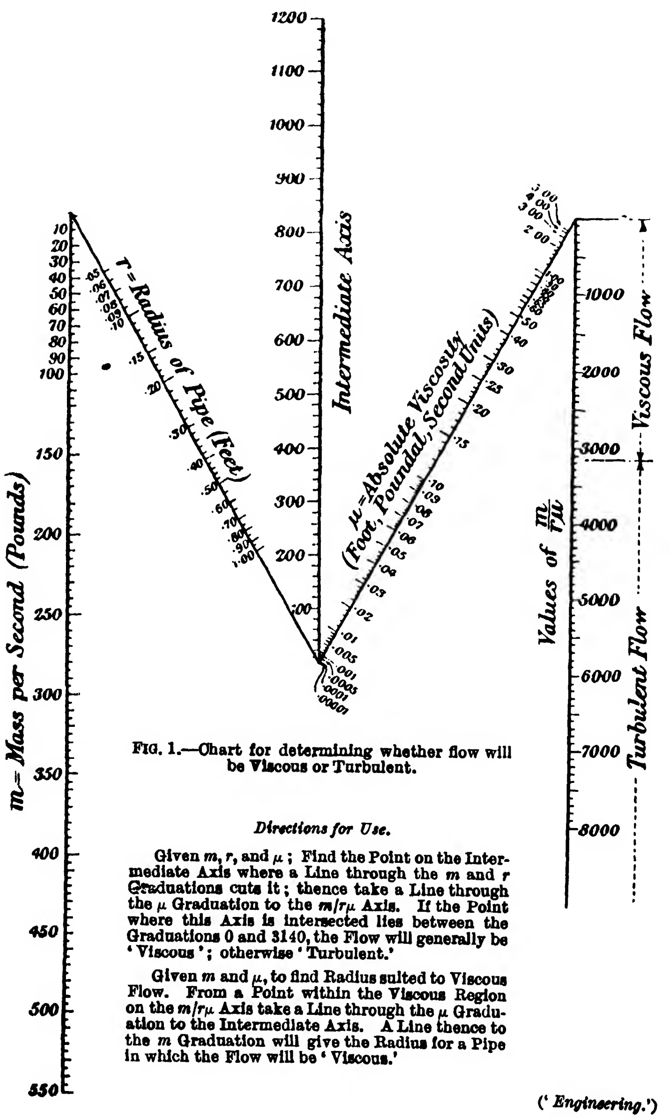


FIG. 1.—Chart for determining whether flow will be Viscous or Turbulent.

Directions for Use.

Given m, r, and μ; Find the Point on the Intermediate Axis where a Line through the m and r Graduations cuts it; thence take a Line through the μ Graduation to the $m/r\mu$ Axis. If the Point where this Axis is intersected lies between the Graduations 0 and 3140, the Flow will generally be 'Viscous'; otherwise 'Turbulent.'

Given m and μ, to find Radius suited to Viscous Flow. From a Point within the Viscous Region on the $m/r\mu$ Axis take a Line through the μ Graduation to the Intermediate Axis. A Line thence to the m Graduation will give the Radius for a Pipe in which the Flow will be 'Viscous.'

('*Engineering.*')

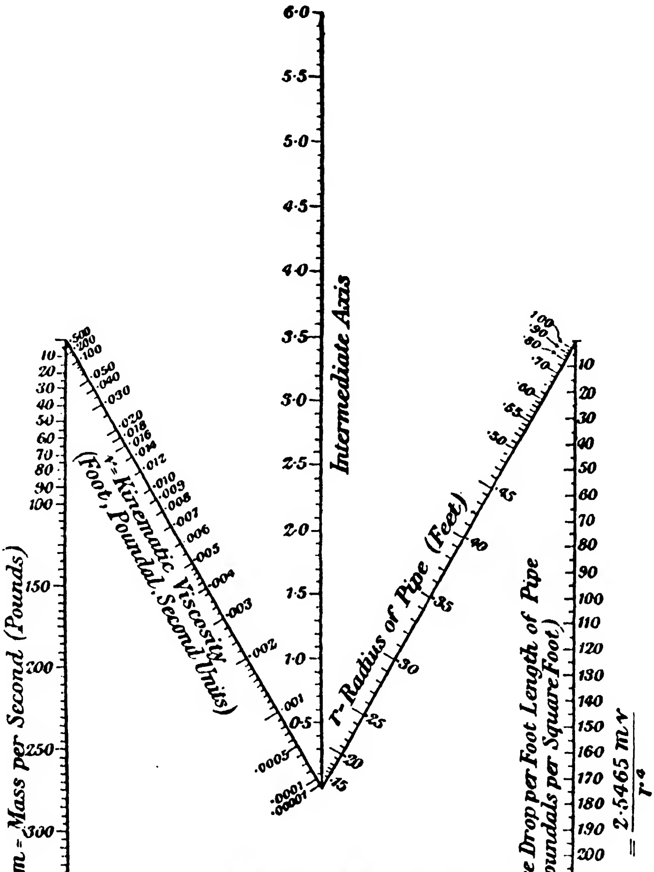


FIG. 2.—Viscous Flow (Absolute Units).

'(*Engineering*.')

TURBULENT FLOW.
(ABSOLUTE UNITS.)

First Term in Lees' Equation,

$$p = \frac{\mu^{0.35}\, m^{1.65}}{55\cdot08\, r^{4.65}\, \rho} + \frac{\cdot0001824\, m^2}{r^5\, \rho}$$

= Pressure Drop in Poundals per Sq. Ft., per Lineal Ft. of Pipe = $p_1 + p_2$, say.

Add the Reading, p_1, on the Right-Hand Axis of this Chart, to the Reading, p_2, on the Right-Hand Axis of the Chart for the Second Term in the above Equation. The Sum is the Total Pressure Drop.

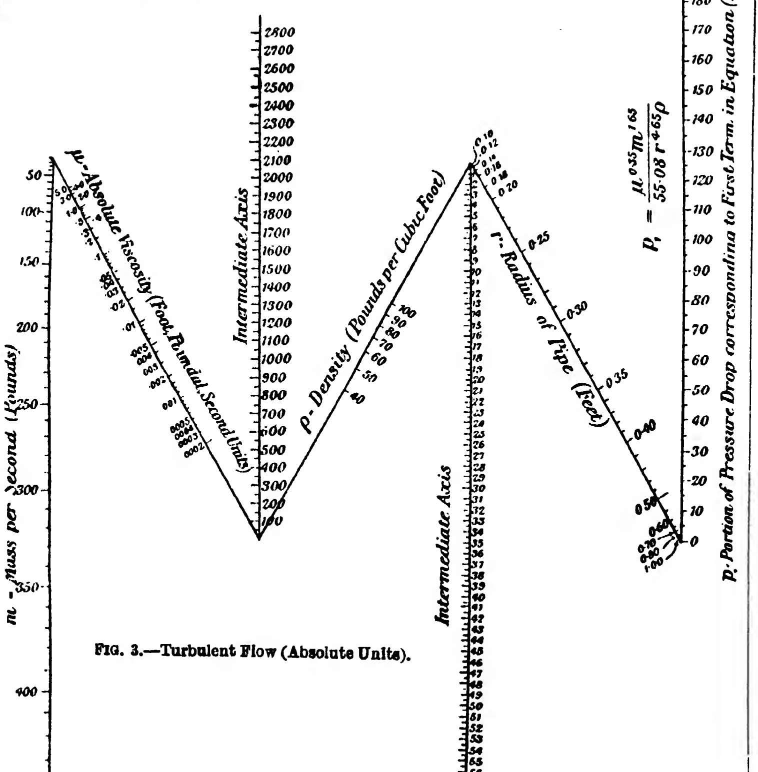


FIG. 3.—Turbulent Flow (Absolute Units).

('*Engineering*.')

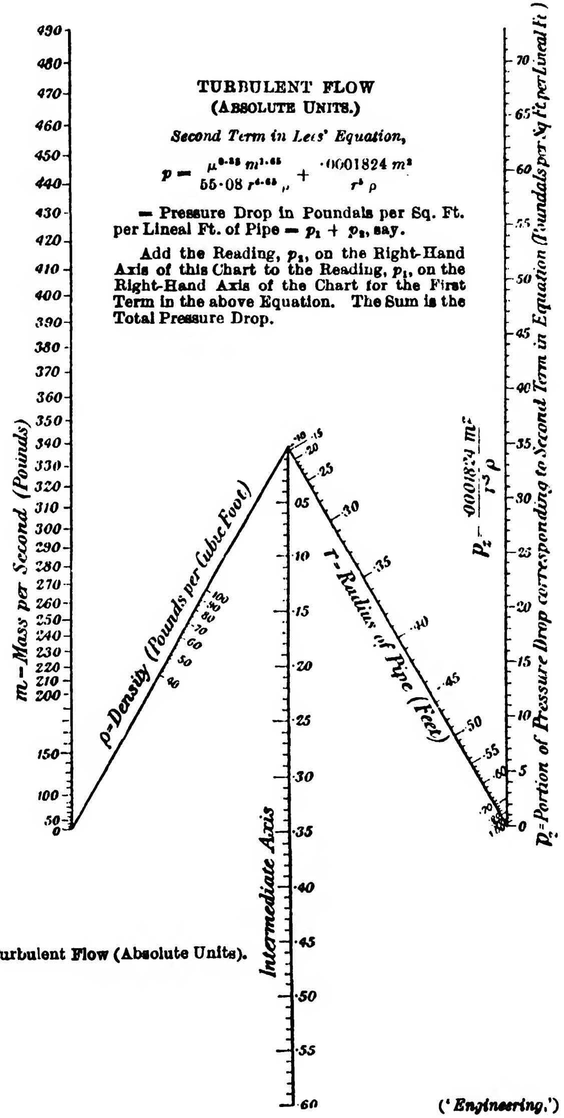


FIG. 4.—Turbulent Flow (Absolute Units).

('*Engineering*.')

Red. = 579, δ = 0·9495. therefore,

$$H = 3 \cdot 73 \frac{579 \times 124}{9^4 \times 0 \cdot 9495} = 3 \cdot 73 \frac{579 \times 124}{6561 \times 0 \cdot 9495} = \cdot 43 \text{ ft.};$$

or

$$43 \times \frac{50}{100} = 21 \cdot 5 \text{ ft. for } 50 \text{ ft.}$$

See also 'The Design of Oil Fuel Pipe Lines,' by W. G. Watkins, *Engineering, Dec.* 19 *and* 26, 1924.

(2)

Oils of various degrees of viscosity are frequently pumped through pipes from the oilfields to ships or to distant storage tanks. To determine the power required in a given case, and to choose the most economical installation of pumping plants and pipes, bearing in mind the life of the field, it is necessary to know the frictional resistance encountered by the liquids when so transported. Such resistance will follow one or other of two laws according to the viscosity and velocity of the liquid, and it is desirable to have at hand ready means of determining which law to apply in a given case. Having determined this, one proceeds to ascertain the magnitude of the resistance. If the flow is viscous, or, as it is sometimes called, stream-line, all the particles then moving parallel to the axis of the pipe, the resistance can be calculated in terms of the viscosity, density, and velocity. If, however, the flow is turbulent, frictional resistance cannot be found from first principles, and resort must be had to experiment, as in the case of the flow of water. Extensive experiments have been carried out with the object of discovering a law connecting viscosity, density, and velocity when the flow is of a turbulent character, and, in this connection, the work done at the National Physical Laboratory, Teddington, takes a foremost place. Various formulæ have been deduced from these experiments, and, of these, probably none is so reliable as that of Professor C. H. Lees; the charts, figs. 1, 2, 3, and 4, are based on his formula.

As the handling of the formulæ for both stream-line and turbulent flow is somewhat difficult in practice, especially when trials must be made of several different schemes, attention has been directed to the use of graphic methods for displaying the results. The alignment charts shown were prepared for the purpose by Mr. George Higgins, formerly lecturer in Civil Engineering at the University of Melbourne.

The chart shown in fig. 1 may be used for determining which of the two laws will be followed when the quantity flowing, its viscosity, and the size of the pipe are known. Directions for the use of this chart are given at its foot, and other uses to which it can be applied are also indicated. For liquids of high initial viscosity, the practice on the oilfields is to apply heat at intervals along the pipe line, and thus reduce the resistance to flow by reducing the viscosity. When this is done, the average viscosity between heating stations is estimated, and the resistance calculated for this average.

The chart fig. 2 is to be used when the chart fig. 1 indicates that the flow will be 'viscous,' *i.e.* stream-line. Given the quantity and kinematic viscosity, *i.e.* the ratio of the absolute viscosity to the density, then a straight line drawn through the axes for mass and kinematic viscosity at the appropriate graduations, will cut the intermediate axis in a certain point. From this point, a straight line drawn through the appropriate point on the radius axis will cut the right-hand axis a a graduation corresponding to the loss of pressure per unit length. It will be seen that swinging the latter line up or down about the said point on the intermediate axis as centre, will show how the adoption of one radius or another will affect the loss of pressure, and the most suitable radius can thus be more quickly found than by repeated calculations using the formula.

Charts figs. 3 and 4 are to be used when the chart fig. 1 indicates that the flow will be 'turbulent.' Here the complexity of the formula rendered it necessary to separate the factors of viscosity and density, and also to construct two charts, the results of which are to be added together. The method of using these two charts is quite similar to that explained when describing the chart fig. 2, except that the chart fig. 3 involves the drawing of three lines across the graduated axes instead of two. Their employment dispenses with much troublesome tentative calculation when the object is to determine the most suitable combination of pumps and pipes in a given case.

In practice, the lines are not, as a rule, actually drawn across the axes, a straight edge or thread being employed; a very suitable appliance for the purpose is a transparent set-square.

(Abstract—'*Engineering*,' *April* 27, 1928.)

PUMPS FOR VISCOUS LIQUIDS.

See Section XIX, Part III.

SECTION XX

PART IV

SHAFTING AND SHAFTS.

Contributed by H. W. Swift, D.Sc., M.I.Mech.E.)

For a shaft carrying torque only and free from bending stress the horse-power transmitted is :

$$HP = \frac{ND^3S}{321,400};$$

where N = r.p.m. ; D = diameter in inches ; S = working shear stress, lbs. per sq. in.

If S = 6,400 lbs./sq. in., a safe working stress for mild steel under reasonably constant torque

$$HP = \frac{ND^3}{50}$$

and the diameter required :—

$$D = 3 \cdot 7 \sqrt[3]{\frac{HP}{N}}$$

Shaft Horse-Power for Torsion.

The table (page 890) is based on the formula $HP = \frac{ND^3}{50}$; where N is the speed in r.p.m. and D the shaft diameter in inches. This formula assumes a working shear stress of 6,440 lbs. per sq. in. or 2·88 tons per sq. in. For other working stresses the horse-power varies in proportion.

If the torque varies over a known range, without reversal, the same formulæ may be applied provided the maximum value of the torque is employed. To allow for cyclic torque variations in steam engines, it is a common practice to increase the diameter D, as calculated above, by 25 per cent. ; alternatively the shaft may be designed as above to carry twice the mean horse-power.

For shafts subject to cyclic reversals of stress the diameter should be increased 25 to 30 per cent.

The formula $D = k \sqrt[3]{\frac{HP}{N}}$ may be employed for mill shafting, the value of k being chosen according to the factor of safety and pulley distribution. Common values are as follows :—

Main shafts, $k = 4 \cdot 3$.

Line shafts with some pulleys, $k = 3 \cdot 7$.

Counter shafts, $k = 3 \cdot 4$.

Shafts under Combined Bending and Torsion.

When calculating bending moments for a length of shafting it is safe to assume that each span is self-supporting, and that the reactions act in the centre line of the bearings. The greatest bending moment always occurs *at* the point of application of a load or *at* a bearing, not at an intermediate point.

The shaft of a stiff rotor may be considered in two parts each assumed to be encastre in the rotor and simply supported in its bearing.

TABLE I.—SHAFT HORSE-POWER FOR TORSION.

R.P.M.	Diameter in Inches. 1	1¼	1½	1¾	2	2¼	2½	2¾	3	3½
	Horse-power Transmitted.									
10	0·20	0·39	0·67	1·07	1·60	2·28	3·12	4·16	5·40	8·57
20	0·40	0·78	1·35	2·14	3·20	4·56	6·25	8·32	10·8	17·1
30	0·60	1·17	2·02	3·21	4·80	6·54	9·37	12·5	16·2	25·7
40	0·80	1·56	2·70	4·28	6·40	9·12	12·5	16·6	21·6	34·3
50	1·00	1·95	3·37	5·35	8·00	11·4	15·6	20·8	27·0	42·9
100	2·00	3·90	6·75	10·7	16·0	22·8	31·2	41·6	54·0	85·7
150	3·00	5·85	10·1	16·0	24·0	34·2	46·9	62·4	81·0	129
200	4·00	7·80	13·5	21·4	32·0	45·6	62·5	83·2	108	171
250	5·00	9·75	16·9	26·7	40·0	57·0	78·1	104	135	214
300	6·00	11·7	20·2	32·1	48·0	68·4	93·7	125	162	257
350	7·00	13·6	23·7	37·4	56·0	79·8	109	146	189	300
400	8·00	15·6	27·0	42·8	64·0	91·2	125	166	216	343
450	9·00	17·6	30·4	48·1	72·0	103	141	187	243	386
500	10·0	19·5	33·7	53·5	80·0	114	156	208	270	429
1,000	20·0	39·0	67·5	107	160	228	312	416	540	857
2,000	40·0	78·0	135	214	320	456	625	832	1,080	1,715
3,000	60·0	117	202	321	480	684	937	1,248	1,620	2,570

TABLE I (*continued*).

R.P.M.	Diameter in Inches.									
	4	4½	5	5½	6	7	8	9	10	12
	Horse-Power Transmitted.									
10	12·8	18·2	25·0	33·3	43·2	68·6	102	146	200	346
20	25·6	36·5	50·0	66·6	86·4	137	205	292	400	691
30	38·4	54·8	75·0	99·9	130	206	307	437	600	1,037
40	51·2	73·0	100	133	173	274	410	583	800	1,382
50	64·0	91·2	125	166	216	343	512	729	1,000	1,728
100	128	182	250	333	432	686	1,024	1,458	2,000	3,460
150	192	274	375	499	648	1,029	1,536	2,190	3,000	5,180
200	256	365	500	665	864	1,372	2,050	2,920	4,000	6,910
250	320	456	625	832	1,080	1,715	2,560	3,650	5,000	8,640
300	384	547	750	998	1,296	2,060	3,070	4,370	6,000	10,370
350	448	639	875	1,165	1,512	2,400	3,580	5,100	7,000	12,100
400	512	730	1,000	1,331	1,728	2,740	4,100	5,830	8,000	13,820
450	576	821	1,125	1,497	1,944	3,090	4,610	6,560	9,000	15,550
500	640	912	1,250	1,664	2,160	3,430	5,120	7,290	10,000	17,280
1,000	1,280	1,825	2,500	3,330	4,320	6,860	10,240	14,580	20,000	34,600
2,000	2,560	3,750	5,000	6,650	8,640	13,720	20,500	29,200	40,000	69,100
3,000	3,840	5,470	7,500	9,980	12,960	20,600	30,700	43,700	60,000	103,700

The shaft design is based either

(*a*) on the *Equivalent Twisting Moment* T_e, which would produce the same shear stress as the combined system :—

$$T_e = \sqrt{M^2 + T^2}$$

or (*b*) on the *Equivalent Bending Moment* M_e, which would produce the same fibre stress as the combined system :—

$$M_e = \tfrac{1}{2}(M + \sqrt{M^2 + T^2})$$

If the equivalent twisting moment is used :—

$$d = 1{\cdot}72\sqrt[3]{\frac{T_e}{f_s}}$$

If the equivalent bending moment is used :—

$$d = 2{\cdot}17\sqrt[3]{\frac{M_e}{f_t}}$$

For mill shafting it is sometimes assumed that M = T, in which case the diameter computed for torque only should be increased by one in eight.

Since the permissible tensile stress for steel is commonly taken as twice the permissible shear stress, while the equivalent bending moment is never as much as twice the equivalent twisting moment, it would appear safer to base the design of steel shafting on the equivalent twisting moment, a practice which is commonly adopted in America.

But when account is taken of the fact that the bending moment produces cyclic reversals of stress, while the twisting moment is usually tolerable constant, the advantage of the equivalent twisting moment is lost and a design based as follows on the equivalent bending moment is more convenient :—

Fatigue range for reversed direct stress $= \frac{f}{3}$ or more.

" " " " shear " $= \frac{f}{6}$ "

Allowing a factor of safety of 2 on the fatigue range, the working stresses will be :—

$$f_t = \frac{f}{6}, \quad f_s = \frac{f}{12}.$$

Under conditions where no bending occurs, the stress is steady and the shaft may be designed with a working shear stress $= \frac{f}{6}$. This produces a working fibre stress of $\frac{f}{6}$ also. Under conditions where the stress is entirely due to bending it is subject to cyclic reversals, and the shaft must be designed with a working fibre stress of $\frac{f}{6}$, which produces a shear stress of $\frac{f}{12}$. Since the working fibre stress has the same value $\frac{f}{6}$ in both extreme cases and does not vary substantially for intermediate combinations it is satisfactory and convenient to design in general on the equivalent bending moment and allow a factor of safety of 6 for a solid shaft and 8 for a shaft with keyway.

Stiffness of Shafts.

(*a*) *Torsional.*—The angle of twist for a shaft subjected to torque producing a shearing stress of 6,400 lb./sq. in. (see Table I) is 1 degree in 16 diameters. If a smaller twist is demanded the stress and power must be reduced in proportion.

(*b*) *Lateral.*—The lateral deflection of shafting is commonly limited to $\frac{1}{100}$ in. per ft. length between bearings. For a shaft (diam. *d* inches) without pulleys this condition determines the permissible distance (L ft.) between bearings :—

$$L = 6{\cdot}3d^{\frac{2}{3}}.$$

For shafting with normal complement of pulleys a common rule is :—

$$L = 5d^{\frac{2}{3}}.$$

(*c*) *Critical Speeds.*—An unloaded shaft will whirl if its length between self-aligning bearings is : $L = 180\sqrt{\frac{d}{N}}$ ft. The whirling speed of a shaft carrying pulleys or other lateral loads may be estimated from the formula :—

$$\frac{1}{N^2} = \frac{1}{N_0^2} + \frac{1}{N_1^2} + \cdots$$

When applied to self-aligning bearings: $N_0 = 33{,}000\,\frac{d}{L^2}$, is the whirling speed without load.

TABLE II.—EQUIVALENT BENDING AND TURNING MOMENTS.
(Stresses in Tons per Square Inch.)

Fibre Stress for M_e.	4	4½	5	5½	6	6½	7	7½	8
Shear Stress for T_e.	2	2¼	2½	2¾	3	3¼	3½	3¾	4
Diameter, Ins.	Equivalent Bending and Turning Moments, lbs. inches.								
1	880	990	1,100	1,210	1,320	1,430	1,540	1,650	1,760
1¼	1,375	1,547	1,719	1,891	2,060	2,230	2,410	2,580	2,750
1½	1.980	2,230	2,470	2,720	2,970	3,220	3,460	3,710	3,960
1¾	2,690	3,030	3,370	3,710	4,040	4,380	4,720	5,050	5,390
2	3,520	3,960	4,400	4,840	5,280	5,720	6,160	6,600	7,040
2¼	4,450	5,010	5,570	6,130	6,680	7,240	7,800	8,350	8,910
2½	5,500	6,190	6,880	7,560	8,250	8,940	9,620	10,310	11,000
2¾	6,650	7,490	8,320	9,150	9,980	10,810	11,650	12,480	13,310
3	7,920	8,910	9,900	10,890	11,880	12,870	13,860	14,850	15,840
3½	10,780	12,130	13,470	14,820	16,170	17,520	18,860	20,200	21,600
4	14,080	15,840	17,600	19,360	21,100	22,900	24,600	26,400	28,200
4½	17,820	20,000	22,400	24,500	26,700	29,000	31,200	33,400	35,600
5	22,000	24,800	27,500	30,200	33,000	35,700	38,500	41,200	44,000
5½	26,600	29,900	33,300	36,600	39,900	43,300	46,600	49,900	53,200
6	31,700	35,600	39,600	43,600	47,500	51,500	55,400	59,400	56,400
7	43,100	48,500	53,900	59,300	64,700	70,100	75,500	80,800	86,200
8	56,300	63,400	70,400	77,400	84,500	91,500	98,600	105,600	112,600
9	71,300	80,200	89,100	98,000	106,900	115,800	124,700	133,600	142,600
10	88,000	99,000	110,000	121,000	132,000	143,000	154,000	165,000	176,000
12	126,700	142,600	158,400	174,200	190,100	206,000	222,000	238,000	253,000

N_1, N_2, etc., are the whirling speeds due separately to individual loads W_1, W_2, etc., at distances A_1, A_2, etc., feet from a bearing, calculated from formulæ such as :—

$$N_1 = 9{,}500 \frac{d^2}{A(L-A)} \sqrt{\frac{L}{W}}$$

With fixing bearings :—

$$N_o' = 75{,}000 \frac{d}{L^2}$$

$$N_1' = N_1 \times \frac{L}{\sqrt{A(L-A)}}$$

Temperature Changes.—Longitudinal expansion of steel shafting $= \frac{T}{15{,}000}$ in. per ft. for T° F. rise of temperature. All probable temperature changes will be covered by an allowance of $\frac{1}{16}$ in. per 10 ft. of shaft.

Allowances for Keyways and Fillets.—According to H. F. Moore, the presence of a keyway of width w and depth h in a shaft of diameter d—

reduces the elastic strength in the ratio : $1 - 0{\cdot}2 \frac{w}{d} - 1{\cdot}1 \frac{h}{d}$

increases the angle of twist in the ratio : $1 + 0{\cdot}4 \frac{w}{d} + 0{\cdot}7 \frac{h}{d}$

Endurance tests by Batson show that the weakening effect of British Standard Keyways is about 20 per cent.

It is a common practice to allow 25 per cent. off the nominal working stress when keyways are employed.

According to E. F. Garner the stress in a fillet of radius r_o in a shaft reduced from radius R to r bears to the stress in the smaller shaft the ratio :—

(a) Fibre stress ratio : $\left(0{\cdot}26 + 0{\cdot}46 \sqrt{\frac{r}{r_o}}\right) \left\{2{\cdot}23 - 1{\cdot}23 \left(\frac{r}{R}\right)^{2\cdot7}\right\}$

(b) Shear stress ratio : $\left(0{\cdot}43 + 0{\cdot}15 \sqrt{\frac{r}{r_o}}\right) \left\{1{\cdot}88 - 0{\cdot}88 \left(\frac{r}{R}\right)^{8}\right\}$

Shafting for Passenger Steamships.

The following particulars are abstracted from the 'Instructions as to the Survey of Passenger Steamships' issued by the Board of Trade.

Ingot steel for shafts has generally a tensile strength of 28–32 tons per sq. in., and fulfils the conditions that the tensile plus elongation per cent. on standard test-piece shall not be less than 57. Couplings of ingot steel shafts are forged from the solid. Cast steel webs for built crankshafts have a tensile strength not exceeding 32 tons per sq. in., and are such that tensile plus elongation is not less than 50.

Turbine-driven (intermediate) shafts are not less in diameter than :—

$$d = \sqrt[3]{\frac{\text{S.H.P.} \times F}{\text{R.P.M.}}}$$

where F = 64 or 58 according to the conditions of service. Wheel shafts of geared installations have diameters $1{\cdot}05\ d$ to $1{\cdot}10\ d$ according to the arrangement of gearing.

Engine-driven (intermediate) shafts are not less in diameter than :—

$$d = \sqrt[3]{\frac{D^2 \times S \times p}{f(r+2)}}$$

where D ins. = equivalent diameter of L.P. cylinder ;
S ins. = stroke, p lb./sq. in. = working boiler pressure;
r = swept volume ratio L.P./H.P. ;
f has values from 1,350 for two cranks at 180° to 2,400 for two cranks at 120°, or four cranks balanced.

Crankshafts of screw reciprocating engines have diameters not less than $1{\cdot}05\ d$.

Thrust shafts transmitting torque are increased in diameter to $1{\cdot}05\ d$ at the collars. Tube shafts have diameters not less than $1{\cdot}05\ d$, and $1{\cdot}075\ d$ where exposed to sea-water.

The minimum diameter for tail shafts is $d + \frac{\text{propeller diameter}}{K}$ where K = 144 or 100 according as a continuous liner is or is not fitted.

The minimum liner thickness is $\frac{\text{tube or tail shaft diameter} + 9\frac{1}{2}}{32}$ in way of bushes and reduced by 25 per cent. between bushes.

These figures apply also to tube and propeller shafts for heavy oil engines.

SECTION XX

PART V

SPRINGS.

(Contributed by R. G. Batson, M.I.C.E.)

I.—Cylindrical Helical Springs, Close-coiled, Axially Loaded.

Section and Dimensions of Wire. In.	Shear Stress in Lb. per Sq. In. f_s.	Load in Lb. W.	Axial Deflection in Ins. δ.
Circular (dia. = d)	$K_1 \cdot \frac{16WR}{\pi d^3}$	$\frac{1}{K_1} \cdot \frac{f_s \pi d^3}{16R}$	$K_2 \cdot \frac{64WR^3N}{Gd^4}$
Elliptical (D and d)	$K_1 \cdot \frac{16WR}{\pi Dd^2}$	$\frac{1}{K_1} \cdot \frac{f_s \pi Dd^2}{16R}$	$K_2 \cdot \frac{32(D^2 + d^2)WR^3N}{GD^3d^3}$
Rectangular { b parallel to Spring Axis, t radial to Spring Axis } (see note below)	$K_1 \cdot \frac{(3b + 1\cdot8t)WR}{b^2t^2}$	$\frac{1}{K_1} \cdot \frac{f_s b^2t^2}{(3b + 1\cdot8t)R}$	$\frac{19\cdot6WR^3N}{Gt^3(b - 0\cdot56t)}$
Square (side = s)	$K_1 \cdot \frac{4\cdot8WR}{s^3}$	$\frac{1}{K_1} \cdot \frac{f_s s^3}{4\cdot8R}$	$\frac{44\cdot7WR^3N}{Gs^4}$

W = Axial load on spring in lb.
R = Mean coil radius in inches.
G = Modulus of rigidity in lb. per sq. in. (= 12×10^6 for steel).
N = Number of effective coils. (For compression springs this is two less than the apparent number, owing to flattening at ends of the bases.)
f_s = Safe shear stress in lb. per sq. in.

The deflection formula for rectangular wire is accurate to 1 per cent. where $\frac{'b'}{t}$ is between 1 and 3. Where 'b' is less than 't' an approximate value is obtained by interchanging 'b' and 't' in the formula.

For approximate results K_1 and K_2 may be taken as 1, but for accurate estimations:

$$K_1 = \frac{4c - 1}{4c - 4} + \frac{0\cdot615}{c}$$

and

$$K_2 = 1 + \frac{0\cdot5}{c^2}$$

where c (the spring index) = $\frac{2R}{d}$ r $\frac{2R}{\ }$ or $\frac{2R}{\ }$

The frequency of vibration per minute of a helical compressure spring $= \dfrac{62 \cdot 4dp\sqrt{G}}{R^2L}$

where p = pitch of the coils in inches.

L = effective length of the spring in inches.

(Other symbols as above.)

II.—Cylindrical Helical Springs, Eccentrically Loaded Parallel to the Axis.

The load W applied at a distance mR from the axis. For open-coiled springs the obliquity of the coils to the horizontal = α.

		Max. Shear Stress Lb. per Sq. In.	Deflection in Ins.
Circular wire dia. = d. Ins.	Close-coiled	$\frac{16WR}{\pi d^3}(1 + m) + \frac{16W}{3\pi d^2}$	$\frac{64WR^3N}{Gd^4}\left(1 + \frac{9}{10}m^2\right)$
	Open-coiled	—	$\frac{64WR^3N}{Gd^4}\left[\frac{8(1 + m^2) + \cos^2\alpha(2 + m^2)}{10}\right]$

(For the meaning of the symbols see Section I., p. 895.)

III.—Conical Helical Springs, Close-coiled, Axially Loaded.

Section and Dimensions of Wire. In.	Shear Stress in Lb. per Sq. In. f_s.	Load in Lb. W.	Axial Deflection in Ins. δ.
Circular (dia. = d) . . .	$\frac{16WR}{\pi d^3}$	$\frac{f_s\pi d^3}{16R}$	$\frac{16WR^3L}{G\pi d^4}$
Rectangular . . . . $\begin{cases} b \text{ parallel to Spring Axis} \\ t \text{ radial to Spring Axis} \end{cases}$	$\frac{9WR}{2bt^2}$	$\frac{2f_sbt^2}{9R}$	$\frac{1 \cdot 8WR^2L(b^2 + t^2)}{Gb^3t^3}$

W = Axial load on spring in lb.
R = *Maximum* mean radius of coils in inches.
G = Modulus of rigidity in lb. per sq. in. (= 12×10^6 for steel).
L = Length of developed spring in inches.

For Sections I, II and III the safe *steady* shear stress for heat-treated steel may be taken as
70,000 lb. per sq. in. for wire up to $\frac{1}{4}$ in. diameter.
60,000 lb. per sq. in. for wire above $\frac{1}{4}$ in. and up to $\frac{1}{2}$ in. diameter.
50,000 lb. per sq. in. for wire above $\frac{1}{2}$ in. diameter.

Steel springs (wire up to $\frac{1}{4}$ in. diameter) subjected to repetitions of deflection may be designed for a maximum shear stress of 50,000 lb. per sq. in. For valve springs where surging may be experienced this stress should be reduced by 25 per cent.

The values of the safe stresses given above apply to heat-treated normal commercial spring wire. Many such wires suffer from a surface effect which lowers the resistance to repeated stresses. For specially prepared wire, up to $\frac{1}{4}$ in. diameter, the maximum shear stress under repetitions of deflection may be taken as 70,000 lb. per sq. in.

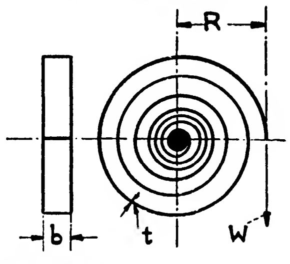
IV.—Cylindrical Helical Springs, Close-coiled, under axial Twist (Helical Power Springs).

Section and Dimensions of Wire. In.	Bending Stress in Lb. per Sq. In. f_t.	Axial Torque in Lb. In. M.	Twist of Free End (one end fixed) in Radians. a.
Circular (dia. = d) . .	$\frac{32M}{\pi d^3}$	$\frac{f_t \pi d^3}{32}$	$\frac{128MRN}{Ed^4}$
Rectangular . . . (b parallel to Spring Axis, t radial to Spring Axis)	$\frac{6M}{bt^2}$	$\frac{f_t bt^2}{6}$	$\frac{24\pi MRN}{Ebt^3}$
Square (side = s) . . .	$\frac{6M}{s^3}$	$\frac{f_t s^3}{6}$	$\frac{24\pi MRN}{Es^4}$

M = Axial torque on spring in lb. in.
E = Young's modulus of elasticity in lb. per sq. in. (30×10^6 for steel).
N = Number of effective coils in spring.

(the safe bending stress) for wires below $\frac{1}{4}$ in. diameter may be taken as 180,000 lb. per sq. in. for steady loading and 100,000 lb. per sq. in. for repeated loading. This assumes heat-treated and tempered steel wire.

V.—Spiral Power Springs (*e.g.* Clock Springs).



Section and Dimensions of Wire.	Bending Stress in Lb. per Sq. In. f_t.	Load in Lb. W.	Angle turned through by Centre Shaft in Radians (Outer end fixed). a.
Circular (dia. = d)	$\frac{64WR}{\pi d^3}$	$\frac{f_t \pi d^3}{64R}$	$\frac{64WRL}{E\pi d^4}$
Rectangular (as indicated in fig.) . .	$\frac{12WR}{bt^2}$	$\frac{f_t bt^2}{12R}$	$\frac{12WRL}{Ebt^3}$
Square (side = s)	$\frac{12WR}{s^3}$	$\frac{f_t s^3}{12R}$	$\frac{12WRL}{Es^4}$

L = length of developed spring in inches.
(Other symbols as for Section IV.)

Steel Helical Springs—Round Wire.

(Contributed by R. A. Sturgeon, A.M.I.C.E.)

Table of Safe Loads and Stiffness.

The stresses from which the safe loads are calculated are given for each size of wire. The stiffness—i.e. the ratio of load to deflection produced—is based on a single coil spring. By dividing the figure given in the table by the number of coils in the spring, the load required to extend or compress the spring 1 in. is obtained. The value of G is assumed to be 12,000,000 lbs. per sq. in.

16 Gauge—0·064 in. (= $\frac{1}{16}$ in.+) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
$\frac{3}{8}$	19·2	477
$\frac{7}{16}$	16·5	301
$\frac{1}{2}$	14·4	201
$\frac{9}{16}$	12·8	141
$\frac{5}{8}$	11·5	103
$\frac{3}{4}$	9·6	59·6
$\frac{7}{8}$	8·2	37·6
1	7·2	25·2
$1\frac{1}{8}$	6·4	17·7
$1\frac{1}{4}$	5·8	12·9
$1\frac{3}{8}$	5·2	9·7
$1\frac{1}{2}$	4·8	7·5
$1\frac{3}{4}$	4·1	4·7
2	3·6	3·1

15 Gauge—0·072 in. (= $\frac{5}{64}$ in.−) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
$\frac{3}{8}$	27·4	764
$\frac{7}{16}$	23·5	481
$\frac{1}{2}$	20·5	322
$\frac{9}{16}$	18·2	226
$\frac{5}{8}$	16·4	165
$\frac{3}{4}$	13·7	95·6
$\frac{7}{8}$	11·7	60·1
1	10·3	40·3
$1\frac{1}{8}$	9·1	28·3
$1\frac{1}{4}$	8·2	20·6
$1\frac{3}{8}$	7·5	15·5
$1\frac{1}{2}$	6·8	11·9
$1\frac{3}{4}$	5·9	7·5
2	5·1	5·0

14 Gauge—0·08 in. (= $\frac{5}{64}$ in.+) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
$\frac{3}{8}$	37·5	1,165
$\frac{7}{16}$	32·2	734
$\frac{1}{2}$	28·2	491
$\frac{9}{16}$	25·0	345
$\frac{5}{8}$	22·5	252
$\frac{3}{4}$	18·8	146
$\frac{7}{8}$	16·1	91·7
1	14·1	61·4
$1\frac{1}{8}$	12·5	43·1
$1\frac{1}{4}$	11·2	31·5
$1\frac{3}{8}$	10·2	23·6
$1\frac{1}{2}$	9·4	18·2
$1\frac{3}{4}$	8·0	11·5
2	7·0	7·7
$2\frac{1}{4}$	6·2	5·4
$2\frac{1}{2}$	5·6	3·9

13 Gauge—0·092 in. (= $\frac{3}{32}$ in.) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
$\frac{3}{8}$	57	2,038
$\frac{7}{16}$	49·0	1,283
$\frac{1}{2}$	42·8	860
$\frac{9}{16}$	38·1	603
$\frac{5}{8}$	34·3	440
$\frac{3}{4}$	28·5	255
$\frac{7}{8}$	24·5	160
1	21·4	107
$1\frac{1}{8}$	19·0	75·5
$1\frac{1}{4}$	17·1	55·0
$1\frac{3}{8}$	15·6	41·3
$1\frac{1}{2}$	14·3	31·8
$1\frac{3}{4}$	12·2	20·1
2	10·7	13·4
$2\frac{1}{4}$	9·5	9·4
$2\frac{1}{2}$	8·6	6·9

12 Gauge—0·104 in. (= $\frac{7}{64}$ in.−) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
$\frac{3}{8}$	82	3,327
$\frac{7}{16}$	71	2,096
$\frac{1}{2}$	62	1,404
$\frac{9}{16}$	55	986
$\frac{5}{8}$	49·5	719
$\frac{3}{4}$	41·2	416
$\frac{7}{8}$	35·3	262
1	30·9	175
$1\frac{1}{8}$	27·5	123
$1\frac{1}{4}$	24·7	89·8
$1\frac{3}{8}$	22·5	67·5
$1\frac{1}{2}$	20·6	52·0
$1\frac{3}{4}$	17·7	32·7
2	15·5	21·9
$2\frac{1}{4}$	13·7	15·4
$2\frac{1}{2}$	12·4	11·2

11 Gauge—0·116 in. (= $\frac{7}{64}$ in.+) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
$\frac{1}{2}$	86	2,173
$\frac{9}{16}$	76	1,526
$\frac{5}{8}$	69	1,113
$\frac{3}{4}$	57	644
$\frac{7}{8}$	49·1	405
1	42·0	272
$1\frac{1}{8}$	38·2	191
$1\frac{1}{4}$	34·3	139
$1\frac{3}{8}$	31·2	104
$1\frac{1}{2}$	28·6	80·5
$1\frac{3}{4}$	24·5	50·7
2	21·5	33·9
$2\frac{1}{4}$	19·1	23·8
$2\frac{1}{2}$	17·2	17·4
3	14·3	10·1

10 Gauge—0·128 in. (= $\frac{1}{8}$ in.) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
$\frac{1}{2}$	115	3,221
$\frac{9}{16}$	102	2,262
$\frac{5}{8}$	92	1,649
$\frac{3}{4}$	77	954
$\frac{7}{8}$	66	601
1	58	403
$1\frac{1}{8}$	51	283
$1\frac{1}{4}$	46·1	206
$1\frac{3}{8}$	41·9	155
$1\frac{1}{2}$	38·4	119
$1\frac{3}{4}$	32·9	75·1
2	28·8	50·3
$2\frac{1}{4}$	25·6	35·3
$2\frac{1}{2}$	23·1	25·8
3	19·1	14·9

9 Gauge—0·144 in. (= $\frac{9}{64}$ in.+) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
$\frac{1}{2}$	164	5,160
$\frac{9}{16}$	146	3,623
$\frac{5}{8}$	131	2,842
$\frac{3}{4}$	109	1,529
$\frac{7}{8}$	94	963
1	82	645
$1\frac{1}{8}$	73	453
$1\frac{1}{4}$	66	330
$1\frac{3}{8}$	60	248
$1\frac{1}{2}$	53	191
$1\frac{3}{4}$	47·0	120
2	41·0	80·6
$2\frac{1}{4}$	36·5	56·6
$2\frac{1}{2}$	32·8	41·3
3	27·3	23·9

STEEL HELICAL SPRINGS—ROUND WIRE—*continued.*

8 Gauge—0·16 in. (=5/32 in.+) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
1/2	225	7,863
9/16	200	5,521
5/8	180	4,027
3/4	150	2,330
7/8	129	1,467
1	113	983
1 1/8	100	690
1 1/4	90	503
1 3/8	82	378
1 1/2	75	291
1 3/4	64	183
2	56	123
2 1/4	50	86·2
2 1/2	45	62·9
3	37·5	36·4

7 Gauge—0·176 in. (=11/64 in.+) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
5/8	240	5,896
3/4	200	3,412
7/8	171	2,148
1	150	1,439
1 1/8	133	1,011
1 1/4	120	737
1 3/8	109	554
1 1/2	100	426
1 3/4	86	268
2	75	180
2 1/4	66	126
2 1/2	60	92·1
3	50	53·3
3 1/2	42·8	33·6
4	37·5	22·5

6 Gauge—0·192 in. (=3/16 in.+) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
5/8	311	8,351
3/4	259	4,833
7/8	222	3,043
1	195	2,038
1 1/8	173	1,432
1 1/4	156	1,044
1 3/8	142	784
1 1/2	130	604
1 3/4	111	380
2	97	255
2 1/4	86	179
2 1/2	78	130
3	65	75·5
3 1/2	56	47·5
4	48·6	31·8

5 Gauge—0·212 in. (7/32 in.+) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
5/8	419	12,413
3/4	349	7,183
7/8	300	4,523
1	262	3,030
1 1/8	233	2,128
1 1/4	210	1,551
1 3/8	191	1,165
1 1/2	175	898
1 3/4	150	565
2	131	379
2 1/4	116	266
2 1/2	105	194
3	87	112
3 1/2	75	70·7
4	65	47·3

4 Gauge—0·232 in. (=15/64 in.) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
5/8	549	17,801
3/4	458	10,302
7/8	392	6,487
1	343	4,345
1 1/8	305	3,052
1	275	2,225
1 3/8	250	1,675
1 1/2	229	1,287
1 3/4	196	813
2	172	543
2 1/4	152	381
2 1/2	137	278
3	114	161
3 1/2	98	101

3 Gauge—0·252 in. (=1/4 in.) diam. Safe Stress—70,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
3/4	587	14,341
7/8	503	9,030
1	440	6,049
1 1/8	391	4,249
1 1/4	352	3,097
1 3/8	320	2,327
1 1/2	293	1,792
1 3/4	251	1,129
2	220	756
2 1/4	195	531
2 1/2	176	387
3	147	224
3 1/2	126	141
4	110	94·5

2 Gauge—0·276 in. (=9/32 in.—) diam. Safe Stress—67,900 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
3/4	746	20,636
7/8	639	12,993
1	559	8,704
1 1/8	497	6,113
1 1/4	447	4,457
1 3/8	407	3,343
1 1/2	373	2,579
1 3/4	319	1,624
2	279	1,088
2 1/4	248	764
2 1/2	223	557
3	186	322
3 1/2	160	203
4	140	136

1 Gauge—0·3 in. (=19/64 in.+) diam. Safe Stress—66,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
7/8	799	18,137
1	699	12,150
1 1/8	622	8,534
1 1/4	559	6,221
1 3/8	509	4,674
1 1/2	466	3,600
1 3/4	400	2,267
2	349	1,519
2 1/4	310	1,067
2 1/2	279	778
3	233	450
3 1/2	200	283
4	175	190
4 1/2	155	133

0 Gauge—0·324 in. (=21/64 in.—) diam. Safe Stress—64,100 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
1	860	16,530
1 1/8	765	11,610
1 1/4	688	8,463
1 3/8	626	6,359
1 1/2	574	4,898
1 3/4	492	3,084
2	430	2,063
2 1/4	382	1,451
2 1/2	344	1,058
3	287	612
3 1/2	246	386
4	215	258
4 1/2	191	181
5	172	132
5 1/2	156	99

00 Gauge—0·348 in. (=11/32 in.+) diam. Safe Stress—62,200 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
1	1,033	22,000
1 1/8	918	15,451
1 1/4	826	11,264
1 3/8	751	8,462
1 1/2	688	6,518
1 3/4	590	4,105
2	516	2,750
2 1/4	459	1,931
2 1/2	413	1,408
3	344	815
3 1/2	295	513
4	258	344
4 1/2	229	241
5	206	176
5 1/2	188	132

000 Gauge—0·372 in. (=3/8 in.) diam. Safe Stress—60,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
1	1,215	28,725
1 1/8	1,080	20,175
1 1/4	972	14,721
1 3/8	883	11,050
1 1/2	810	8,511
1 3/4	694	5,360
2	607	3,591
2 1/4	540	2,522
2 1/2	486	1,839
3	405	1,064
3 1/2	347	670
4	304	449
4 1/2	270	315
5	243	230
5 1/2	221	173

0000 Gauge—0·4 in. (=13/32 in.—) diam. Safe Stress—58,000 lbs. per sq. in.

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
1	1,297	26,970
1 1/8	1,167	19,661
1 1/4	1,061	14,772
1 1/2	973	11,378
1 3/4	834	7,165
2	730	4,800
2 1/4	648	3,371
2 1/2	584	2,458
3	486	1,422
3 1/2	417	896
4	365	600
4 1/2	324	421
5	292	307
5 1/2	265	231
6	243	178

STEEL HELICAL SPRINGS—ROUND WIRE—*continued.*

Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.	Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.	Mean Diam. of Coil. Ins.	Safe Load. Lbs.	Stiffness. Lbs. per In.
00000 Gauge—0·432 in. (= $\frac{7}{16}$ in.—) diam. Safe Stress—55,400 lbs. per sq. in.			000000 Gauge—0·464 in. (= $\frac{15}{32}$ in.—) diam. Safe Stress—53,700 lbs. per sq. in.			0000000 Gauge—0·5 in. (= $\frac{1}{2}$ in.) diam. Safe Stress—50,000 lbs. per sq. in.		
$1\frac{1}{8}$	1,559	36,692	$1\frac{1}{4}$	1,686	35,599	$1\frac{3}{8}$	1,782	36,063
$1\frac{1}{4}$	1,403	26,749	$1\frac{3}{8}$	1,533	26,746	$1\frac{1}{2}$	1,634	27,784
$1\frac{3}{8}$	1,275	20,096	$1\frac{1}{2}$	1,405	20,601	$1\frac{3}{4}$	1,400	17,493
$1\frac{1}{2}$	1,169	15,479	$1\frac{3}{4}$	1,204	12,973	2	1,225	11,179
$1\frac{3}{4}$	1,002	9,748	2	1,054	8,691	$2\frac{1}{4}$	1,089	8,230
2	877	6,530	$2\frac{1}{4}$	937	6,104	$2\frac{1}{2}$	980	6,000
$2\frac{1}{4}$	779	4,586	$2\frac{1}{2}$	843	4,450	3	817	3,472
$2\frac{1}{2}$	701	3,344	3	703	2,575	$3\frac{1}{2}$	700	2,187
3	584	1,935	$3\frac{1}{2}$	602	1,622	4	612	1,465
$3\frac{1}{2}$	501	1,218	4	527	1,086	$4\frac{1}{2}$	545	1,029
4	438	816	$4\frac{1}{2}$	468	763	5	490	750
$4\frac{1}{2}$	390	573	5	422	556	$5\frac{1}{2}$	445	564
5	351	418	$5\frac{1}{2}$	383	418	6	408	434
$5\frac{1}{2}$	319	314	6	351	322	7	350	273
6	290	242	7	301	203	8	306	183
			8	263	136	9	272	129

Multiplying Constants.

1. For compression springs with little variation of load multiply *Safe Loads* by 1·25.
2. For Springs, especially tension springs subjected to considerable variation of load suddenly applied, multiply *Safe Loads* by 0·67.
3. For Square Wire Springs where the Side equals the diameter given in the table—multiply

Safe Load by 1·06
Stiffness by 1·5

4. For Brass Wire Springs (round) multiply the Stiffness by 0·42. (Only for *Hard* Brass.)

Note.—In getting out Compression Springs allow for the fact that one coil at each end is idle. These two coils do not affect the Stiffness, but do affect the closed length.

VI.—LAMINATED SPRINGS.

All leaves graduated.

Type of Spring.	Fibre Stress in Lb. per Sq. In. f_t.	Load in Lb. W.	Deflection in Ins. δ.
Quarter Elliptic (cantilever) .	$\frac{6WL}{nbt^2}$	$\frac{f_t nbt^2}{6L}$	$\frac{6WL^3}{Enbt^3}$ or $\frac{f_t L^2}{Et}$
Semi-elliptic	$\frac{3WL}{2nbt^2}$	$\frac{2f_t nbt^2}{3L}$	$\frac{3WL^3}{8Enbt^3}$ or $\frac{f_t L^2}{4Et}$
Full elliptic	$\frac{3WL}{2nbt^2}$	$\frac{2f_t nbt^2}{3L}$	$\frac{3WL^3}{4Enbt^3}$ or $\frac{f_t L^2}{2Et}$

Where,

W = load in lb.;
L = span (or length of cantilever) in inches;
E = Young's modulus of elasticity in lb. per sq. in. (= 29×10^6 for steel);
n = number of leaves;
b = breadth of leaves in inches;
t = thickness of leaves in inches;
f_t = fibre stress in lb. per sq. in.

The above formulæ assume that all the leaves are of the same thickness and no allowance is made for friction between the leaves. Modifications are introduced where one or more leaves are thickened, where the thickness is graduated, or where initial stress in the leaves is caused by 'nip' (see under 'Nip').

For heat-treated and tempered spring plates the safe fibre stress may be taken as 30,000 lb. per sq. in. It has been found that most heat-treated and tempered spring plates have a decarburised surface layer which reduces the resistance to repeated stresses. If the leaves be machined after treatment or be heat-treated in such a way as to reduce the decarburising effect to a minimum then the safe fibre stress may be taken as 70,000 to 80,000 lb. per sq. in.

Nip.—Nip is a manufacturing practice according to which the individual leaves of a spring before assembly are given different curvature so that when the leaves are clamped together, generally by means of a bolt, they are all pulled to approximately the same curvature and initial stress is given to certain of the leaves. By carrying out such a procedure the stresses in the spring leaves of the unloaded spring may be considerable, and also may be of opposite sign to that produced by loading. The advantage of nip if correctly adjusted is to give the master leaf of the spring a reduced mean stress when under load; by this means the master leaf is in a more favourable condition to resist the complex stresses imposed upon it.

The stress in lb. per sq. in. in the master leaf (the longest leaf) due to nip

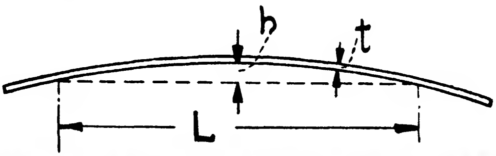
$$= \left(\frac{1}{R_1} - \frac{1}{R}\right)\frac{Et}{2}.$$

where,

R_1 = radius of curvature of leaf before assembly in inches;
R = radius of curvature of leaf after assembly in inches;
t = thickness of leaf in inches;
E = Young's modulus of elasticity in lb. per sq. in. (= 29×10^6 for steel).

R and R_1 can be calculated by measuring the height 'h' in inches on a span of 'L' inches as shown in the figure),

then $\dfrac{1}{R}$ or $\dfrac{1}{R_1} = \dfrac{8h}{L^2}$.



The amount of 'nip' should be small, and, if given, should be between the three longer leaves.

Camber.—For smooth riding the camber should be such that the spring is flat when under load.

Leaf Ends.—The tapering of the end of each leaf should be carried back to the next shorter leaf and should be such that the deflection of the end or overlap is 1½ times that of the untapered overlap. This can be done by making the end spear-shaped in plan or a semi-cubic parabola in elevation. The latter is achieved if the cube of the thickness is proportional to the distance from the tip.

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SECTION XXI

PART I

FACTORY PLANNING AND LAY-OUT

(pp. 905–908)

PART II

BELTING — ROPE DRIVING — CLUTCHES

(Revised by H. W. Swift, D.Sc., M.I.Mech.E.)

(pp. 909–932)

PART III

WIRE ROPES FOR ENGINEERING AND MINING PURPOSES

(Contributed by Norman Whincup.)

(pp. 933–959.)

PART IV

CHAIN DRIVING

(pp. 961–966)

PART V

SPUR-GEARING — BEVEL GEARING — HELICAL GEARING TURBINE GEARS — WORM GEARING

(Contributed by H. T. Davey, M.I.Mech.E.)

(pp. 967–1015

SECTION XXI

PART I

FACTORY PLANNING AND LAY-OUT—CHOICE OF POWER TRANSMISSION TO MACHINES.

The importance and advantage of advanced planning for works and factories is now more generally recognised than formerly, and experience has shown that a factory which has been designed and planned on lines suitable for a given class of manufacture or process, always proves more satisfactory and economical than a factory where advanced planning has been neglected.

While no rigid rule or formula can be devised to govern planning and lay-out for all forms of manufacturing process, there are, nevertheless, certain aspects of the problem which are common to every case, and there are broad principles of planning that can be applied whenever new, or modernisation of existing, factories are contemplated.

The one law that should always be remembered when designing a new factory is that it is always the 'class' of manufacture or process that governs the type, and 'quantity' that governs the size of the factory required.

Some General Rules to be Considered when Planning the Lay-out of Works and Factories.

The total area of factory is governed by the maximum output required.

Allocating Space for the Various Departments.

This is one of the most important points to be decided when planning the general lay-out. The main aim should be to plan a well-balanced factory, wherein each separate department or each separate process of manufacture can always keep in step with all other departments and never be found lagging behind, nor yet getting to far ahead with its part of the work.

As a guide to the allocation of space that should be allowed for the various departments in a modern engineering works, the author gives the following particulars of a works for which he was responsible for the whole arrangement and lay-out.

The works were designed for the production of medium heavy engineering products, all the departments, including the foundry, were under one roof, so that straight-line production was adopted without difficulty.

The main building was 500 ft. long × 100 ft. wide giving a total area of 50,000 sq. ft. The area allocated to each department being as follows :—

Department.	Space in Sq. Ft.	Percentage of Total Area.
Foundry	12,800	25·6
Rough Material Stores	3,400	6·8
Machine Shop	12,600	25·2
Inspection Department	1,100	2·2
Tool Room	1,600	3·2
Finished Parts Stores	3,200	6·4
Tool Stores	500	1
Assembling and Erecting Shop . . .	8,400	16·8
Testing Department	2,400	4·8
Warehouse, Packing and Despatch . .	4,000	8
Total	50,000	100·0
Pattern Shop and Stores under separate roof .	5,500	9

Note.—All forgings and stampings were bought in and delivered by rail, the rough material stores and the foundry being connected to the railway siding, so that handling of material was reduced to a minimum. Pattern shop and stores were under a separate roof adjoining the foundry. Works administration offices—planing, rate-fixing, etc.—were housed in an annex.

These works have been in operation upwards of five years, and the space allocated to the several departments has proved quite satisfactory, and may be accepted as an example of the relative space required for medium heavy engineering work.

There are three methods that may be used to determine the space required for each department in any factory.

1. By exact time study of every process and of every operation involved in the process.
2. Drawing on experience and judgment.
3. Using a combination of time study and experience.

In practice the third method proves the most reliable of all.

But a point to remember is that a factory should never be planned on too rigid lines where it can only run smoothly in a given course. The lay-out engineer should always make his plans sufficiently elastic to cover contingencies and to allow for the fluctuations and variations in the flow of work, to which every factory is liable.

The rate at which work must flow through each department is necessarily governed by the maximum output for which the factory has been designed. A time study of all the operations in the various processes will dictate the number of units and size of the plant required, and experience will be the guide to show what allowances must be made for contingencies throughout the whole ambit of manufacture.

Sequence or Order in which Departments are Arranged.

Whenever possible departments should be arranged in the order that corresponds to the sequence of process through which the material passes in the course of manufacture. In an engineering works the pattern shop and pattern stores will be adjacent to the foundry, the rough stores will lie between the machine shop, foundry and forge, while the finished parts stores should follow the machine shop, and so on. The aim being to keep the flow of work in one direction, and always towards the despatch end of the factory.

The position in which the tool and jig stores are located depends mainly on the system of distribution adopted, the most natural position is in close proximity to the inspection room, so that the working gauges can be easily checked with the master gauges used by inspectors and gaugers.

Lay-out of the Machine Shop.

Before commencing the lay-out of the machine shop it is necessary to decide the system on which machine work is to be dealt with. Broadly speaking the choice lies between two well defined systems.

1. The group system, where a group of similar type machines are placed together, forming for instance a planing, milling or drilling group.

2. Sequence of operation system, where the machines are placed to deal with the work in sequence of operation.

In practice the group system is the more elastic, and lends itself to advantage in dealing with a large variety of work. It has the disadvantage of sometimes causing the work to pass through a circuitous course before the last operation is completed and the finished stores reached.

The sequence of operation system is more rigid and can only be adopted where the machine parts are specific and vary but little in character over a long period. It also calls for a close time study of all the operations in every part of the manufactured product. But where the conditions are favourable this system has enormous advantages. It cuts transportation costs to a minimum. Every piece of work passes from one machine to another in proper sequence of operation until completed. Thus a prearranged routine for every piece of work is firmly established without the aid of what are called 'router' or 'chaser' men.

Choice of Power Transmission to Machines.

The method to be adopted for driving the machine plant should be decided before the lay-out of the machine shop is commenced, as the type of drive selected will affect the position in which the machines can be placed.

Broadly speaking, the choice of power transmission is confined to three systems.

1. Line-shaft drive.
2. Group drive.
3. Individual motor drive.

All three systems are now extensively used. In certain cases a combination of these systems can be used to advantage.

The direct-motor drive offers many advantages when applied to heavy machines, such as large planers, shapers, millers, grinding and drilling machines, while for multi-head machines the direct-motor drive is practically indispensable.

When determining the type of drive to be adopted the following considerations should be taken into account.

Line-Shaft Drive.—Main line-shaft driving is the oldest system of all, and in the modern sense was the first form of power transmission to machines ever used. It remained standard practice for many years, but the extreme length to which many lines of shafts were carried, and the deplorable lack of alignment often found, resulted in heavy frictional losses.

But given proper alignment, both initial and permanent, there are still some advantages in this type of drive.

The first cost of line-shaft drive is lower than any other form yet devised, and when run in well supported, properly designed bearings of the ball, roller, or plain self-oiling types, the frictional losses are not necessarily high.

In good modern practice the length of shaft lines are limited to 100 to 120 ft., with bearings spaced 10 to 12 ft. apart, and driving, say, 10 to 12 medium-sized machines. The power drive is by motor through roller chain and sprocket wheels, or through vee pulleys and belts. The motor should be supported overhead, and have a minimum of 3 ft. centres between the motor and line shaft, the drive being placed near the middle of the shaft and as close as possible to one of the bearings.

There are various factors governing the power of the motor required for driving all the machines from the shaft, and investigation into the probable use of the machines is necessary. For instance, if the machines are to be used for the usual purpose of roughing and finishing work, the maximum load of each machine will be intermittent, and taking into account the stopping time for setting and gauging, it follows that the maximum demand from the driving motor never equals the total power of all the machines driven from the shaft.

Experience shows that a simple formula may be used for finding the power of the motor required for driving a line shaft, or a group of machines having a known maximum power demand,

Where M = permanent safe load of motor;
T = total maximum power required by all the machines in the group;
c = a constant;

then $$M = \frac{T}{c}.$$

In the above formula the value of c varies according to the number of machines being driven.

Thus, for 12 machines $c = 4$.
„ 8 „ $c = 3$
„ 6 „ $c = 2{\cdot}5$
„ 4 „ $c = 2$.

Taking a line shaft driving twelve machines whose aggregate rated power is 160 h.p.

Then $M = \frac{T}{c} = \frac{160}{4} = 40$ h.p. power of motor required.

Note.—The cost of one 40 h.p. motor with switch and starter plus the cost of shafting, bearings and pulleys, is considerably less than that of twelve small motors with switches and starters, plus the extra wiring involved; also maintenance and running charges will be less. A large motor running on full load is more efficient than small motors running on light loads. But it is not economical to run a large motor and line shaft in an unbalanced factory, where only one or two machines are in constant use, or where only a few of the machines are required to run for long periods on overtime or night-shift work.

Group Drive.—A system that has come to be known as the group drive for machines, belongs to the same catagory as the line-shaft drive, but generally there are fewer machines in the group than in the line drive. Its advantage lies in the fact that a convenient number of machines may be placed in any part of the factory, and driven as a group from one motor.

The group drive makes an alternative choice between the common line-shaft drive and the more expensive individual motor drive. A preponderance of machine tools are now made with the all-gear head and single pulley drive, and this renders group driving both convenient and economical. There is no necessity for any overhead shafting, pulleys and belts which are sometimes found to interfere with the lighting system, and form an obstruction to a clear view throughout the shop.

Another advantage of the group drive is that a small number of machines can be run on overtime and night shift, should that become necessary, without undue loss, and even when only one or two machines in the group are required to run, the frictional and current losses are relatively small, as only a very short line shaft and medium powered motor have to be run for power transmission.

The outline, fig. 1 (p. 908) gives a convenient and inexpensive lay-out for a group drive, the arrangement being suitable for 4, 6 or 8 machines. The figure shows four machines with the motor placed at the end of the shaft, when other machines are added to the group they should be placed at the motor end, and the shaft extended accordingly.

The advantages of this arrangement are, there is an entire absence of overhead transmission gear, and therefore no interference with light or obstruction to view. In this respect it has all the advantages of the individual motor drive, while the cost of installation and running are found to be appreciably lower.

In factories where the general machines are driven in groups or from overhead line shafts, it is good practice to standardise the power of the driving motors, as this allows a minimum of spare parts to be stocked.

Individual Motor Drive.—With this form of drive each machine becomes a separate and independent unit, it can be placed in any convenient part of the factory and be run at any time without affecting any other part of the plant. Its advantages are most pronounced when applied to large machines and to machines of the multi-head type.

Machine tools fitted with this form of power transmission have a relatively high first cost, and unless special precautions are taken the maintenance and depreciation charges also tend to become high.

The power of the motor to be fitted for individual drive requires special consideration, and the nature of the work on which the machine is to be engaged should be taken into account. If the machine is to be used for roughing out the work with heavy cuts and coarse feeds, care must be

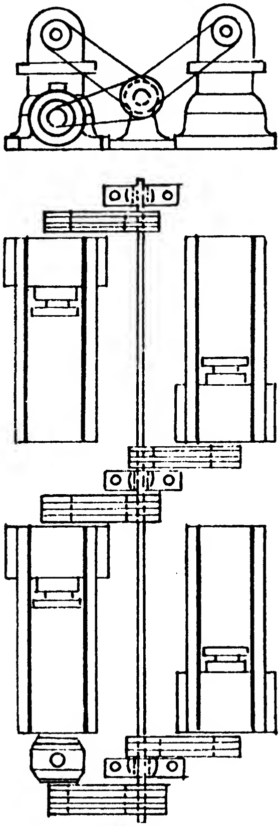


FIG. 1.

taken to ascertain the maximum power the machine requires for these conditions, and to fit a motor of sufficient power to meet this demand, otherwise overloading and breakdown are likely to occur. On the other hand, if the machine is only required for light duty it is a mistake to fit a high powered motor, because on light loads the efficiency of the motor is low. This is particularly the case with small motors and machines.

There are three separate applications of individua lmotor drive :—

1. Belt drive from motor to machine.
2. Direct coupled motor to machine.
3. Built-in motor with armature mounted on main spindle of machine.

The second and third methods are the more compact, but for convenience of cleaning, repair and replacement, the first method is of greatest advantage.

SECTION XXI

PART II

BELTING—ROPE DRIVING—CLUTCHES.

(Revised and amplified by H. W. Swift., D.Sc., M.I.Mech.E.)

Belts and Belt Drives.

In a general way belting may be classified under three headings:—

1. Leather; oak-tanned, mineral tanned, raw-hide, laminated.
2. Solid woven; cotton, hair, 'tar' dressed, oil dressed, rubber impregnated.
3. Canvas; friction surface rubber, balata.

1. Leather.

Oak-tanned leather belting is pre-eminently a general utility belting; there are few applications for which it is definitely unsuitable. When properly made from properly selected material it is durable, flexible, reasonably stretchless within its economic range, suitable for holding fasteners, for endless construction and for fork action. Standards for the selection of raw material and methods of manufacture are laid down in B.S.S. 424—1931 together with test requirements

Mineral-tanned (usually chrome) leather makes a durable, flexible and strong belt with power to withstand damp conditions when the spliced joints are made with waterproof cement. When suitably prepared chrome leather has better frictional properties than oak-tanned leather but some types have a greater tendency to stretch. Certain black-dressed types have the best frictional properties of any known belting, while the adhesion of the typical grey finished belt is relatively low.

Raw-hide belting has a compact structure and good flexibility and is specially suitable for small pulley service It is durable, strong and relatively stretchless and for transmission calculation may be regarded as equivalent to an oak-tanned belt of 20 per cent. greater thickness.

Laminated belting by virtue of its construction is strong but heavy. It can be made endless on site. Its special field of usefulness is for heavy, slow-speed drives, particularly where metal fasteners cannot be used.

2. Solid Woven Belts.

Solid woven belting consists essentially of a number of fabric plies bound together in the process of weaving. The main warp according as it is cotton or hair, gives its name to the type of belting. The weft and binding yarn are usually of cotton for reasons of strength.

Solid woven belts can now be manufactured endless. The impregnation employed affects the life and flexibility of the belt and its susceptibility to changing atmospheric conditions as well as its adhesion to the pulleys. Good bituminous dressings which do not contain constituents harmful to cotton fibres protect them against atmospheric changes and internal abrasion and provide excellent adhesive properties but are not specially conducive to flexibility. Oil dressings (of which 'red' dressing is typical) give less stretch and improved fastener holding properties but lower adhesion. They are more frequently used for hair belts than for cotton belts.

Woven Cotton belts are strong and cheap. The bituminous dressed types are eminently suited for steady loads under reasonably constant atmospheric conditions. On the other hand they have not the flexibility and resilience of leather and hair belts.

Woven Hair belts are more suited for changeable atmospheric conditions and are more resistant to acid fumes. They have greater resilience and are more satisfactory under shock or cyclic variations of load. On the other hand they are not suitable for use with claw-type fasteners.

Rubber-impregnated Woven belting is produced by impregnating the solid woven belting with rubber by a special process either during or after manufacture. In this way the fibres are to a large extent encased and cushioned with rubber. After vulcanisation a belt is produced with low stretch, good fastener-holding capacity, frictional properties and flexibility. In most cases it is insensitive to changes in atmospheric conditions. This relatively new type of belting is specially suitable for small pulley service and short centre drives.

3 Canvas Belts.

Canvas belting consists of plies of canvas or duck either folded or superposed and bound together by either rubber or balata gum. Both types have good mechanical strength and good fastener-holding capacity Their intrinsic friction is moderate; it may be improved by suitable dressing but since the belt itself, being impervious to dressing, is incapable of acting as a reservoir frequent applications are necessary.

Frictioned Surface transmission belting (rubber frictioned) is useful for general purposes. It resists wet and heat, and has fair elasticity. The folded edge of the normal construction is not conducive to flexibility, but the straight-edged type though not so pleasing to the eye is more flexible and not so subject to opening at the plies. Rubber belting deteriorates rapidly in the presence of the lighter mineral oils.

Balata belting is proof against damp but not against high temperatures. It is not very resilient; otherwise its general properties are similar to those of folded rubber-frictioned belting.

The tensile strength of belting material varies over a wide range as shown in the table :—

Type of Belting.	Tensile Strength. Lbs. per sq. in.
Leather (oak-tanned)	3,000– 6,000
Leather (mineral-tanned)	4,000– 7,000
Solid woven cotton	8,000–12,000
Solid woven hair	4,000– 5,500
Rubber (frictioned)	6,000– 8,000
Balata	6,500– 9,000

The British Standard Specification 424—1931 requires a minimum strength of 2,500 lbs. per sq. in. for splices of oak-tanned leather belting.

Horse-Power Ratings.

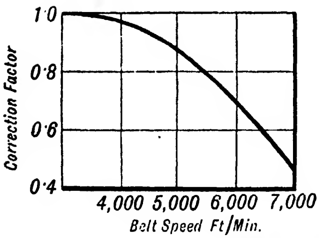
The horse-power transmitted by a belt per inch of width is given by the formula :—

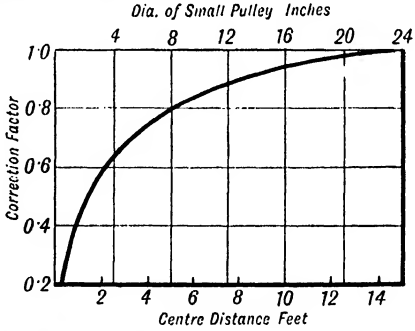
$$H.P. = \frac{T.V.}{33,000}$$

where T is the effective tension ($T_1 - T_2$) in pounds per inch of width and V is the belt speed in feet per minute.

The belt speed and horse-power capacity may conveniently be determined from a chart of the type shown in Chart A, fig. 3.

On this chart any given pulley diameter D and speed N determine the relevant speed curve. This curve cuts the V scale at a point giving the belt speed in feet per minute.





Figs. 1 and 2.—Correction Factors.

To find the horse-power per inch of width : join the origin O to the point at which the relevant speed curve cuts either H.P. scale (chosen for convenience), produce to the tension abscissa corresponding to the working effective tension and read back the H.P. per inch on the selected H.P. scale.

On the chart an example is taken in which a pulley of effective diameter 10 ins. runs at 1,000 r.p.m. The belt speed V is shown to be 2,650 ft. per minute and the corresponding capacity 5·5 H.P. per inch of width.

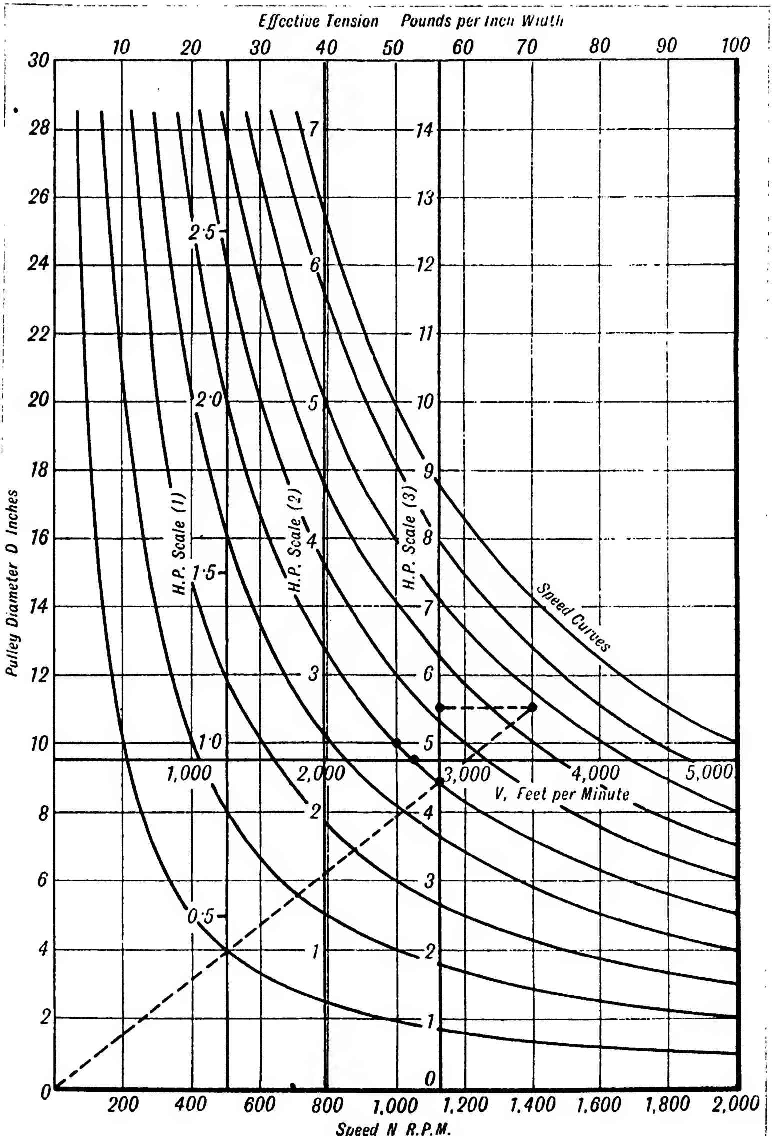
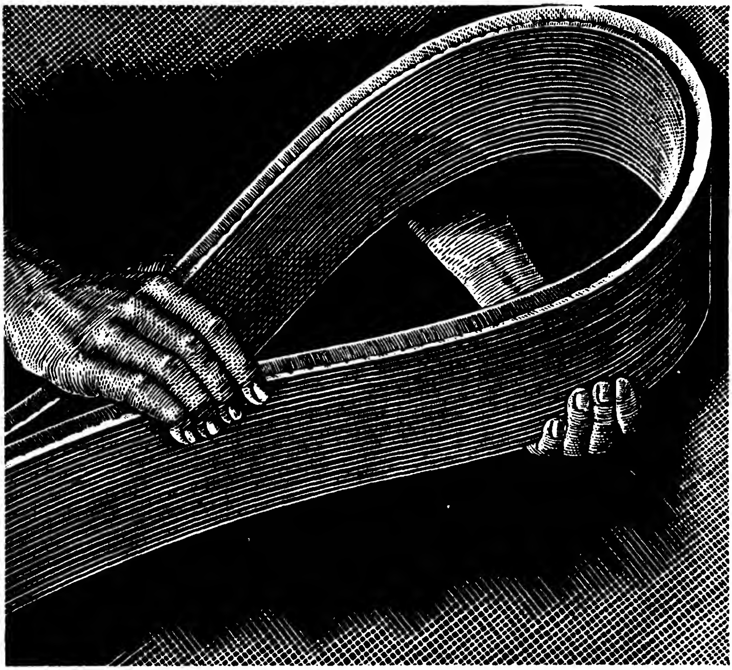

Effective Tension Pounds per Inch Width
10 20 30 40 50 60 70 80 90 100
Pulley Diameter D Inches
30 28 26 24 22 20 18 16 14 12 10 8 6 4 2 0
H.P. Scale (1)
2·5 2·0 1·5 1·0 0·5
H.P. Scale (2)
7 6 5 4 3 2 1
H.P. Scale (3)
14 13 12 11 10 9 8 7 6 5 4 3 2 1 0
Speed Curves
1,000 2,000 3,000 4,000 5,000
V, Feet per Minute
200 400 600 800 1,000 1,200 1,400 1,600 1,800 2,000
Speed N R.P.M.

Fig. 3.—Chart A.

The effective tension is best determined from a basic value depending on the type of belt and corrected for belt speed, pulley diameter and length of drive. Suitable basic values of the effective tension are given in the table, together with the smallest diameter of pulley for which the belt is suitable. Correction factors are shown graphically in fig. 1 and 2.

Type of Belt.	Thickness or Ply.	Basic Effective Tension. Lbs. per in.	Minimum Pulley Diameter, ins.
Leather oak-tanned .	4 mm. single	40	$2\frac{1}{2}$
„ .	5 mm. „	50	3
„ .	6 mm. „	60	$3\frac{1}{2}$
„ .	7 mm. „	70	4
„ .	8 mm. double	80	8
„ .	10 mm. „	90	10
„ .	12 mm. „	100	12
„	14 mm. „	110	14
Leather, chrome-tanned .	Add 20–40 per cent. to effective tensions above according to frictional properties, and add 40 per cent. to minimum pulley diameters.		
Leather, raw-hide . .	Add 20 per cent. to effective tensions and subtract 20 per cent. from minimum pulley diameters.		
Solid woven cotton bituminous) . . .	$\frac{3}{16}$ in.	60	$7\frac{1}{2}$
„ . . .	$\frac{1}{4}$ „	80	10
„ . . .	$\frac{5}{16}$ „	100	$12\frac{1}{2}$
„ . . .	$\frac{3}{8}$ „	120	15
(rubber impregnated) .	$\frac{3}{16}$ „	60	4
„ . .	$\frac{1}{4}$ „	80	6
„ . .	$\frac{5}{16}$ „	100	9
„ . .	$\frac{3}{8}$ „	120	12
(rubber impregnated and made endless) . .	$\frac{1}{16}$ „	30	1
„ . .	$\frac{1}{8}$ „	50	2
„ . .	$\frac{3}{16}$ „	70	3
„ . .	$\frac{1}{4}$ „	90	4
Solid woven hair (bituminous) . .	$\frac{1}{4}$ „	70	$7\frac{1}{2}$
„ . .	$\frac{5}{16}$ „	90	$9\frac{1}{2}$
„ . .	$\frac{3}{8}$ „	100	11
(rubber impregnated) .	$\frac{1}{4}$ „	80	6
„ . .	$\frac{5}{16}$ „	100	8
„ . .	$\frac{3}{8}$ „	110	10
Balata or friction surface rubber (folded) . .	3-ply	40	$7\frac{1}{2}$
„ . .	4 „	60	9
„ . .	5 „	70	10
„ . .	6 „	80	$12\frac{1}{2}$
„ . .	7 „	100	14
„ . .	8 „	110	15
„ . .	9 „	120	16
„ . .	10 „	130	18
Friction-surface rubber (straight-edged) . .	3 „	40	5
„ . .	4 „	60	$5\frac{1}{2}$
„ . .	5 „	70	$7\frac{1}{2}$
„ . .	6 „	80	$9\frac{1}{2}$
„ .	7 „	100	$10\frac{1}{2}$

The correction factors shown in the figures do not take abnormal conditions into account, but are such as should give a reasonable belt life under normal conditions. It is good practice for example to allow 15–25 per cent. additional width for vertical drives, the allowance increasing

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with the length of drive, and 10 per cent. for horizontal drives with the slack side below. Wet or oily conditions, inaccessible positions and variable loads also require more generous belting. In the case of twist or cross drives excessive width of belt is undesirable and additional thickness may be required.

Method of Design.

When designing a belt drive, first decide what maximum H.P. will have to be transmitted by the belt. This maximum H.P. should include peak loads or high starting loads. It may be necessary to increase the nominal H.P. by 30 per cent. to 50 per cent. for disintegrators or pumps where the load is intermittent or suddenly applied.

Examples.—What belt will be required to drive a 20-in. diameter driven pulley from a 6-in. diameter driving pulley at 10 ft. horizontal centres? Belt speed 3,800 f.p.m. Max. H.P. = 34. Slack side of belt on top.

A ½-inch endless woven, rubber impregnated cotton belt could be chosen as joint trouble at the rather high speed on a 6-in. pulley would be eliminated. From the tables:—

$$\begin{array}{rl} & \text{Basic} \quad \begin{matrix}\text{C.F.}\\ \text{Speed.}\end{matrix} \quad \begin{matrix}\text{C.F.}\\ \text{Centres.}\end{matrix} \quad \begin{matrix}\text{C.F.}\\ \text{Pulley.}\end{matrix} \\ \text{Effective tension } T = & 90 \times 0{\cdot}98 \times 0{\cdot}95 \times 0{\cdot}74 \\ = & 62 \text{ lb. per in. width.} \end{array}$$

$$\text{H.P. per in width} = \frac{62 \times 3{,}800}{33{,}000} = 7{\cdot}14.$$

$$\text{Width of belt} = \frac{34}{7{\cdot}14} = 4{\cdot}76 \text{ ins.} \quad \text{Hence } \textit{5-in. belt required.}$$

Selection of Belt and Fastener.

Endless belts are preferred where practicable, particularly at high speeds. If it is not possible to mount an endless belt the question of a suitable joint is of importance.

For average drive conditions on pulleys 10 ins. diameter and upwards, solid woven belts give efficient service. They grip well and for rather light applications they can be joined with wire hook lacing of the clipper type or alligator type fastening. Where pulleys are not less than 12 ins. diameter, plate fasteners can be relied upon.

For general machine shop work, etc., where pulleys less than 10 ins. diameter are often encountered, either leather, rubber impregnated solid woven, friction-surface, rubber or balata may be used, fastened with claw type fasteners. Rubber impregnated belts are useful where small amounts of lubricating oil are present. Where water-soluble cutting compound is present, high friction black chrome leather works well and if the pulleys are very small rawhide leather is best.

For damp, exposed drives, solid woven hair belts, friction-surface rubber or balata may be used. Solid woven cotton is specially suitable for high temperatures but oak tanned leather or balata should not be used if the temperature exceeds 100° F.

For motor and dynamo drives where endless belts cannot be mounted, hair belts, rubber impregnated cotton belts or leather belts give good, lasting service. A good joint for such applications is the Lagrelle turn-up joint. This is especially useful if the belt speed exceeds 2,500 f.p.m.

Pulley Camber.

The purpose of pulley camber is to cause the belt to track properly in spite of errors of alignment (obliquity and twist) of the shafts and want of truth in the belt itself. Since the effect of camber is to cause differences of strain between the various fibres of the belt it is important that no greater camber should be used than is necessary for its purpose, and the better the millwrighting and the belting the less camber is required.

The effectiveness of pulley camber depends on the ratio of the height of camber to the diameter of the pulley; it is practically independent of the width of the belt. It increases somewhat with the length of drive and diameter of pulley but not to an extent calling for attention in design.

The camber necessary to correct for a given misalignment also depends on the margin by which the width of the pulley face exceeds that of the belt. For errors of alignment and want of truth in belts prevailing in good practice sufficient camber is provided if the pulley width is 15 per cent. greater than the belt width and the ratio $\frac{\text{diameter difference}}{\text{diameter}}$ is made ½ per cent. on each pulley or 1 per cent. on one pulley the other being flat. If these cambers are doubled they will cover any misalignment which should be tolerated in practice. Where one pulley is much smaller than the other it is good practice to leave the small pulley flat and provide sufficient camber on the larger pulley to secure proper tracking.

The profile of a crowned pulley should be a smooth curve or two symmetrical smooth curves with a flat central portion not exceeding half the width of the belt. Flat tapers set up local strains and impair the contact between belt and pulley.

Velocity Ratios.

The effective radius of a pulley is the radius measured to the neutral layer of the belt running over the pulley. For drives between pulleys of very different diameters it is necessary to take account of the belt thickness if exact speed ratios are required. The neutral layer of solid woven, ply and double leather belting is at mid-section. For single leather belting it is about $\frac{5}{8}$ of the thickness from the grain (skin) side. Hence the effective pulley diameter depends on the belt used as follows:—

Belt.	Effective Pulley Diameter.
Single leather (grain to pulley) .	$D + \frac{5}{4}t$
Single leather (flesh to pulley) . .	$D + \frac{3}{4}t$
Double leather / Fabric	$D + t$

Length of Belts.

The exact geometrical length of a belt in feet to run over pulleys of diameters D ,d feet at a centre distance l feet is:—

(*a*) Open drive:

$$L_1 = \frac{\pi}{2}(D + d) + \alpha(D - d) + 2l \cos \alpha.$$

$$\text{where } \sin \alpha = \frac{D - d}{2l}$$

(*b*) Crossed drive:

$$L_2 = \frac{\pi}{2}(D + d) + \beta(D - d) + 2l \cos \beta.$$

$$\text{where } \sin \beta = \frac{D + d}{2l}$$

For most purposes these formulæ can be simplified to:—

$$L = 2l + 1{\cdot}57\,(D + d) + \frac{(D \mp d)^2}{4l}$$

the upper or lower sign being taken according as the drive is open or crossed.

For drives which are not self-tensioning it is necessary to cut the belt to a length somewhat shorter than the above geometrical length. A reasonable allowance for ordinary conditions is 1 per cent. or $\frac{1}{8}$ in. per foot. With vertical drives and with inferior belting it may be necessary to increase this allowance by one-half; while for reasonably long horizontal drives and good belting it may be possible to reduce it by one-half.

Belts exceeding 6 ins. in width should be mounted with the aid of belt clamps. Narrower belts may be mounted with the help of a special mounting pole; if mounted by hand the shaft must be turned slowly under complete control to avoid accident or damage to the belt.

Belt Performance and Efficiency.

Under reasonable conditions a leather belt or a suitably impregnated solid woven belt is capable of a tension ratio $\frac{T_1}{T_2} = 5$. The permissible tension ratio falls with the pulley diameter, but even over small pulleys a suitable type of belt will give this tension ratio provided atmospheric conditions and maintenance are satisfactory.

The conditions are exceptional when a properly chosen belt is incapable of a tension ratio $\frac{T_1}{T_2} = 3$. With a tension ratio of 5, 80 per cent. of the tension in the belt is effective ; with a tension ratio of 3 the percentage is 67.

The power losses in a belt drive arise partly from the speed loss due to belt creep or local slip and partly from the loss of torque due to bending, windage and bearing friction. This last item is attributable to the belt in the sense that it arises from the belt tension. The creep loss increases with the load transmitted while the torque losses are more or less constant. At light loads the efficiency is low because the torque losses are proportionately high and at heavy loads it falls again because of the increasing creep loss. At a certain load an optimum eccentricity is attained ; if the creep loss is proportional to load this occurs when the creep loss and torque loss are equal, more usually it occurs when the creep loss is about half the torque loss.

As a rule the creep loss is less for fabric belts, while torque losses are less for leather belts. Creep loss (based on pulley diameter plus effective belt thickness) should never exceed 1 per cent. in a properly belted drive. Bending losses depend on the pulley diameters and belt thickness. Windage depends on the belt speed and length and the loss of effective tension involved may be taken as $\frac{LV^2}{200}$ lb. per in. width, where L is the total length of *belt* in ft. and V its speed in thousands of feet per minute. Apart from windage the diminution of effective tension on an ordinary motor drive due to torque loss is approximately $\frac{1}{3}$ lb. per inch width for rawhide belts, $\frac{1}{2}$ lb. per inch width for single leather belts, $\frac{3}{4}$ lb. per in. for double leather, for friction-surface rubber and rubber impregnated belts and 1 lb. per in. for solid woven belts with bituminous or oil impregnation and laminated leather belts.

The actual efficiency of a belt drive under well-designed conditions should be from 97 to 98½ per cent., higher values ruling for larger pulleys. A rawhide belt, for instance, may give an efficiency of 98½ per cent. over a motor pulley as small as 6 ins. in diameter.

Short-Centre Drives.

Special care s required in the design of short-centre drives if satisfactory service is to be assured. These drives work under more onerous conditions than long horizontal drives because (*a*) the frequency of stress cycles in the belt is greater, (*b*) the belt usually has to run over a small pulley, and (*c*) the intrinsic self-tensioning property due to belt sag is ineffective.

The disadvantages arising from frequent stress changes are more apparent than real. Under reasonable conditions a belt will transmit $\frac{1}{5}$ horse-power year per cubic inch, which corresponds to about 6 horse-power years per square foot of single leather belting, and a short belt while lasting a shorter time is less expensive to buy.

A small pulley in a short-centre drive involves a small angle of embrace and reduces the area of belt contact besides introducing severe bending strains in the belt. These effects can be minimised by adopting a wide flexible type of belt, preferably made endless, by ensuring that its frictional properties are kept high by intelligent dressing, and by making the small pulley without camber, perfectly true on its shaft and with a polished surface. The pulley diameter should be as large as possible provided the belt speed does not exceed about 5,000 ft. per minute. Belts for small pulley service should not exceed 5 mm. in thickness and should not be loaded to give a greater effective tension than 20–25 lbs. per in. width for leather belts or 25–30 lbs. per in. width for fabric belts. Rawhide and rubber impregnated woven belts are specially suited for small pulley service.

Suitable horse-power ratings for small-pulley short-centre drives are as follows:—

Belt.	Horse-power.
Leather	$b\frac{nd}{50}\left\{1-\left(\frac{nd}{270}\right)^2\right\}$ per inch.
Fabric	$b\frac{nd}{40}\left\{1-\left(\frac{nd}{300}\right)^2\right\}$,,
V-belt	$b^2\frac{nd}{12}\left\{1-\left(\frac{nd}{300}\right)^2\right\}$ per rope.

where b = width of belt, ins.,

d = pulley diameter, ins.,

n = *hundreds* of r.p.m.

In a long horizontal drive the total tension in the belt increases considerably with the load transmitted owing mainly to the catenary form of the free belt between the pulleys. This means that the full tension is only applied at full load and is automatically relieved at smaller loads and when the drive is standing. Hence take-up is less frequent and belt life is increased.

Tensioning Devices.

The self-tensioning property is dependent on and increases with the value of $\frac{w^2h^2}{eT_0{}^2}$, where w is the weight of belt per unit length, h the horizontal centre distance, e the initial elastic stretch of the belt and T_0 the initial tension. This 'sag effect' can only be incorporated in short-centre drives by artificial means, and various devices have been applied with varying success.

Self-acting idler pulleys, either gravity or spring controlled, may be applied to the slack strand of the belt. Owing to the bending action involved they reduce the life of the belt, but a lighter type of idler of more generous diameter than is usual, set at a reasonable distance from the small pulley will give equal performance with less sacrifice of belt life, especially if the belt is made endless. Self-acting coupled idlers are occasionally useful where the necessary weight of an ordinary idler would be excessive.

An idler made of resilient material can be employed to press on to the belt in contact with the pulley. In this case the belt must be endless. Bending action is eliminated by this device but some 'ironing' is introduced in its place. Pressure rollers of this kind can be made self-adjusting.

The most satisfactory self-tensioning method consists in making one of the pulley axes so that it automatically adjusts itself about some pivotal axis in response to changes in the power transmitted by the drive.

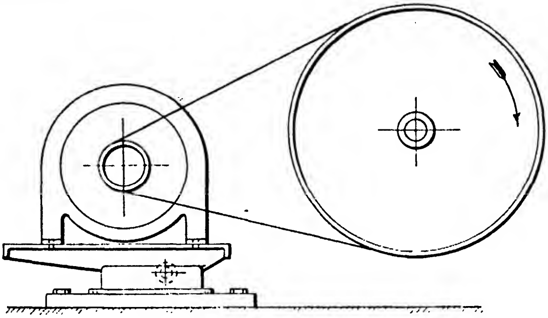

FIG. 4.—Pivoted Motor Base.

An electric motor can be mounted on a pivoted base in such a way that its own weight produces tension in the belt. This device (fig. 4) gives increasing tension with increasing load if the tight side of the belt is below; it is specially suited for short horizontal drives arranged in this way. Care should be taken not to over-tension the belt and the necessary overhang should be calculated from the weight of motor and base and the power and tension ratio required from the belt.

If the motor frame itself is mounted on pivots relatively close to the axis of the pulley a self-tensioning property can be obtained independent of the weight of the motor such as to maintain an almost constant tension ratio at all loads. For very short drives this device is valuable but its capacity for automatic take-up is limited.

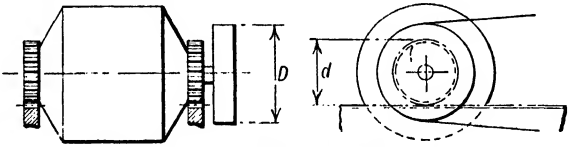

FIG. 5.—Rack and Pinion Mounting for Motor.

With the rack and pinion mounting (fig. 5) the motor or machine frame is pivoted about an axis which automatically adjusts itself as stretch takes place and the belt tension is automatically controlled to give a constant tension ratio of $\frac{T_1}{T_2} = \frac{D + d}{D - d}$ at all loads, where D is the pulley diameter and d the pitch diameter of the pinions. If, for example, pinions are made half the pulley diameter the tension ratio will always be 3 : 1. If this device is used for oblique or vertical drives means must, of course, be employed to keep the gears in mesh.

VEE-BELT DRIVES.

The vee-belt made of fabric and rubber with various arrangements in cross-section, moulded and vulcanised into an endless rope of trapezoidal section, has several important advantages for use in short drives. Owing to the absence of joints as well as the intrinsic strength of the material, it is possible to employ a factor of safety sufficiently large to enable the belt to be run over small pulleys with a reasonable length of life, and at the same time to effect a saving in pulley width over ordinary belting varying from 10 per cent. with 6-inch diameter pulleys to 30 per cent. with 15-inch pulleys. The use of multiple ropes makes sudden breakdown of the drive a remote contingency, although the breakage of a single rope necessitates the replacement of the whole set for satisfactory operation. The fact that the ropes are made endless helps to eliminate noise and intermittent slip on the pulleys, but involves limitations in the design of machinery to enable the ropes to be mounted. The vee-belt drive is capable of accommodating greater errors of alignment than the flat belt drive, and such errors may therefore pass undetected, though they cause

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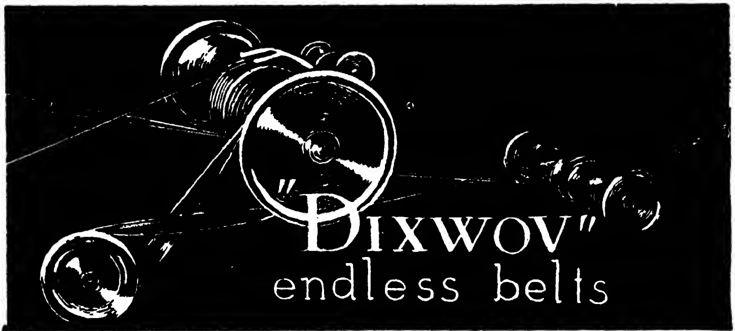

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inequality in the distribution of tension and load between the various ropes, and give rise to wear against the sides of the grooves. Inequality in distribution of load may also arise from want of accuracy in machining the grooves in the pulleys or from the use together of new and worn ropes. Moreover, the precise velocity ratio of a vee-belt drive is uncertain to some extent; it varies somewhat with the tension and age of ropes, and on the relatively small pulleys common with motor drives the effective diameter may be as much as 1 per cent. less than the nominal pitch diameter. The smaller the motor pulley diameter the greater does this discrepancy become, unless the angle of the grooves is diminished to suit the modified cross-section of the bent rope.

The great advantage of the vee-belt lies in the large effective coefficient of friction produced by the use of 40° grooves. This enables tension ratios as high as 20/1 to be employed without bodily slip and gives values of the coefficient of performance $\frac{T_1-T_2}{T_1}$ as high as 0·95. This not only increases the power which can be transmitted for a given maximum tension, in proportion to the coefficient of performance, but makes the capacity of the drive less sensitive to the effects of the small angle of embrace which is common with motor drives. The high coefficient of performance also enables the drive to take advantage of any intrinsic sag effect in the drive to an extent out of all proportion to the improved coefficient of performance and an extent which becomes still greater at high speeds. On the other hand, since the drive does not provide *per se* any automatic tension control, periodic take-up is necessary, and this is liable to introduce an excessive tension unless carried out with discretion.

The expense of the vee-belt drive can be reduced in some cases by the use of the vee-flat drive in which the ropes run from vee-grooves in the small pulley on to the flat surface of the larger pulley. This arrangement is satisfactory so long as the diameter of the large pulley is large enough to provide a good tension ratio, a condition which will generally be fulfilled if this diameter exceeds forty times the width by which the rope is rated.

Vee Ropes

Vee ropes are made in five different sizes. The sketch below shows the profile of the pulley groove and the dimension 'A' designates the size of the rope.

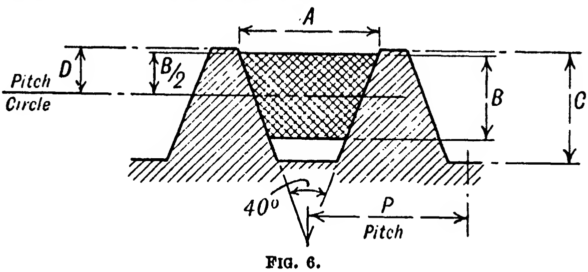


Fig. 6.

Rope Size. A	$\frac{1}{2}$ in.	$\frac{21}{32}$ in.	$\frac{7}{8}$ in.	$1\frac{1}{4}$ ins.	$1\frac{1}{2}$ ins.
P.C.D. of smallest advisable pulley . .	ins. 3	ins. 5	ins. 9	ins. 13	ins. 21
B	$\frac{5}{16}$	$\frac{7}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	1
C	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{15}{16}$	$1\frac{3}{16}$	$1\frac{7}{16}$
D	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{9}{16}$
P	$\frac{5}{8}$	$\frac{3}{4}$	1	$1\frac{7}{16}$	$1\frac{3}{4}$

Vee-rope drives should be capable of having endless ropes mounted, and there should be accommodation for about 2 per cent. of stretch on normally loaded drives. The ropes are particularly useful for driving at short centres, and the following list of standard ropes indicates the range of drive lengths on which vee-ropes can be installed.

STANDARD LENGTHS OF ENDLESS VEE ROPES.

Nominal Length. Inches.	Pitch Circle Length. Inches.				
	½ in. Ropes.	21/32 in. Ropes.	⅞ in. Ropes.	1¼ in. Ropes.	1½ in. Ropes.
26	27	—	—	—	—
35	36	36·5	—	—	—
38	39	39·5	—	—	—
42	43	43·5	—	—	—
43	44	—	—	—	—
46	47	47·5	—	—	—
47	48	—	—	—	—
49½	—	51·0	—	—	—
51	52	52·5	—	—	—
55	56	—	—	—	—
60	61	61·5	62	—	—
62½	—	64·0	—	—	—
68	69	69·5	70	—	—
70	71	—	—	—	—
75	76	76·5	77	—	—
80	81	81·5	—	—	—
81	—	—	83	—	—
82	—	83·5	—	—	—
85	86	86·5	86·5	—	—
90	91	91·5	92	—	—
96	—	—	98	—	—
97½	—	99	—	—	—
105	106	106·5	107	—	—
108	—	109·5	—	—	—
109½	—	—	—	112	—
112½	—	114	—	—	—
120	121	121·5	122	122·5	—
128	—	129·5	130	130·5	—
134	—	—	136	—	—
144	—	145·5	146	146·5	—
158	—	159·5	—	—	—
162	—	—	164	165·5	—
173	—	174·5	175	—	—
180	—	181·5	183	182·5	—
195	—	196·5	197	197·5	198·5
210	—	211·5	213	213	213·6
240	—	240	240	240	240
270	—	270	270	270	270
300	—	300	300	300	300
314	—	—	314	316	314
330	—	—	330	330	330
360	—	—	360	360	360
420	—	—	420	420	420
480	—	—	—	480	480
540	—	—	—	540	540
600	—	—	—	600	600
660	—	—	—	—	660
720	—	—	—	—	720

POWER RATINGS.

In designing a vee-rope drive, first decide what maximum horse-power will be imposed on the ropes. This maximum should allow for any peak loads or high starting torque. It may be necessary to increase the nominal horse-power by 30 to 50 per cent. or more for drives to machines such as compressors, disintegrators, or pumps where the load is intermittent or suddenly applied.

The choice of rope size will then be made according to the horse-power, the diameter of the small pulley and the desirable maximum width of pulleys. It is usually best to choose the smallest size rope for the horse-power and width available. It is good practice to aim at a rope speed of about 3,000 ft. per min.

Having determined the maximum horse-power, size of rope, the pulley diameters and the speed, the horse-power available per rope is then obtained from the tables of horse-power rating and correction factors, which are self-explanatory.

In order to find the number of ropes required, divide the maximum horse-power by the horse-power per rope. Remainder fractions greater than 0·2 count as one rope.

*** Basic Horse-Power Rating of Endless Vee Ropes in Vee-Grooved Pulleys.**

The figures give the maximum rating on the minimum recommended pulley diameters when the speed ratio of the drive is 1.

Rope Speed. Ft. per Min.	Basic Horse-Power.				
	½ in. Rope.	21/32 in. Rope.	⅞ in. Rope.	1¼ in. Rope.	1½ in. Rope
400	0·36	0·57	1·11	2·20	2·97
600	0·56	0·85	1·72	3·29	4·45
800	0·72	1·10	2·25	4·38	5·89
1,000	0·9	1·33	2·82	5·46	7·33
1,200	1·08	1·57	3·34	6·56	8·88
1,400	1·23	1·81	3·87	7·56	10·23
1,600	1·39	2·07	4·39	8·52	11·55
1,800	1·56	2·29	4·88	9·42	12·83
2,000	1·69	2·51	5·32	10·30	14·00
2,200	1·85	2·74	5·76	11·20	15·15
2,400	1·98	2·91	6·19	12·10	16·20
2,600	2·11	3·11	6·58	12·95	17·18
2,800	2·24	3·28	7·00	13·76	18·16
3,000	2·35	3·45	7·34	14·55	19·10
3,200	2·46	3·62	7·69	15·30	20·00
3,400	2·58	3·76	8·00	15·95	20·85
3,600	2·68	3·89	8·25	16·54	21·55
3,800	2·78	4·00	8·51	17·06	22·20
4,000	2·86	4·11	8·72	17·43	22·80
4,200	2·90	4·20	8·90	17·73	23·27
4,400	2·93	4·26	9·06	17·91	23·57
4,600	2·96	4·30	9·19	18·06	23·80
4,800	2·97	4·33	9·28	18·12	23·95
5,000	2·97	4·33	9·34	18·15	24·00

Note.—For larger pulley diameters, for higher speed ratios, and for vee-flat drives, the basic horse-power rating should be multiplied by the factors given in the following tables:—

Correction Factors for Speed Ratios.

Speed Ratio of Drive.	Correction Factors.	
	VV Drive.	V Flat Drive.
1·0	1·0	—
1·4	0·97	—
1·8	0·95	—
2·2	0·93	—
2·6	0·92	—
3·0	0·9	0·82
3·4	0·89	0·81
3·8	0·88	0·8
4·2	0·87	0·79
4·6	0·86	0·78
5·2	0·85	0·77
5·6	0·84	0·76
6·0	0·83	0·75
7·0	0·8	0·72

Correction Factors for Diameter of Small Pulley.

P.C.D. of Smallest Pulley, Inches.	Correction Factors.				
	½ In. Rope.	$\frac{21}{32}$ In. Rope.	⅞ In. Rope.	1¼ In. Rope.	1½ In. Rope.
3	1·00	—	—	—	—
4	1·09	—	—	—	—
5	1·15	1·00	—	—	—
6	1·20	1·06	—	—	—
7	1·24	1·10	—	—	—
8	1·26	1·15	—	—	—
9	1·28	1·18	1·00	—	—
10	1·30	1·20	1·04	—	—
13	1·30	1·26	1·13	1·00	—
16	1·30	1·30	1·20	1·09	—
21	1·30	1·30	1·27	1·18	1·00
24	1·30	1·30	1·30	1·22	1·09
28	1·30	1·30	1·30	1·26	1·16
33	1·30	1·30	1·30	1·30	1·22
38	1·30	1·30	1·30	1·30	1·26
44	1·30	1·30	1·30	1·30	1·30

Spring Belts.

Close wound helical springs are sometimes used to transmit small powers at high rotational speeds. The two ends of a spring belt may be connected either by a tapered end type coupling in which one end is tapered and screwed into the other, or by a loose type coupling consisting of a short coupling piece of spring of suitable diameter to screw into both ends of the main belt.

The pulleys on which the spring belts run are made with a total groove angle of 15° and the maximum belt speed is 4,800 ft. per minute. The minimum pulley diameters suitable for various standard sizes of belt are shown together with power ratings and permissible extensions in the table:—

Size of Belt. Ins.	Maximum Extension. Ins. per Ft.	Minimum Pulley Diameter. Ins.	Horse-Power at 4,800 Ft. per Min.
$\frac{3}{16}$	1¼	2¾	0·295
$\frac{7}{32}$	1	3¼	0·401
¼	1	3¾	0·525
$\frac{9}{32}$	1	4¼	0·668
$\frac{5}{16}$	1	4¾	0·825
$\frac{11}{32}$	1	5⅛	0·995
⅜	1	5⅝	1·100
$\frac{7}{16}$	1	6¼	1·450
½	1	7½	1·900

Rope Drives.

Three- and four-strand cotton ropes are much used for the transmission of power in main drives and group drives in this country. The three-strand rope has somewhat greater flexibility and strength, while the four-strand rope is less extensible and generally provides a greater surface of contact with the pulley grooves.

As an indication of the strength of cotton ropes the following figures for cotton rope slings supplied by Thomas Hart, Ltd., of Blackburn, may be useful. A factor of safety of 10 is common for slings. In rope driving there is generally a very large margin of strength.

Diameter of Rope.	Approximate Breaking Stress.	Diameter of Rope.	Approximate Breaking Stress.
Ins.	Lbs.	Ins.	Lbs.
$\frac{3}{4}$	4,150	$1\frac{5}{8}$	17,500
$\frac{7}{8}$	5,250	$1\frac{3}{4}$	21,000
1	6,560	$1\frac{7}{8}$	24,000
$1\frac{1}{8}$	8,530	2	27,000
$1\frac{1}{4}$	10,380	$2\frac{1}{4}$	33,000
$1\frac{3}{8}$	12,250	$2\frac{1}{2}$	39,000
$1\frac{1}{2}$	14,870		

Rope Pulleys.

Rope pulleys are usually made of cast iron of good quality, providing a hard bright finished surface in the grooves after machining. The grooves are generally cored in the casting, but may be milled from the solid for small diameters. Careful design and foundrywork are needed to obviate porosity in the rim at the roots of arms, cracked bosses and transverse cracks at the smallest sections of the arms. The grooves are almost invariably straight-sided and may or may not have flanges between the grooves. The total groove angle is usually 45° or 40°, but for small ropes on short centre drives 30° total groove angle is occasionally used. Methods of setting out grooves are indicated in fig. 7 for flangeless grooves of 40° angle and in fig. 8 and the accompanying table for flanged grooves of 45° angle. It will be clear that both types of groove give ample allowance for rope wear and permit of a reasonable range of rope diameters without any fear of the rope bearing on the bottom of the groove.

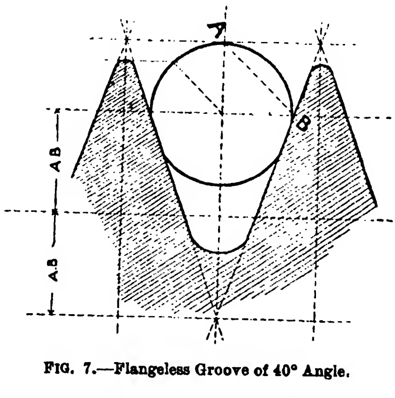


Fig. 7.—Flangeless Groove of 40° Angle.

The usual limit of peripheral speed for cast-iron pulleys is 6,000 ft. per minute for solid pulleys, and 5,300 ft. per minute for split pulleys. For higher speeds up to 8,000 ft. per minute pulleys are built up with steel arms or spokes.

Table of Measurements in Inches.

Diameter of Rope	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2
Pitch of Grooves .	·69	·84	1·02	1·18	1·35	1·51	1·675	1·82	2	2·16	2·35	2·51	2·72
Depth of Groove .	·75	·93	1·125	1·31	1·5	1·68	1.875	2·06	2·25	2·43	2·625	2·81	3
Thickness of Mid-feather . .	·125	·156	·187	·21	·25	·281	·31	·343	·375	·40	·43	·46	·5

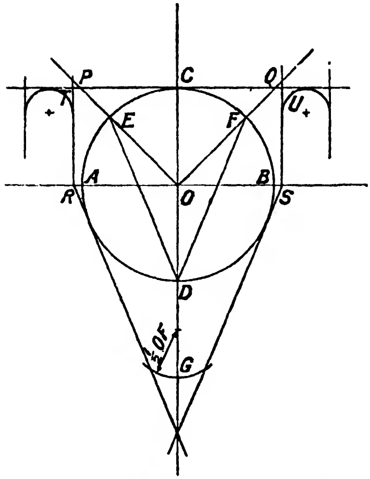


FIG. 8.—Flanged Groove of 45° Angle.

Rope Sizes:	¾ In.	⅞ In.	1 In.	1⅛ In.	1¼ In.	1⅜ In.	1½ In.	1⅝ In.	1¾ In.	1⅞ In.	2 Ins.
	Minimum advisable Pulley Diameters in Inches.										
Rope Speeds:	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
1,000	10	12½	15	18½	20	24	28	31	35	40	45
1,500	11½	14	16½	20	22	26	29	33	38	42	47
2,000	12½	15	18	21½	24	28	31	35½	41	44½	50
2,500	13½	16½	19½	23	26	30	34	38½	44	48	53½
3,000	15	18	21	25	28½	32½	37	41½	47	51½	57
3,500	16½	19½	23	27	31	35	40	44	50	55	61
4,000	18½	21	25	29	33½	38	43	47½	53½	59	65
4,500	—	23	27	31	36	41	45½	51	57	63	69
5,000	—	25	29	33½	38½	44	49½	54½	60½	67	74
5,500	—	27½	31	35½	41	47	52	58	65	71½	79
6,000	—	30	33	38	44	50	55½	62½	69½	76	84
6,500	—	—	35	40½	47	54	59	67	74	82	90
7,000	—	—	37	43	50	58	63	71½	79	88	96

SYSTEMS OF DRIVING.

The multiple rope drive consists of a number of independent endless ropes running side by side. It has a wide margin of reliability and the incipient failure of a single rope does not endanger the whole drive. The multiple rope system is in general use in this country.

The continuous rope drive consists of a single continuous rope passing from groove to groove of the drive and tensioned by means of a tension pulley. Applications for which it is specially suited are:—

1. Short centre distances with large loads.
2. Vertical drives, in which tension control is important.
3. Special forms of angular drive.

A crossed right-angle drive on the continuous system is shown diagrammatically in fig. 9.

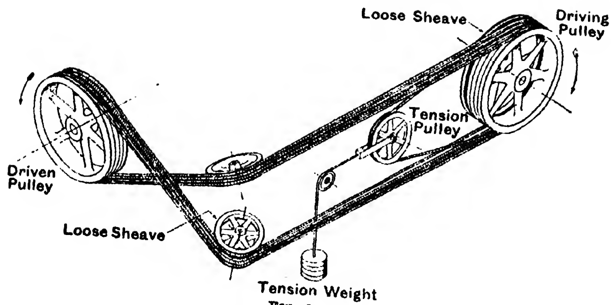


FIG. 9.

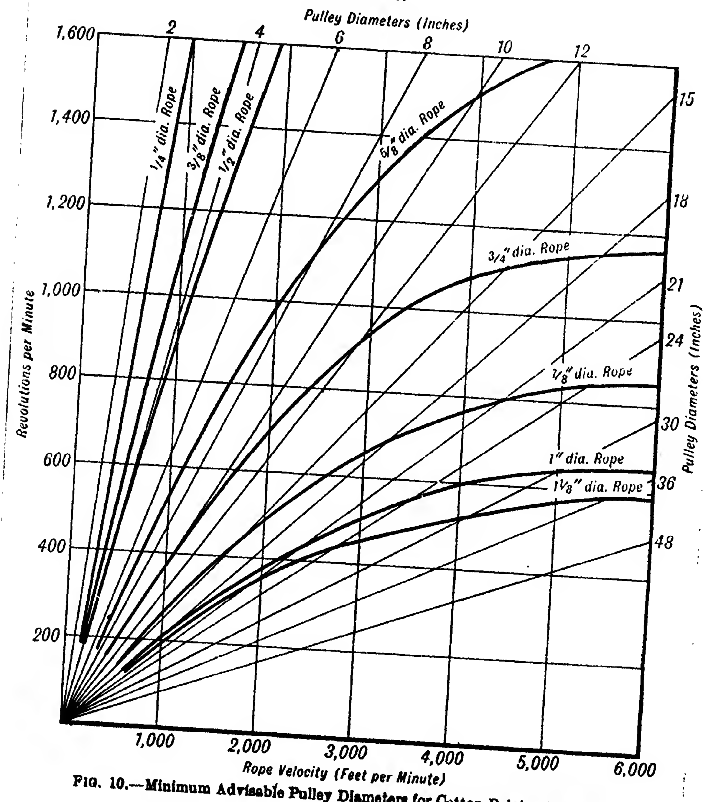


FIG. 10.—Minimum Advisable Pulley Diameters for Cotton Driving Ropes

Either system of rope drive provides greater flexibility than belt drives for skew and angle drives and with the aid of guide pulleys almost any relative position of driving and driven shafts can be accommodated.

Design of Rope Drives.

As in the case of belt drives the horse-power transmitted by a rope drive is $\text{H.P.} = \frac{T \times V}{33{,}000}$, where T is the total effective tension of all the ropes. The belt speed V is determined by the angular speed N r.p.m. and the pulley diameter D measured to the centre of the ropes. Where possible it is good practice to arrange the pulley diameters to give a rope speed of 4,500–5,000 ft. per minute. At high rotational speeds this requires a relatively small pulley diameter and consequently a relatively large number of small ropes. The permissible ratio of pulley to rope diameter then becomes important.

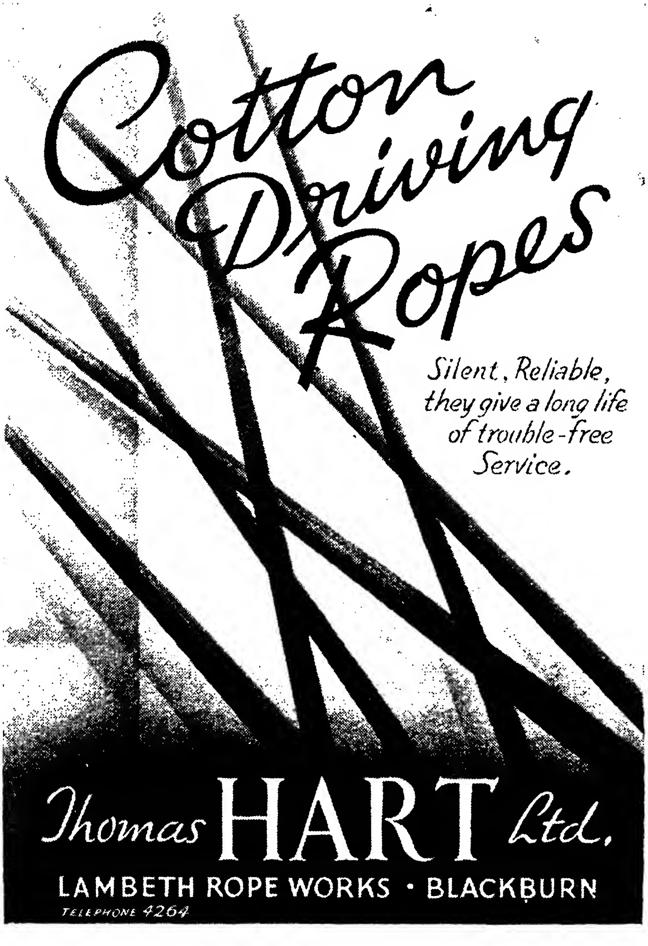
This ratio depends on the life required from the ropes, on the perfection of pulley and groove design and on the power rating demanded from each rope. The accompanying chart (fig. 10) shows the minimum pulley diameters advisable for use with good cotton ropes of various sizes at speeds of revolution up to 1,600 r.p.m. It is based on the assumption of a rope-life of 5–8 years and on the use of power capacities chosen according to the conditions of operation.

For main drives over pulleys of ample diameter the following table of horse powers will be found useful.

Table of Powers for Cotton Ropes.

Velocity in Ft. per Min.	Diameter of Ropes.										
	¾″	⅞″	1″	1⅛″	1¼″	1⅜″	1½″	1⅝″	1¾″	1⅞″	2″
2,000	3·6	5·0	6·5	8·2	10·1	12·2	14·9	17·2	20	22·9	26·1
2,200	3·9	5·5	7·1	9·0	11·1	13·4	16·3	18·9	22	25·3	28·7
2,400	4·3	6·0	7·8	9·9	12·1	14·7	17·8	20·7	24	27·6	31·3
2,600	4·6	6·5	8·4	10·7	13·1	15·9	19·2	22·4	26	29·9	33·9
2,800	5·0	7·0	9·1	11·5	14·1	17·1	20·8	24·1	28	32·2	36·5
3,000	5·4	7·5	9·7	12·3	15·1	18·3	22·3	25·8	30	34·5	39·1
3.200	5·7	8·0	10·4	13·2	16·2	19·6	23·8	27·6	32	36·8	41·8
3,400	6·1	8·5	11·0	14·0	17·2	20·8	25·3	29·3	34	39·1	44·4
3,600	6·4	9·0	11·7	14·8	18·2	22·0	26·7	31·0	36	41·4	47·0
3,800	6·8	9·5	12·3	15·6	19·2	23·2	28·2	32·7	38	43·7	49·6
4,000	7·2	10·0	13·0	16·4	20·2	24·5	29·7	34·5	40	46·0	52·2
4,200	7·5	10·5	13·6	17·3	21·2	25·7	31·2	36·2	42	48·2	54·8
4,400	7·9	11·0	14·3	18·1	22·2	26·9	32·7	37·9	44	50·6	57·4
4,600	8·2	11·5	14·9	18·9	23·2	28·1	34·2	39·6	46	52·9	60·0
4,800	8·6	12·0	15·6	19·8	24·2	29·4	35·7	41·4	48	55·2	62·7
5,000	9·0	12·5	16·2	20·6	25·3	30·6	37·1	43·1	50	57·5	65·3
5,200	9·3	13·0	16·9	21·4	26·3	31·8	38·6	44·8	52	59·8	67·9
5,400	9·7	13·5	17·5	22·2	27·3	33·0	40·1	46·5	54	62·1	70·5
5,600	10·0	14·0	18·2	23·1	28·3	34·3	41·6	48·3	56	64·4	73·1
5,800	10·4	14·5	18·8	23·9	29·3	35·5	43·1	50·0	58	66·7	75·7
6,000	10·8	15·0	19·5	24·7	30·3	36·7	44·6	51·5	60	69·0	78·3
6,200	11·1	15·5	20·1	25·5	31·3	37·9	46·1	53·4	62	71·3	80·9
6,400	11·5	16·0	20·8	26·4	32·4	39·2	47·6	55·2	64	73·6	83·6
6,600	11·8	16·5	21·4	27·2	33·4	40·4	49·0	56·9	66	75·9	86·2
6,800	12·2	17·0	22·1	28·0	34·3	41·6	50·5	58·5	68	78·2	88·8
7,000	12·6	17·5	22·7	28·8	35·4	42·8	52·0	60·3	70	80·5	91·4

The choice of a suitable diameter of rope in relation to the pulley diameter is affected by consideration of the two principal causes of rope deterioration: bending and surface abrasion. Larger ropes increase the tendency to failure by bending while surface abrasion is more important with small ropes. Provided the pulley grooves are highly polished experience shows that the susceptibility to surface wear is less important than that to bending effects, so that it may be regarded as good practice to employ as small ropes as the width of pulley available will justify.


Cotton Driving Ropes
Silent, Reliable, they give a long life of trouble-free Service.
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Some ropes revolve in the grooves. Others do not, and wear to some extent to the shape of the groove. There is no obvious reason for the different behaviour and in actual practice it seems to make no difference to their life.

For small-pulley and short-centre drives allowance should be made for:—

(1) the acute angled grooves commonly employed on these drives.
(2) the diminished arc of contact on the smaller pulley.

These allowances are made by means of correction factors f_1 and f_2 respectively, where f_1 is given by the following table:—

Groove Angle.	Factor f_1
45°	0·94
40°	1·0
35°	1·06
30°	1·20

while f_2 is derived from the formula:—

$$f_2 = 1 - \frac{5}{11}\,\frac{D - d}{l}$$

where D, d are the pulley diameters and l the centre distance, all in the same units.

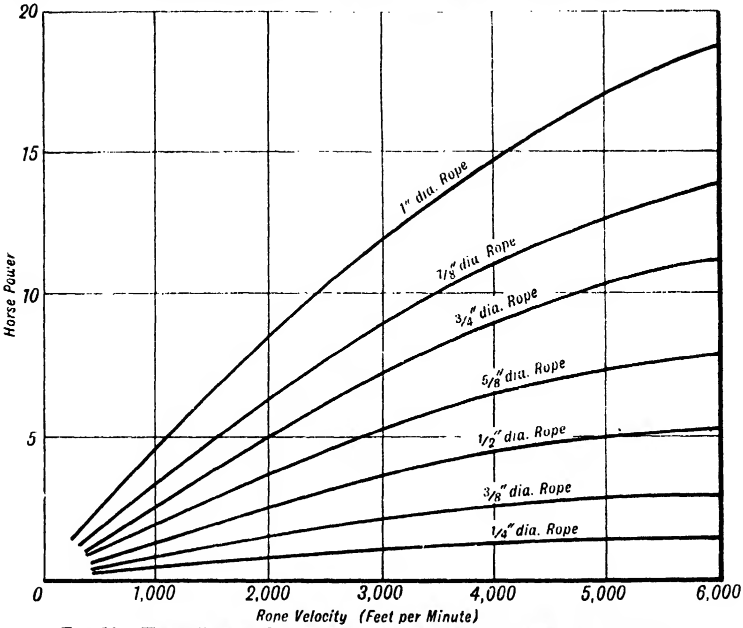
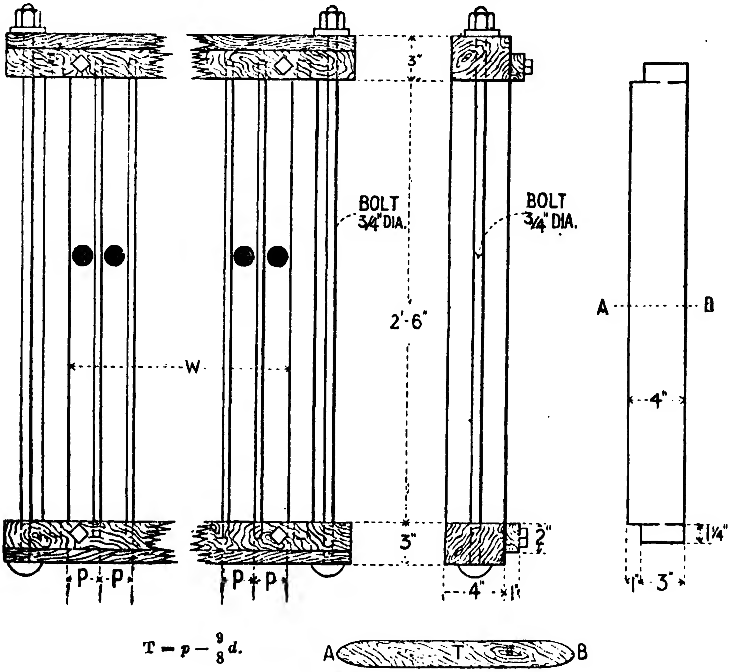


FIG. 11.—HORSE-POWER CURVES FOR SMALLER COTTON ROPES 40° GROOVES.

For convenience of reference the accompanying chart fig. 11 shows the power capacities for small rope drives already corrected for rope speed. The correction factors f_1 and f_2 should be applied to these capacities as necessary.

Fluctuating Loads.

The resilience of cotton rope drives makes them specially suitable for cushioning heavy cyclic peak loads whether the fluctuations are due to the driving or the driven element. This resilience



$$T = p - \frac{9}{8}d.$$

Fig. 12.

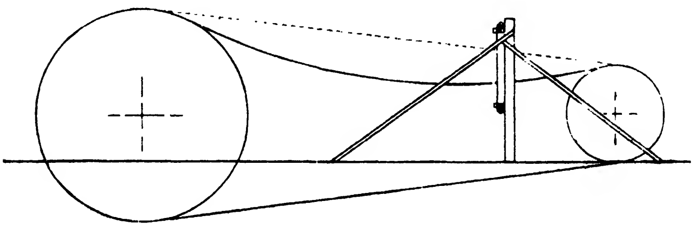


Fig. 13.

increases with the length of drive which from the operational standpoint is only limited by the dangers of rope swinging and rubbing. Rope swinging is effectively remedied by means of a rope guard such as that shown in figs. 12 and 13. In some cases where this swinging is due to synchronism it is competely cured by altering either the speed or centre distance of the drive When a rope drive is employed to transmit the power from engines, particularly internal combustion engines, swinging is often an indication of insufficient flywheel effect, the remedy for which is obvious.

In designing a rope drive for an engine with cyclic irregularities of torque it is sound practice to base calculations on the maximum rather than the mean cyclic torque or to allow extra ropes above those required under steady conditions. This allowance may amount to 50 per cent. in exceptional cases of single-cylinder engines.

Instructions for Splicing Four-Strand Cotton Driving Ropes.

Storing, Stretching, etc.

If it is necessary to store the ropes before fixing, they should be kept in a dry room, not on an earthen floor. When ready for fixing uncoil the ropes left-handed—the opposite to the way the hands of a clock move. Then stretch out with a pair of hand blocks, but do not allow the rope to revolve during this operation, or the 'turn' or 'twist' will be taken out of it. Everything depends upon the strength of shafting, etc., as to how tight the rope should be stretched, but under ordinary circumstances it is usual to have about four men pulling on a pair of hand blocks. Pass a string around the drive to get the exact running length of rope, and by tying two bands around the rope, mark off the exact running length whilst it is on the stretch, allowing ten feet (five feet each end) for the splice. For ropes 2 ins. diameter allow 12 feet of splice.

Splicing.

Unlay the five feet each end that has been allowed for splice, that is, as far back as the bands which mark the running length of rope; it is advisable to tie a string around the end of each of the loose strands to prevent the twist from coming out. The strands are now interlaced together, one from the right-hand side being next to one from the left-hand side, and so on, as shown in

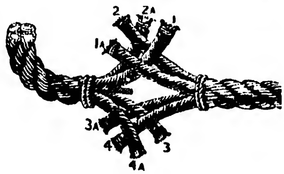


FIG. 14.

fig. 14. These strands should be pulled tight so that the bands butt together, and the loose strands on one side tied temporarily around the rope to keep the two ends of the rope in the position they now are.

Cut one of the bands, say the one on the right, and unlay strand No. 1A, at the same time laying in its place its fellow strand No. 1 from the opposite side of the rope. Do this for about four feet, and tie the two strands temporarily until all are ready for tucking in. The next strand to this, as will be seen in fig. 14, is numbered 2, but do not touch this now; take the next strand but one, numbered 3A, and unlay it, laying in its place strand No. 3, but only about 1 ft. 6 ins.; tie these temporarily. It is very important in splicing four-strand ropes that two strands next to each other should not be laid up in the same direction.

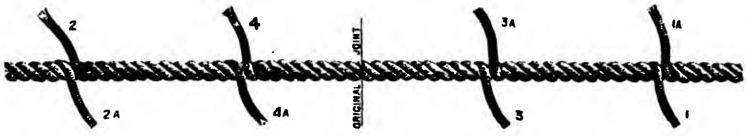


FIG. 15.

Next cut the other band, and proceed in the same manner with strands No. 2 and No. 4, but of course in the opposite direction. Care should be taken to keep the 'turn' or 'twist' in the strands, and they should be laid well and evenly down in their places. The splice should now be as fig. 15.

Begin now at joint No. 1 and 1A; untie the temporary fastening and shorten the strands to equal lengths of about 1 ft. 6 ins. An examination of the strands will show that the outside layer is composed of ten threads; for reference we will call these the friction bands; the internal portion of the strand we will call the tension strand. Take No. 1A and remove from it the ten friction bands and divide them into two lots of five, but do not cut them off. Now take strand No. 1 and unlay it two turns or laps, and whilst it is there remove from it the ten friction bands, divide them into two lots of five, but do not cut off. Now lay in again the tension strand only of No. 1 turn, and reduce it by leaving out about one-quarter, lay up the remaining three-quarters one turn, thus bringing it up to tension strand of No. 1A; tie the two with an overhand knot.

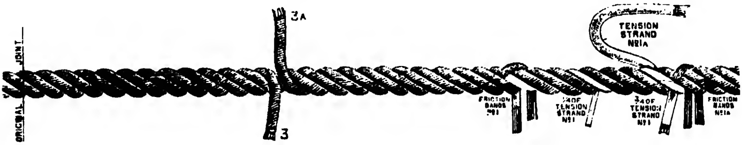


FIG. 16.

At this knot the rope should be about its original diameter, and the joint should appear as shown in fig. 16. Take tension strand 1A and with the splicing pin work it under and over tension strand No. 1, as shown at fig. 17, passing it under and over spirally about five times; it will then have reached friction bands No. 1. Now reduce it by cutting out a portion, and pass the remainder

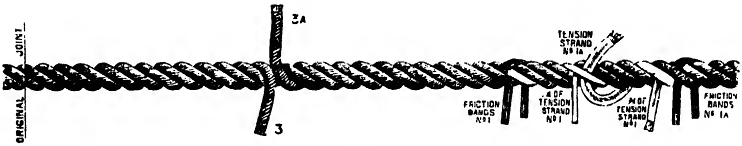


FIG. 17.

between the two lots of five friction bands, and lock by passing once or twice through the centre of the rope. This is clearly shown at fig. 18. Five friction bands are also locked through the rope as shown. This complete, turn to three-quarters of tension strand No. 1; this is locked exactly as tension strand No. 1A, five of the friction bands are also locked as previously shown, and the splice should now be as fig. 18.

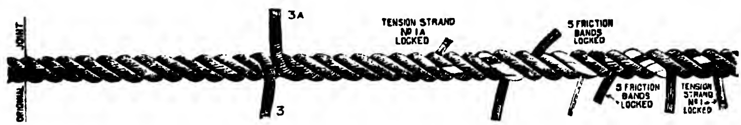


FIG. 18.

All loose ends are cut off and this portion is complete. Treat the two other joints in the same way and the splice is complete.

Mounting Rope on the Pulleys.

Put the rope on the small pulley, and as far around the large pulley as it will go; lash it there, but leave the lashing so that the rope has freedom to slip through it. Then bar pulley slowly round by band, or barring engine if there is one, and the rope will fall into its place. The groove of the small pulley and the face of the large pulley where the rope will touch in barring on should be well greased with tallow to prevent wedging. A piece of canvas may also be put on the rim of the large pulley to prevent it cutting the rope.

FAST AND LOOSE PULLEY.

Fig. 19 shows this combination in its most approved form. The working groove is set out at an angle of 30° on the lines previously described. A shallow intermediate groove provides for a gradual start and knock off, corresponding to the slip of a belt. The loose groove on the right is simply hollowed out to fit the rope, and is flanged on one side only. The rope shifter usually

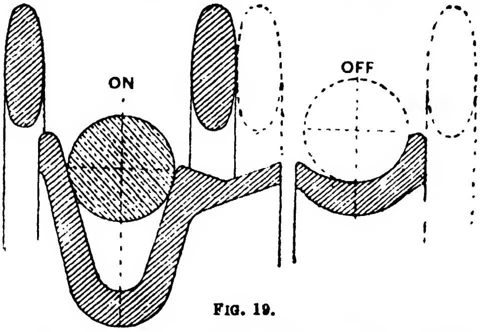


FIG. 19.

takes the curve of the pulleys, and is attached to a radius shaft and actuated by a small lever. A slight force on the slack side suffices to pass the rope in or out of gear, and there is very little friction.

Friction Clutches.

The choice of type depends on the purpose for which the clutch is to be employed, the space available, the frequency of operation and the dynamic requirements of the gearing.

It is generally desirable to provide sufficient adhesion for rapid acceleration without wearing the mating surfaces and without shock. The dimensions must be sufficient to keep temperature rise under operating conditions within reasonable bounds, but otherwise they should be as small as possible to reduce inertia and space occupied. Simplicity, compactness of design and accessibility are all desirable features, and a self-sustaining or locking device to maintain engagement during operation is usually necessary.

The chief types of friction clutch are :—

1. Axial clutches, in which the movement of engagement of the mating surfaces is axial: disc, plate, cone clutches.
2. Rim clutches, in which engagement is by radial movement of one of the mating surfaces: band, block, rim, centrifugal and coil clutches.

A typical cone-clutch is illustrated in fig. 20 and a simple multi-disc clutch in fig. 21.

The essential elements are the friction surfaces A. Half of these are keyed or bolted to the driving shaft or pulley B, while the other half are mounted on the driven shaft (or pulley) C, by means of a feather key D, which permits axial travel of the moving part of the clutch along the shaft. The two sets of surfaces are brought into or out of engagement by a system of levers E, operated by a sliding shifter collar.

The clutch is usually provided with some form of spring which may either tend to keep the mating surfaces apart, so as to produce a quick release, or may press the surfaces together as in the case of a motor-car clutch. In the former case the operator causes engagement, in the latter case disengagement of the clutch. The example in fig. 20 has springs of the former type. In order to ensure that the surfaces do not disengage of their own accord, the bell crank levers E, in fig. 20 are opened by the movement of a relatively steep coned surface which abuts on to a slight contrary cone F, on which the ends of the levers rest during operation. With this device no thrust is transmitted through the actuating gear in the fully engaged position, and external action is required before disengagement can occur. In other words the clutch is 'self-sustaining.' It will be noticed that the maximum engaging pressure occurs slightly before the sliding collar reaches its fully engaged position.

Plate clutches lend themselves to either pulley or coupling designs, and suitable constructions may be employed as slip couplings to protect against torque overload.

A rim clutch of a compact type common in machine tool practice is illustrated in fig. 22. In this case an expanding ring G, is opened into engagement with the wheels C, by the movement of the sliding sleeve F, operating through the toggle levers E. These move from their disengaged position (*a*), through their critical position into their normal working position (*b*), and therefore require positive initial disengagement.

In other designs of the internal expansion type of clutch, toggle effect is obtained through right- and left-hand screws, or through tapered dovetail slides, or actuation may be through cams after the fashion of a double-shoe motor-car brake.

Another type of rim friction clutch employs internal and external slippers or blocks which grip the rim of (usually) the driving member under the action of a linkage system operated by a

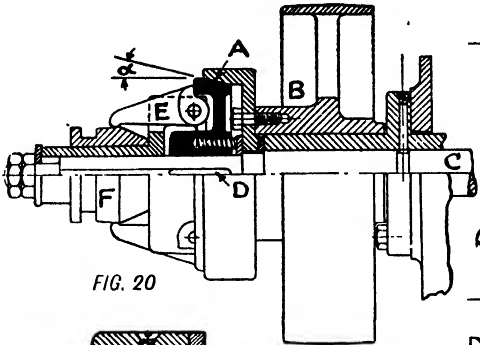


FIG. 20

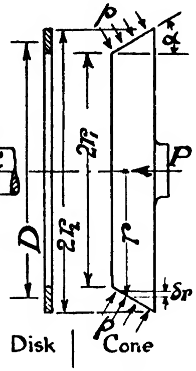


FIG. 23

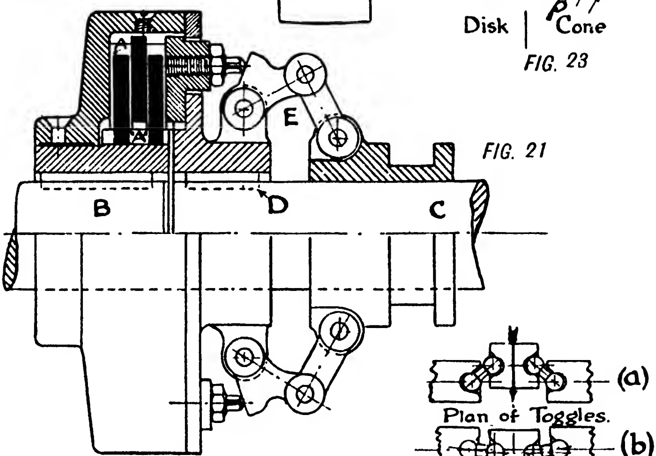
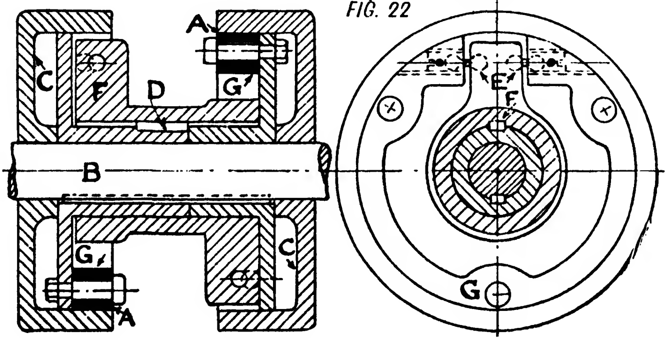


FIG. 21

FIG. 22



sliding collar on the other member. When this type is used in combination with a pulley, the rim is more conveniently incorporated in the pulley, whether this is the driving or driven member.

The band clutch is frequently used for haulage and similar purposes where considerable power has to be transmitted at low speeds. The strap is lined with bonded asbestos, and is tightened into engagement by means of a lever at the slack end, actuated by means of a sliding sleeve. Screw adjustment is usually provided at the tight end of the strap.

In centrifugal clutches engagement is brought about by centrifugal action on suitably weighted blocks. These clutches enable a motor or engine to start up light and gradually to take up the load as the speed increases; they are therefore specially useful when the driven machinery has a large inertia. Centrifugal clutches can be simple in construction with no linkage and few working parts. Types are available with loose shoes and with hinged shoes; the latter are preferred for pulsating loads, but are essentially unidirectional.

An improved engagement characteristic in centrifugal clutches can be obtained by a spring control which delays engagement until a desired speed is attained, which may be half or three-quarters of full speed. Delay may also be caused by trip weights which overcome the pressure of their restraining springs at a predetermined speed and so release the slippers which are then free to act under centrifugal action.

Centrifugal clutches can be constructed to slip at a specified overload and hand-controlled types are available which normally operate by centrifugal action, but can be engaged (at speed) or disengaged by hand. Centrifugal brakes to limit overspeed are similar in principle to the spring-controlled centrifugal clutch, but the shoes engage with a stationary external brake drum.

Principles of Clutch Action.

1. *Cone and Plate Clutches.*

After an initial period of running-in the wear will be uniform over the mating surfaces of these clutches. It is generally accepted and reasonably assumed that the rate of wear is proportional to the product of pressure (p) and rubbing speed (v). Hence over the mating surface of a cone or plate clutch :—

$$pv = \text{constant} = C, \text{ say.}$$

For a cone clutch with a semi-cone angle α, the applied axial face is found to be :—

$$P = 2\pi CB$$

where B is the width of the lining on the clutch, and the corresponding torque transmitted is :—

$$T = \mu\pi CBD \operatorname{cosec} \alpha$$

where D is the mean clutch diameter.

Hence the ratio of transmitted torque to applied axial face is $\dfrac{T}{P} = \dfrac{\mu}{2} D \operatorname{cosec} \alpha$.

In the case of a single plate clutch the same formula may be applied with $\alpha = 90°$, giving $\dfrac{T}{P} = \dfrac{\mu D}{2}$ where D is again the mean diameter $\dfrac{D_1 + D_2}{2}$

In the case of a multi-plate clutch having n pairs of mating surfaces:—

$$\frac{T}{P} = \frac{\mu}{2} n D$$

In order to avoid difficulty in disengagement it is desirable that the semi-cone-angle α of a cone clutch should be greater than the angle of friction: $\tan \alpha > \mu$. It is common practice to specify a value of α between 10 and 15°.

2. *Block Clutches.*

The wear on the lining of a block type of clutch is not uniform, but is proportional at any point to $\sin\theta$ where θ is the angle which the radius vector to the point makes with the line joining the centre of the drum to the (actual or virtual) centre of movement of the block or slipper. Since the rubbing speed in this type of clutch is uniform it follows that the intensity of pressure varies with $\sin\theta$ as defined above: $p = kR \cos\theta$, say.

In this case the radial engaging force is found to be :—

$$P = kBR^2 \int \sin^2 \theta \, d\theta;$$

while the transmitted torque is:—

$$T = \mu kBR^2 \int \sin\theta \, d\theta,$$

the limits of integration being chosen according to the angle subtended by the lining.

For a lining symmetrically disposed about the line of maximum pressure and subtending an angle β the ratio

$$\frac{T}{P} = \mu R \cdot \frac{4 \sin \frac{\beta}{2}}{\beta + \sin\beta}$$

The influence of the angle β is small for blocks subtending angles up to 90°, but becomes appreciable with larger arcs of contact.

3. *Band Clutches.*

Apart from the stiffening effect of the band itself, calculations for this type of clutch may be based on the ordinary laws of coil friction. If the actuating force P is applied at the slack end of the band and this band embraces an arc of α radians, then the transmitted torque will be:—

$T = RP\,(e^{\mu\alpha} - 1)$ where e is the base of Naperian logarithms.

The intensity of pressure p varies of course from point to point, and may be calculated from the formula:—

$$p = \frac{Pe^{\mu\theta}}{BR}$$

where θ is the angle measured from the slack end of the band, where the actuating force P is applied.

It will be appreciated that the differences of pressure are very considerable, and result in corresponding inequalities of wear round the rubbing surfaces.

Power Capacities.

If all the dimensions are measured in inches and forces in pounds, then the transmitted torque T is, of course, in pound-inches, and the corresponding horse-power at a speed of rotation N r.p.m. is:—

$$HP = \frac{TN}{63{,}000} \text{ nearly.}$$

It will be understood that the torque required to start up and accelerate machinery will commonly exceed that required to maintain it at its normal running speed and appropriate allowance should be made for this in clutch design.

Frictional Coefficients and Pressures.

The effective value of the coefficient of friction depends on the materials in contact, the state and finish of their surfaces, the presence of oil or contaminating matter and the pressure applied.

Reasonable working values under ordinary conditions of operation are as follows:—

Mating Surfaces.	*Coefficient of Friction.*
(a) *Dry.*	
Cast iron—cast iron	0·18
Mild steel—cast iron	0·25
Asbestos fabric—cast iron or steel	0·25
Wood—cast iron or steel	0·2
Leather—cast iron or steel	0·25–0·3
Cork—cast iron or steel	0·25
(b) *Lubricated.*	
Metal—metal	0·05–·010
Asbestos fabric—metal	0·125
Cork—metal	0·15

The permissible pressure depends largely on the intensity of heat transfer, and therefore varies inversely with the rubbing speed and frictional coefficient, diminishes with the frequency of operation and must be kept low in the case of materials, such as cork and leather, which are liable to char.

The permissible pressure is frequently determined from a relationship of the orm pv = constant or alternatively from a specified limit of horse-power transmitted per square inch of surface. For asbestos fabric linings a common rule is $pv = 50{,}000$, where p is the permissible pressure in lb./sq. in. and v the mean rubbing speed in feet per minute. Alternatively, manufacturers of clutch linings recommend that the power transmitted per square inch of lining should not exceed from ½ horse-power for clutches subject to frequent engagement to 1 horse-power for clutches which are only operated occasionally.

As a rough rule, over the range of service to which they are suited, other brake surfaces may be loaded to the following pressures: metal on metal 30 lb./sq. in., wood on metal 25 lb./sq. in., leather on metal 5–10 lb./sq. in. In America cone clutches are loaded considerably in excess of these figures.

The operating effort allowed in designing the engaging mechanism is of the order of 20 pounds for hand operation or 50 pounds for operation by foot.

See also Descriptive Section XXI., Part II.

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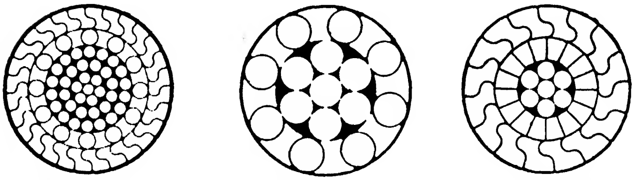
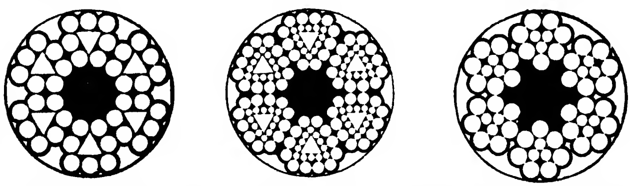
AND

WIRE ROPE MANUFACTURERS



Locked Coil Ropes of every Type. Patented by us in 1884.



Flattened Strand Ropes of every Type. Patented by us in 1888.

Head Office and Works:

Hay Mills, BIRMINGHAM, England

Telegraphic Address: **"Latch, Hay Mills."**

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Cable Codes: **A 1 and A B C (4th & 5th Editions). Marconi, Bentley's, Western Union, Hamilton's Rope Code, Private Code.**

SECTION XXI

PART III

WIRE ROPES

WIRE ROPES FOR ENGINEERING AND MINING PURPOSES —ROPES FOR GENERAL ENGINEERING PURPOSES

(Contributed by Norman Whincup.)

Under this heading is included wire rope for cranes, hoists, lifts, excavators, and grabs, for various branches of engineering, quarrying, shipping, etc. The various types available being illustrated in fig. 1 (a and b).

The process of wire drawing has advanced considerably during recent years, and manufacturers are now able to produce ductile wire of high tensile strength, capable of withstanding wear and abrasion over long periods. Danger of brittleness has been overcome to a very large extent, and rope wire is being regularly produced with a tensile strength of 110/120 tons per sq. in. for special work; in some cases, ropes are made from wire of 120/130 tons per sq. in.

The analysis of steel varies in accordance with the tensile strength required, as follows:—

	Per Cent.
Carbon	0·4 to 0·8
Silicon	0·05 to 0·15
Sulphur for Special Acid	0·04 max.
Ordinary Acid	0·05 max.
Phosphorus for Special Acid	0·04 max.
Ordinary Acid	0·05 max.
Manganese	0·45 to 0·65

BREAKING STRESSES.

Breaking Stress in Tons per Sq. In.	Trade Description.
80–90	Best Patent Steel.
90–100	Special Improved Patent Steel.
100–110	Best Plough Steel.
110–120	Special Improved Plough Steel.
120 and over	Extra Special Improved Plough Steel.

In selecting a wire rope for a purpose where wear and abrasion is likely to take place, wires of 100/110 tons per sq. in. have been found to give satisfactory results.

Lang's Lay and Ordinary Lay.—When ordering a wire rope some engineers are often in doubt as to whether or not the rope should be Lang's lay or ordinary lay. A Lang's lay rope is one in which the lay of the wires in the strands and the lay of the strands in the rope are in the same direction: whereas in ordinary lay ropes the wires in the strands are laid in one direction and the strands in the rope in the opposite direction (fig. 2, p. 936).

In a Lang's lay rope the wear is shared over a plurality of wires, and in an ordinary lay rope the wear at any section is taken by one or two crown wires in each strand. Thus the Lang's lay rope gives a longer life as regards wear, but it is not suitable for conditions where one end is free, on account of its pronounced tendency to untwist. Lang's lay rope can only be used where

ENGINEERING ROPES.

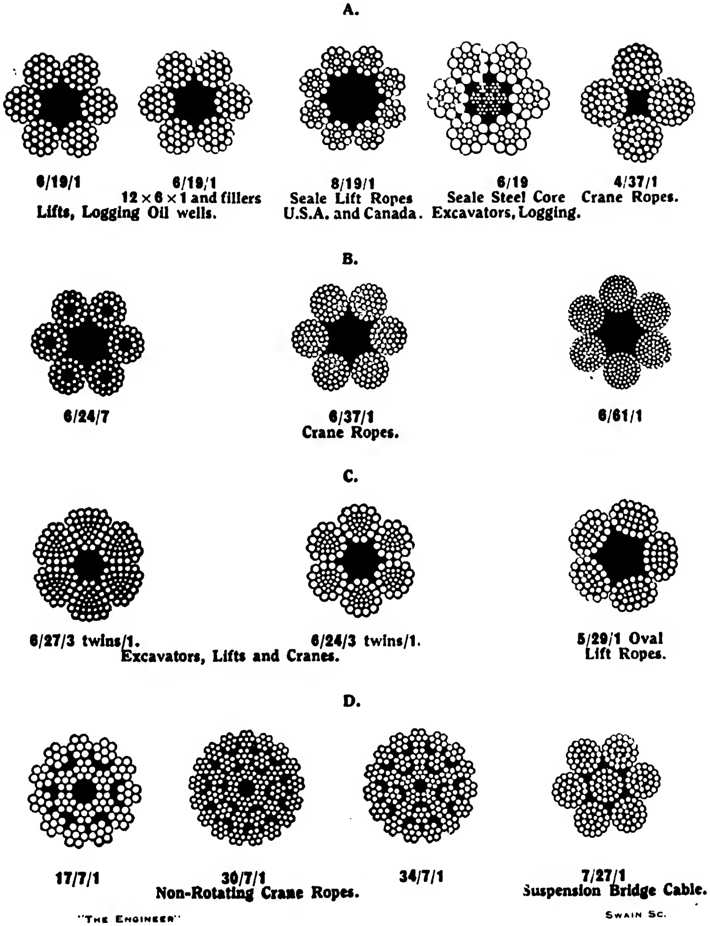
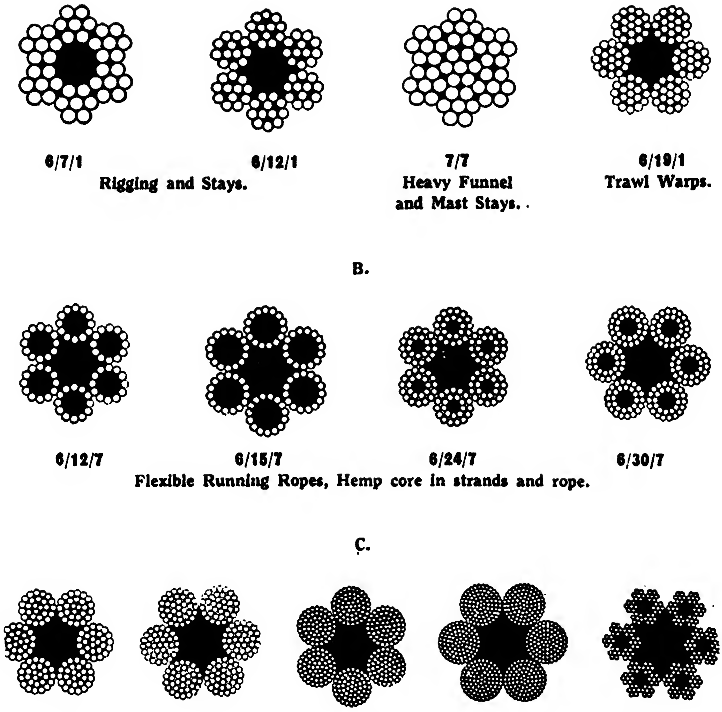


FIG. 1 (a).

SHIPPING ROPES.

A.



6/7/1 6/12/1 Rigging and Stays.

7/7 Heavy Funnel and Mast Stays.

6/19/1 Trawl Warps.

B.

6/12/7 6/15/7 6/24/7 6/30/7
Flexible Running Ropes, Hemp core in strands and rope.

C.

6/27/1 6/37/1 Flexible Running Ropes, Hemp main core.

6/61/1 6/91/1 Hauling up ropes for Slipway.

Tiller Rope.

"THE ENGINEER" SWAIN Sc.

FIG. 1 (b).

both ends are fixed, and in other circumstances ordinary lay is used or one of the types of non-spinning rope.

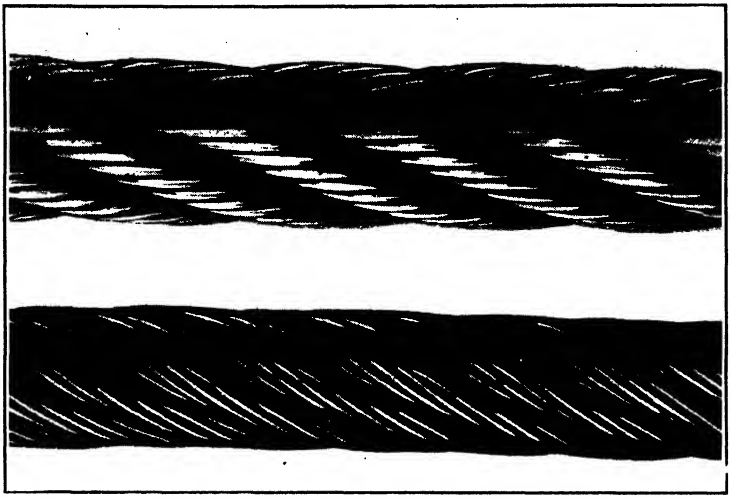


FIG. 2.—(Above) Ordinary Lay. (Below) Lang's Lay.

Crane Ropes.—The majority of these are constructed in 6/19/1 to 6/37/1 ordinary lay, depending on the size of the smallest pulley, which governs the degree of flexibility required. If, however, the hook is attached to a single free end, then one of the various types of non-spinning rope should be used, to prevent the load from spinning during lifting or lowering. If the crane operates over heat, such as molten metal, or over a furnace where excessive heat is present, the rope should have a wire core. If a rope works in wet or corrosive atmosphere it should be galvanised.

Crane ropes are usually of one of the following constructions :—

Round Strand	6/19/1, 6/24/7, 4/37/1, 6/37/1.
Flattened Strand	6/25/Δ, to 6/30/Δ according to flexibility required.
Non-Spinning	17/7/1, 30/7/1, 34/7/1, or of special oval strand construction.

MINIMUM SIZES OF PULLEYS AND DRUMS.

Crane Ropes from B.S.S. 302. Revised 1938.

Construction.	Size of Pulley.
4/37/1	8·5 × circ.
6/19/1	7·5 × circ.
6/24/7	7·0 × circ.
6/37/1	6·0 × circ.
6/61/1	5·5 × circ.

For each increase of rope speed of 100 ft. per min., 5 per cent. of the basic figure must be added to the diameter of the sheave.

The steel from which crane ropes are made should be one of the following qualities as specified by the purchaser.

Grade 'A'	Special Acid, or Swedish Quality
Grade 'B'	Acid Quality.

The wire used in the manufacture of ropes to this specification shall conform to one of the ollowing descriptions, as specified by the purchaser.

Tensile Breaking Strength in Tons per Sq. In.	Trade Description.
80–90	Best Patent Steel.
90–100	Special Improved Patent Steel.
100–110	Best Plough Steel.
110–120	Special Improved Plough Steel.

The fibre used for the central or main core of the rope shall be new, acid free, long fibre good quality hemp, manila or jute.

MECHANICAL PROPERTIES.

The trade designations and mechanical properties of the most important types of wire used in the construction of wire ropes are shown in the Table. The figures for torsions in 100 diameters are in accordance with the various British Standard Specifications concerned. Columns A r fer to the wire at the manufacturer's works before stranding and columns B to wire unstranded from the finished rope. This unstranded wire is required to give a strength not less than the lower limit of its specified range and, if its diameter exceeds 0·036 in., not in excess of the upper limit by more than 5 tons per sq. in. Tolerances in rope strengths are 5 to 7½ per cent. below the loads tabulated for the rope in the specifications.

Trade Description.	Tensile. Strength. Tons per sq. in.	Torsions in 100 diameters. Black. A	Black. B	Galvanised. Smaller. A	Smaller. B	Larger. A	Larger. B
Mild Patent Steel .	60–70	34	26	28	21	23	17
Patent Steel . .	70–80	34	26	28	21	23	17
Best Patent Steel .	80–90	34	26	28	21	23	17
Spec. Imp. Pat. Steel	90–100	34	26	28	21	23	17
Best Plough Steel .	100–110	32	24	23	17	19	14
Spec. Imp. Plough Steel . . .	110–120	30	23	15	11	11	8
	115–125	28	21	10	8	7	5

HOOKS.

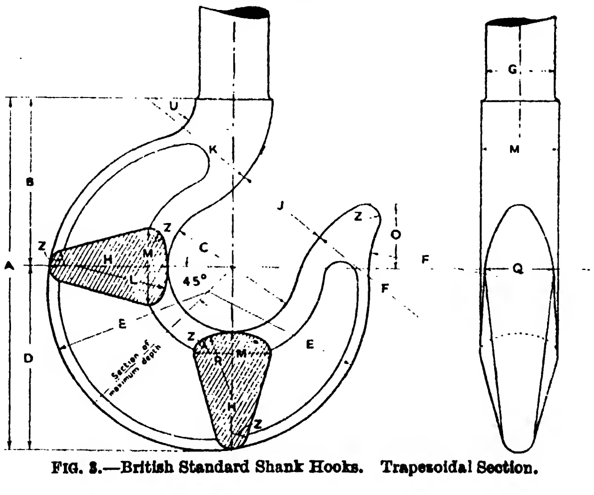


FIG. 3.—British Standard Shank Hooks. Trapezoidal Section.

Dimensions of British Standard Shank Hooks, Trapezoidal Section.

All dimensions in inches.

NOTE.—The metric equivalents of the dimensions given in this table may be obtained from B.S.S. No. 350 Conversion Tables (inches to millimetres and millimetres to inches).

W	C	A	B	D	E	F	G	H	J	K	L	M	O	Q	R	U	Z
W = Safe Working Load.	$C=1{\cdot}5\sqrt{W}$	A = 2·75 C	B = 1·31 C	D = 1·44 C	E = 1·25 C	F = 1·00 C	G = 0·55 C	H = 0·93 C	J = 0·75 C	K = 0·92 C	L = 0·70 C	M = 0·60 C	O = 0·50 C	Q = 0·54 C	R = 0·50 C	U = 0·30 C	Z = 0·12 C
Tons.																	
1/4	3/4	2 1/8	1	1 1/16	15/16	3/4	7/16	11/16	9/16	11/16	1/2	7/16	3/8	3/8	3/8	1/4	1/16
1/2	1 1/16	2 15/16	1 3/8	1 1/2	1 5/16	1 1/16	9/16	1	13/16	1	3/4	5/8	1/2	9/16	1/2	5/16	1/8
1	1 1/2	4 1/8	1 15/16	2 3/16	1 7/8	1 1/2	13/16	1 3/8	1 1/8	1 3/8	1 1/16	7/8	3/4	13/16	3/4	7/16	3/16
1 1/2	1 13/16	5 1/16	2 7/16	2 5/8	2 5/16	1 13/16	1	1 11/16	1 3/8	1 11/16	1 5/16	1 1/8	15/16	1	15/16	9/16	1/4
2	2 1/8	5 13/16	2 3/4	3 1/16	2 5/8	2 1/8	1 3/16	2	1 9/16	1 15/16	1 1/2	1 1/4	1 1/16	1 1/8	1 1/16	5/8	1/4
3	2 5/8	7 1/8	3 3/8	3 3/4	3 1/4	2 5/8	1 7/16	2 7/16	1 15/16	2 3/8	1 13/16	1 9/16	1 5/16	1 3/8	1 5/16	3/4	5/16
4	3	8 1/4	3 15/16	4 5/16	3 3/4	3	1 5/8	2 13/16	2 1/4	2 3/4	2 1/8	1 13/16	1 1/2	1 5/8	1 1/2	7/8	3/8
5	3 3/8	9 1/4	4 3/8	4 13/16	4 3/16	3 3/8	1 7/8	3 1/8	2 1/2	3 1/16	2 3/8	2	1 11/16	1 13/16	1 11/16	1	3/8
7 1/2	4 1/8	11 5/16	5 3/8	5 15/16	5 1/8	4 1/8	2 1/4	3 13/16	3 1/8	3 3/4	2 7/8	2 1/2	2 1/16	2 1/4	2 1/16	1 1/4	1/2
10	4 3/4	13 1/16	6 3/16	6 13/16	5 15/16	4 3/4	2 5/8	4 7/16	3 9/16	4 3/8	3 5/16	2 7/8	2 3/8	2 9/16	2 3/8	1 7/16	9/16
12 1/2	5 5/16	14 5/8	6 15/16	7 5/8	6 5/8	5 5/16	2 15/16	4 15/16	4	4 7/8	3 11/16	3 3/16	2 5/8	2 7/8	2 5/8	1 9/16	5/8
15	5 13/16	16	7 5/8	8 3/8	7 1/4	5 13/16	3 3/16	5 3/8	4 3/8	5 3/8	4 1/16	3 1/2	2 7/8	3 1/8	2 7/8	1 3/4	11/16
20	6 11/16	18 7/16	8 13/16	9 11/16	8 3/8	6 11/16	3 11/16	6 1/4	5	6 3/16	4 11/16	4	3 3/8	3 5/8	3 3/8	2	13/16
25	7 1/2	20 5/8	9 13/16	10 13/16	9 3/8	7 1/2	4 1/8	7	5 5/8	6 7/8	5 1/4	4 1/2	3 3/4	4 1/16	3 3/4	2 1/4	7/8
30	8 3/16	22 9/16	10 3/4	11 13/16	10 1/4	8 3/16	4 1/2	7 5/8	6 3/16	7 9/16	5 3/4	4 15/16	4 1/8	4 7/16	4 1/8	2 7/16	1
35	8 7/8	24 7/16	11 5/8	12 3/4	11 1/16	8 7/8	4 7/8	8 1/4	6 11/16	8 3/16	6 3/16	5 5/16	4 7/16	4 13/16	4 7/16	2 11/16	1 1/16
40	9 1/2	26 1/16	12 7/16	13 11/16	11 7/8	9 1/2	5 3/16	8 13/16	7 1/8	8 3/4	6 5/8	5 11/16	4 3/4	5 1/8	4 3/4	2 7/8	1 1/8
50	10 5/8	29 3/16	13 7/8	15 1/4	13 1/4	10 5/8	5 13/16	9 7/8	7 15/16	9 3/4	7 7/16	6 3/8	5 5/16	5 3/4	5 5/16	3 3/16	1 1/4
60	11 5/8	31 15/16	15 1/4	16 3/4	14 1/2	11 5/8	6 3/8	10 13/16	8 11/16	10 11/16	8 1/8	7	5 13/16	6 1/4	5 13/16	3 1/2	1 3/8
70	12 9/16	34 1/2	16 7/16	18 1/16	15 11/16	12 9/16	6 7/8	11 11/16	9 7/16	11 9/16	8 13/16	7 1/2	6 1/4	6 3/4	6 1/4	3 3/4	1 1/2
75	13	35 3/4	17	18 11/16	16 1/4	13	7 1/8	12 1/16	9 3/4	11 15/16	9 1/16	7 13/16	6 1/2	7	6 1/2	3 7/8	1 9/16

Note.—The proportion of each dimension in terms of C is indicated below each reference letter.

Hooks should be forged from mild steel complying with the requirements for class 'A' steel in British Standard Report, No. 24, part 4, specifications 8 and 9, or best Yorkshire iron to B.S.S. No. 51, and annealed after forging by uniformly heating in a furnace to a temperature between 980° C. and 1030° C. for wrought iron and 880° C. and 930° C. for mild steel, and then allowed to cool in still air. Annealing should be done periodically, at intervals of about six months for a hook in constant use lifting the full load or subject to heat and at intervals of twelve months for hooks on ordinary workshop cranes.

Crane hooks hanging from large travelling cranes in workshops should be painted white so as to be easily seen by the workmen.

Fig. 3 and the accompanying table give the proportion of standard trapezoidal section hooks and are based on the formula :—

$$C = 1 \cdot 5\sqrt{W}$$

Where C is the internal diameter in inches and W the working load in tons.

The diameter of the shank measured at the bottom of the thread if screwed should be proportioned for a stress of about 3 tons per square inch.

For further particulars refer to B.S.S. No. 482—1933, revised 1945.

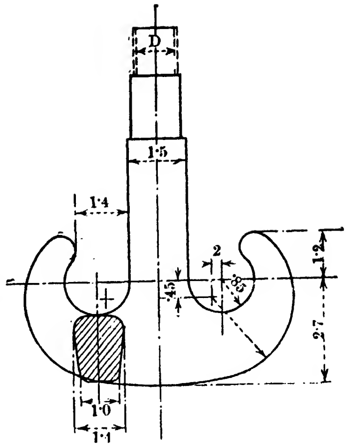


FIG. 4.

Fig. 4 gives the proportion of ramshorn hooks based on the minimum diameter of the shank as unity.

Long ladle hooks are sometimes made of a series of shaped plates about 1 in. thick riveted together to the required thickness for strength and fitted with a gunmetal pad in the saddle of the hook. This construction is considered more reliable than a forging and is cheaper to manufacture.

Lift and Hoist Ropes.—These are of numerous constructions, depending on the size of the pulley and other conditions of working. They are usually in Lang's lay construction because invariably both ends of the rope are fixed, and there is little danger of twisting under load.

They are usually one of the following constructions :—

6/12/7, 6/19/1, 6/19/1 Seale's, 6/24/7, 8/19/1 Seale's, 5/29/1 Oval and 6/37/1. In practice ropes with a large proportion of fibre core give best results on 'V' grooved driving pulleys, and they are more readily deformed to fit the groove than the more solid constructions, 8/19/1, 6/24/7 and 5/29/1 oval being examples.

MINIMUM SIZES FOR PULLEYS AND DRUMS.

6/12/7 = 13 × circ.	6/19/1 Seale's = 15 × circ.
6/19/1 = 13 × circ.	6/24/7 = 13 × circ.

Excavator Ropes.—Ropes for excavators are subjected to very severe duty. They are required to withstand wear, abrasion, and severe stresses, and it is almost impossible to keep them well lubricated. Hence a long life is difficult to secure.

The usual construction is 6/19 Seale's over a wire rope core, or in flattened strand construction of 6/25/Δ, the pulleys being similar to those for crane ropes.

In a flattened strand rope the centre of each strand may be a solid triangle, 3 or 6 wires, or plaited core.

In a Seale's lay rope the strands have two layers of wires round a centre wire ; the outer and inner layers consist of the same number of wires. The outer wires lie in the grooves of the inner ones, and are stranded at one operation. They are consequently in line contact throughout, and there is no internal cross-cutting.

DESIGN OF BARRELS AND PULLEYS FOR WIRE ROPES.

The bottom of a groove for the pulley supporting a rope should be a true arc of a circle for a distance equal to one-third of the circumference of the rope, and the radius of the groove should be larger than the radius of the rope by not less than the following amounts :—

2 in. circumference and under	$\frac{1}{32}$ in.
2¼ in. to 2¾ in. circumference	$\frac{3}{64}$ in.
3 in. to 3½ in. circumference	$\frac{1}{16}$ in.
3¾ in. and over	$\frac{3}{32}$ in.

The pulleys should be grooved to a depth equal to one and a half times the diameter of the rope, and the groove should be smoothly finished.

The angle of flare of the sides of the pulley groove should be 52°. Barrel grooves should be of the same radii as the pulley grooves, but should have a depth not less than one-third of the diameter of rope, and should be so pitched that there is a clearance of not less than $\frac{1}{16}$ in. between the parts of the rope when coiled on the barrel. The groove should be smoothly finished, and the edges should be rounded.

For grooved drums and pulleys the angle of lead should not exceed 1 in 12, or 5°. For plain drums, which are detrimental to the life of the rope, this angle should not exceed 2° on each side of the centre line, anything in excess causes a side pressure and abrasion on adjacent laps, and uneven coiling. (B.S.S. No. 302.)

The above remarks apply to the idler pulleys on lifts and hoists.

Traction Sheaves.—The included angle for the flare of straight-sided V-grooves of traction sheaves should not exceed 42°, or be less than 35°. There should be sufficient metal provided at the root of the grooves to allow the sheave to be re-turned in the wearing surface, so as to permit the reduction of the diameter of the sheave to an extent equal to 75 per cent. of the diameter of the rope. (B.S.S. No. 329.)

Durability of Wire Ropes.

The two most importent conditions appertaining to the manufacture and use of steel wire ropes that affect their durability are:—

(a) Quality of material and size of wire.
(b) Diameter of pulleys and arrangement of ropes.

(a) Wire used for lifting ropes is of steel whose ultimate tensile strength varies from 80 to 130 tons per square inch. Ropes of material having a higher tensile strength are of smaller diameter for a given load and factor of safety, but since they are stiffer, larger drums are necessary to obtain the same durability.

The stress in a wire due to bending round a pulley is proportional to

$$\frac{(\text{Modulus of elasticity}) \times (\text{Diameter of wire})}{(\text{Diameter of pulley})}$$

hence large wires are more quickly fatigued than small ones. On the other hand, small wires are more quickly worn through.

For a given ratio of pulley to rope diameter, given rope diameter and load, the life of the rope, if limited by abrasion and not by fatigue, increases as the square of the diameter of the wires of which it is made.

(b) For a given diameter of rope (d), diameter of wire (a) and load (P), the life of the rope depends on the square of the ratio of pulley to rope diameter (m) for large values of the latter.

The above statements have also been deduced from theoretical considerations by Prof. J. T. Nicolson, D.Sc., and may be represented by the formula given by him.

$$\text{Life of rope} = C\frac{m(m+1)d^2a^2}{P}$$

where the symbols m, d and a have the meanings given above, and C is a constant.

The practice of passing wire ropes on a crab as shown by fig. 5, entailing successively a 90° bend, a 180° reverse bend, and a 90° bend, is a bad one and deteriorates the strand wires;

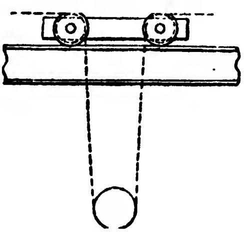


Fig. 5.

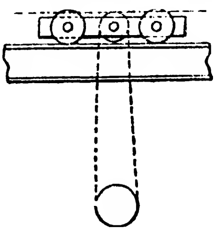


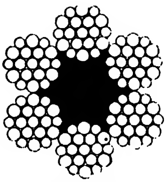
Fig. 6.

a better method is shown by fig. 6, the rope being taken over one upper pulley, down to the block, and up over the second pulley, the bend being always in the same direction, the reverse bend being thus abolished; this arrangement will increase the life of a rope as much as 50 per cent.

Oil Well Ropes.—Wire ropes are largely employed in drilling oil wells, particularly when the American system of drilling is in use, which is known as the Californian Cable Rig. This system uses a drilling line for manipulating the tools, a casing line for the casing, and a sand line for clearing out the walls. The two former are generally 6/19/1 Seale's or filler wire construction, and the latter 6/7/1. Although several constructions are laid down in B.S.S. 366, as follows:—

6/7/1, 6/12/1 laid 9 over 3, 6/19/1, 6/19/1 filler, 6/19/1 Seale's, 6/19/1 Warrington, 6/27/1 and 6/37/1.

Oil well drilling is very severe, and calls for ropes of a high quality in preferably acid grade steel. Galvanised ropes are sometimes advisable on account of the corrosive action of the water in the wells. Drilling lines are left hand lay unless otherwise specified by the purchaser.



6 x 19
Warrington

FIG. 7.

Pulley Sizes.—Diameter of sheaves and drums should be as large as possible and in no case should be less than the following measures at the bottom of the groove.

6/7/1 and 6/12/1	33 × dia. of rope.
6/19/1 and 6/27/1	20 × dia. of rope.
6/37/1	18 × dia. of rope.

The wires in oil well ropes call for special tests, for which reference should be made to B.S.S. 366—1929.

Flattened Strand Ropes.—For general engineering work and mining work a most useful construction is made up of strands in sectional outline, somewhat like an equilateral triangle with the corners rounded off. Each strand presents a flattened side outwards throughout the whole length of the rope, and, in consequence, a larger wearing surface is always exposed. In addition the construction gives an increased breaking strength and wearing surface, without increased diameter of the rope, this being due to the fact that they are more compact, and the area of steel is greater than that of an ordinary round strand rope of the same diameter.

Flattened Strand Ropes splice very effectively, as the core is slightly smaller than the strands, and there is less risk of the splice drawing, but the tucks should be carefully made, by reducing the size of the strands, so that the surface of the splice does not present a lumpy and uneven appearance, otherwise premature wear of the wires in the splice will take place.

Tru-Lay Ropes.—In this type of rope the strands are preformed to their final helical pitch before being laid up into the rope. Ropes made by this process show no tendency to unstrand when cut, and the wires lie in the natural formation without the ends having to be served. In fact, a piece of Tru-lay rope can be unwound strand by strand, and then reassembled with the strands in their relative positions, without using excessive force. The life obtained from this type of rope is far in excess of that of ordinary ropes of equal sectional area under reverse bending stresses. Should the outer wires become broken through long wear, they remain in position and do not spring out, as in ordinary ropes. Some engineers do not appreciate this feature, as broken wires are extremely difficult to detect, and give rise to a false impression of the condition of the rope. In any case, this type of rope has found popularity in America, the country of its invention, and is strongly recommended for all purposes where round strand ropes are used, particularly in cases where small drums and pulleys exist. In fig. 8 a preformed rope is illustrated.

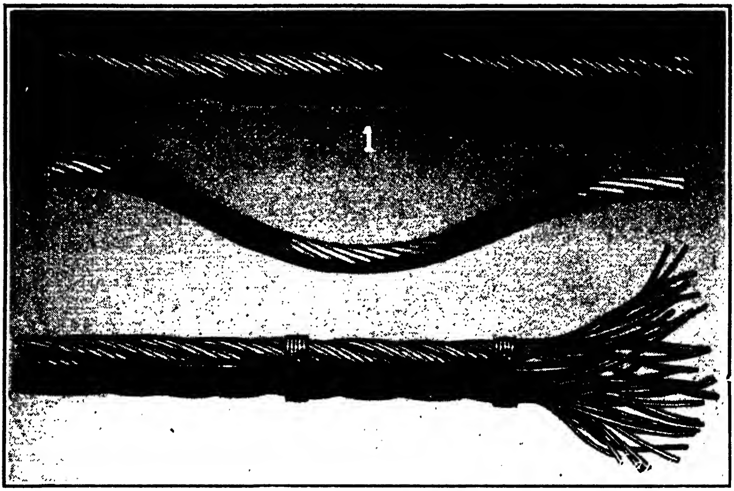


FIG. 8.—Preformed Rope.

The illustration shows :—

(1) A piece of Tru-Lay rope with a strand removed, showing how the remaining strands remain *in situ*.
(2) The strand itself, showing how the normal formation is retained.
(3) An ordinary rope when cut; the ends have opened out, and the ends frayed. Such a rope as this requires a wire serving at each end.

Shipping Ropes.—Tables No. 51, 53, 54, 55 and 56, published by Lloyd's Register of Shipping, lay down for sailing ships and steam vessels, steam trawlers and tugs, the minimum size, breaking load, and construction of rigging, hawsers, tow lines, and warps for various tonnages, and reference to the tables gives most of the information required for ships' outfits of wire ropes.

Also reference should be made to B.S.S. 365—1929, revised 1942.

Non-Rotating Ropes.—This type of rope has been adopted in many cases where a strictly non-rotating rope is required. Such as on cranes with a single free end, sinking, winding, and for mine cage balance ropes, also for Koepe system of winding.

Constructions vary with different makers, and accepted standards are amongst the following :—

17/7/1, 30/7/1, 34/7/1, or special oval strand construction over a concentric wire core, or round strands over a concentric wire core.

Mining Ropes

Winding.—In selecting a winding rope to suit any particular installation, the following details must necessarily be considered: Diameter of drum and overhead sheave; depth of shaft; load to be hoisted; kinetic shocks due to acceleration and deceleration; bending stresses; cage clearance and shaft side clearance; surface of drum (wood lagged or steel); whether or not the rope winds in multiple layers; whether or not water is present; factor of safety; type of guides and rubbing ropes; fleet angle.

In medium depth and shallow pits, round strand or flattened strand work quite satisfactorily, but, unless these are made on the Preformed Dead Lay principle, they sometimes give trouble on account of their pronounced tendency to untwist, causing excessive wear on the guides and rubbing ropes. Where the cage clearance is small, the twisting might cause fouling of the cages or contact with the shaft walls. These troubles do not exist where fixed guides are in use. Twisting

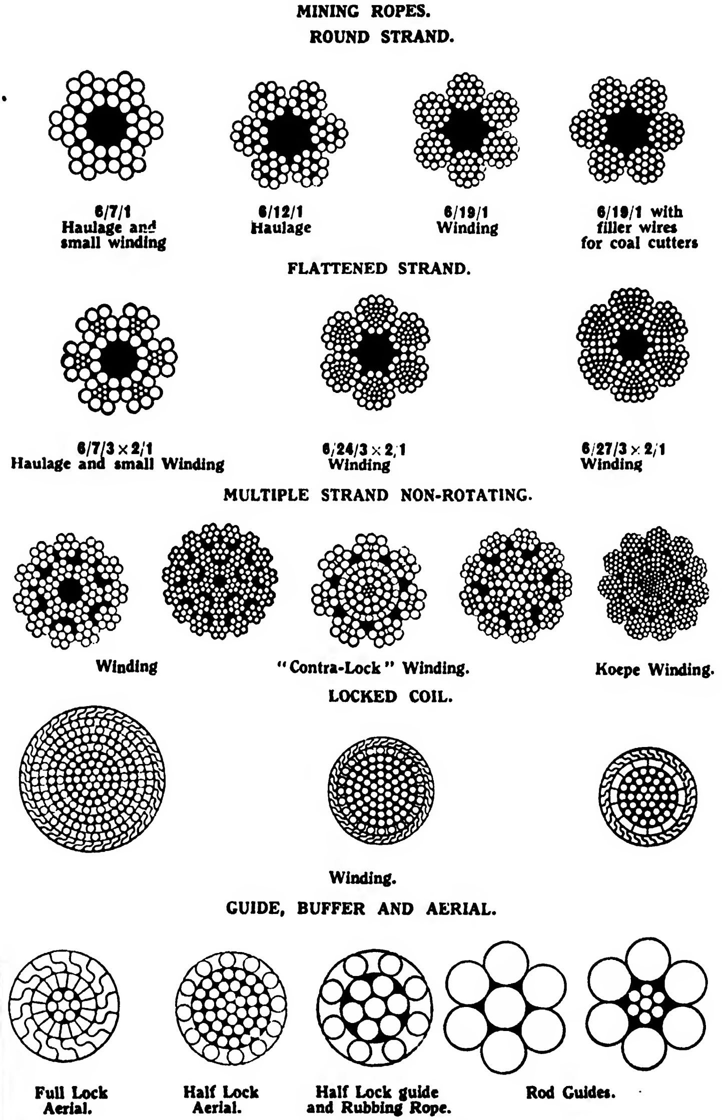

MINING ROPES.
ROUND STRAND.
6/7/1
Haulage and
small winding
6/12/1
Haulage
6/19/1
Winding
6/19/1 with
filler wires
for coal cutters
FLATTENED STRAND.
6/7/3 × 2/1
Haulage and small Winding
6/24/3 × 2/1
Winding
6/27/3 × 2/1
Winding
MULTIPLE STRAND NON-ROTATING.
Winding
"Contra-Lock" Winding.
Koepe Winding.
LOCKED COIL.
Winding.
GUIDE, BUFFER AND AERIAL.
Full Lock
Aerial.
Half Lock
Aerial.
Half Lock guide
and Rubbing Rope.
Rod Guides.

FIG. 9.

of ropes, however, due to winding with slack chains, often causes broken wires at the capel end, equally as much as lack of lubrication and consequent corrosion fatigue, when moisture runs down the rope towards the capel end.

For the deeper pits, from which heavy loads are hoisted, locked coil or non-spinning ropes give best service, because the greatest breaking load can be obtained for the minimum of diameter of rope, and the elongation is very much less than that of round strand or flattened strand ropes of equal diameter. In cases where a rope winds on itself in multiple layers, locked coil ropes give best service. They are absolutely free from any tendency to rotate, and therefore are specially suitable for sinking purposes, and winding from great depths. In the event of a wire breaking in service, it is held in position and does not unravel and lay across the adjacent wires. If it is necessary to repair a broken wire, this can be readily done by the makers on site.

The full lock wires on the cover of a locked coil rope have an overlap which provides a seal against the ingress of corrosive matter, and the escape of internal lubricant, there is, therefore, less danger of internal corrosion than in any other type of rope—the core being packed with special rust resisting lubricant during manufacture as each successive layer of wires is layed up. The wear is lighter, as they provide a smooth wearing surface.

In all cases of wet and damp pits, and those in which corrosive fumes are present, anti-corrosive ropes should be used, and adequate attention should be given to frequent periodical lubrication.

Pulley Sizes.—The construction of a winding rope is largely dependent on the existing drum and pulley sizes, depth of pit, and load to be hoisted. In the case of round strand and flattened strand ropes the wires should not be greater than the $\frac{\text{drum diameter}}{1{,}200}$ in order to provide a reasonable bending stress, in accordance with accepted practice. However, the larger the pulley, the longer the expected life of the rope, and some investigators say that a winding rope should not be bent round a pulley of less than 100 rope diameters, others put this ratio at 120. In the case of electric winders, however, this ratio is often as small as 90 diameters of rope, and in these cases the construction has to be made to suit, sometimes with detrimental results.

The following is a useful guide :—

Wire rope should never be bent round a drum or pulley whose diameter is less than 100 D, or 1,000 *d*; 150 D, or 1,500 *d*, are preferable, and the respective co-efficients 120 and 1,200 are quite frequently employed. In Belgium, diameters of pulleys greater than 750 *d* for flat ropes, or greater than 1,000 *d* for round ropes, are allowed. In the Pas de Calais the following proportions of drum diameter have been adopted : for Lang's lay, greater than 68 D, for flattened strand, greater than 72 D, and for lock coil, greater than 80 D. Where D = rope diameter and *d* = wire diameter. (J. F. Perry, *Mining Engineer*, vol. iv, No. 34, p. 24.)

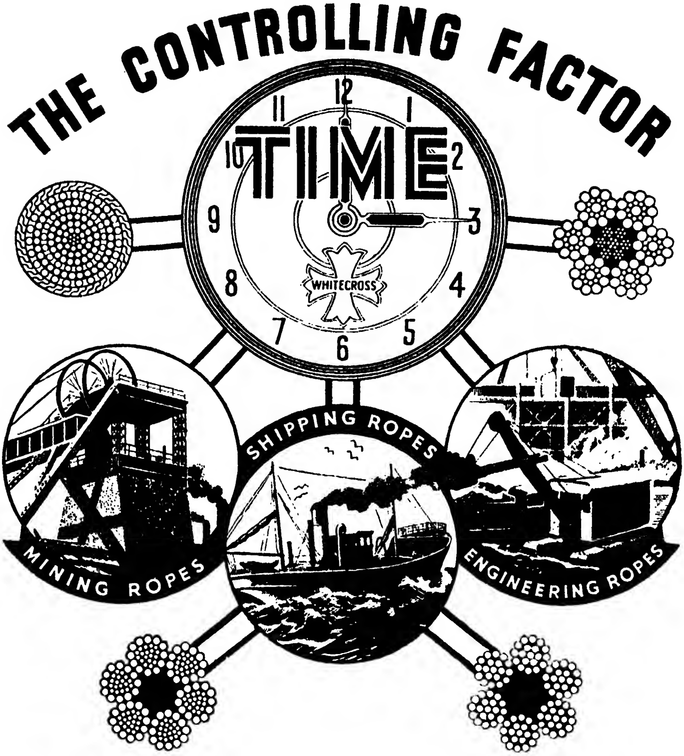
Wires in round strand and flattened strand winding ropes vary from about 0·128 in. diameter to about 0·040 in. diameter, but up to 0·140 dia. is used in some single constructions when the drum is suitable.

Designs normally work out as follows :—

⅞ in. to 1⅛ in. diameter	6/7/1 construction.
1⅛ in. diameter to 1¼ in. diameter	7 or 8 outer wires per strand.
1⅜ in. diameter to 1½ in. diameter	9 or 10 ,, ,, ,, ,,
1½ in. diameter and over	12 or more outer wires per strand.

The rule for diameters of outer wires is shown in table below :—

Round Strand.		Flattened Strand.	
No. of Outer Wires.	Size of Outer Wire.	No. of Outer Wires.	Size of Outer Wire.
	Ins.		Ins.
5	Circumference × 0·0397	7	Circumference × 0·0353
6	,, × 0·0353	8	,, × 0·0320
7	,, × 0·0320	9	,, × 0·0290
8	,, × 0·0290	10	,, × 0·0264
9	,, × 0·0264	11	,, × 0·0245
10	,, × 0·0245	12	,, × 0·0227
11	,, × 0·0227	13	,, × 0·0212
12	,, × 0·0212	14	,, × 0·0198
15	,, × 0·0175	15	,, × 0·0185
18	,, × 0·0150	16	,, × 0·0175
24	,, × 0·0115	17	,, × 0·0165
		18	,, × 0·0158



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Approximate weight of wire ropes in lbs. per fathom where
C = the circumference.

Round strand with hemp core	$= 1 \cdot 031$ lbs. $\times C^2$.
Round strand with wire core	$= 1 \cdot 145$ „ $\times C^2$.
Flattened strand with wire core	$= 1 \cdot 134$ „ $\times C^2$
Locked coil	$= 1 \cdot 5$ „ $\times C^2$
Half-locked guide rods	$= 1 \cdot 5$ „ $\times C^2$
„ „ aerial ropes	$= 1 \cdot 5$ „ $\times C^2$
Round steel guide ropes	$= 1 \cdot 34$ „ $\times C^2$

The approximate breaking loads of wire ropes :—

Ordinary ropes 6/7/1 . . .	Circumference²	× 3·24 at 80/90	tons per sq. in.
	„	× 3·62 „ 90/100	„ „ „
	„	× 4 „ 100/110	„ „ „
	„	× 4·38 „ 110/120	„ „ „
Round strand, compound construction	„	× 2·9 „ 80/90	„ „ „
	„	× 3·23 „ 90·100	„ „ „
	„	× 3·55 „ 100/110	„ „ „
	„	× 3·91 „ 110/120	„ „ „
Flattened strand	„	× 3·27 „ 80/90	„ „ „
	„	× 3·64 „ 90/100	„ „ „
	„	× 3·97 „ 100/110	„ „ „
	„	× 4·28 „ 110/120	„ „ „
Locked coil	„	× 4·33 „ 90/100	„ „ „
	„	× 4·77 „ 100/110	„ „ „
	„	× 5·27 „ 110/120	„ „ „

From the aggregate breaking load of the total number of wires in a rope, the following deductions should be made to determine the actual breaking load :—

		Per Cent.
6/7/1 Round strand	deduct	7½
6/19/1 „ „	„	12½
6/24/7 „ „	„	12½
6/37/1 „ „	„	17½
6/61/1 „ „	„	22½
Non-spinning ropes	„	20
6/7/Δ flattened strand	„	7½
Up to 6/28/Δ flattened strand . .	„	12½
Locked coil winding ropes . . .	„	16
„ „ aerial ropes . . .	„	10
Concentric strands	„	7½
7 wire guide rods	„	5

Bending Stresses.

Hrabak gives the following formulæ for bending stresses in wire ropes :—

' Where E is modulus of elasticity = 28,500,000 lbs. per sq. in.
K is a constant 0·44 for round strand ropes, 0·6618 for a strand, and 0·2918 for cables.
***d* is the diameter of the wire.**
D is the diameter of the pulley.'

Then bending stress $= \dfrac{Ed}{D} \times K.$

Tests have been taken on various ropes and for a flattened strand rope of six strands of 10/8/Δ, 1⅜ in. diameter, the modulus of elasticity for the complete rope was 13,650,000 lbs. per sq. in.

For a round strand of 6/7/1, 1¼ in. diameter	17,500,000	lbs. per sq. in.
„ „ „ „ 7/7/– 3 in. circumference	21,000,000	„ „ „
„ locked coil . . 1⅜ in. diameter	16,450,000	„ „ „

It will be seen that there is a considerable difference in the modulus of elasticity for various constructions, as compared with the formulæ used by Hrabak. Other opinions are set out below :—

The maximum bending stress in lbs. per sq. in. produced by bending a rope round a pulley of diameter Δ is theoretically $E\dfrac{d}{\Delta}$, **where E is the modulus of elasticity of the wire used; it ranges from 10,000,000 to 30,000,000 lbs. Owing to the strain not acting directly at right angles**

to the cross-seciton of the whole rope, it is found in practice that the mean stress is more correctly represented by the expression $\frac{3}{8} E \frac{d}{\Delta}$. According to R. W. Chapman (*Proc. Austr. Inst. Min. Eng.*, vol. iii., No. 3, April 1908), if a is the angle of pitch of the wires in a strand and b of the strands in a rope, the bending stress is $E \frac{d}{\Delta} \cos^2 a \cos^2 b$.

If s be the mean bending stress as above, T the breaking strength in lbs. per sq. in., m the factor of safety (ratio of total breaking load to total working load), W the total load to be carried, and A the total cross-section of steel in the rope, $A = \frac{mW}{T - s}$.

For ordinary winding ropes $A = 0 \cdot 6 D^2$ approximately.

Coefficients of friction of wire rope :—

Dry rope on a grooved iron drum . .	0·120	Greased rope on a wood filled sheave	0·140
Wet ,, ,, ,, ,, . .	0·070	Dry rope on rubber leather filling .	0·495
Greased ,, ,, ,, . .	0·70	Wet ,, ,, ,, ,, . .	0·400
Dry rope on a wood filled sheave . .	0·235	Greased ,, ,, ,, ,, . .	0·205
Wet ,, ,, ,, ,, . .	0·170		

According to G. Raw (*Trans. Inst. Min. Eng.*, iv., p. 180) the coefficient of friction of a slightly oiled flattened strand rope upon an elm-filled sheave, fibres on end, is between 0·3 and 0·35.

The resistance to bending of a rope under tension T lbs. round a pulley of radius r ft. $= 1 \cdot 26 + 0 \cdot 002276 \frac{T}{r}$ (*Weisbach*). The maximum admissible stress in a medium wire rope may be taken as averaging 5,000 times the weight of 1 ft. of the rope.

Elongation of a Suspended Rope.—The elongation of a suspended rope may be worked out from :—

$$\frac{L}{2AE} \times (2W + W_r)$$

Where L = Suspended length of rope.
W_r = Weight of suspended rope.
A = area of wires.
E = modulus of elasticity.
W = load to be hoisted.

The acceleration stress in a vertical shaft may be calculated from the following :—

Where A = Acceleration stress.
W = Weight of cage, tubs and coal.
w = Weight of suspended rope.
t = Acceleration time in secs.
V = Velocity of rope in ft. per sec.
g = Acceleration due to gravity.
s = Distance moved through in ft.

$$\text{Acceleration in ft. per sec.} = \frac{V}{t} = \frac{V^2}{2s} = \frac{2s}{t^2}$$

$$A = (W + w)\left(1 + \frac{V}{gt}\right) \text{ or } \frac{2s(W + w)}{gt^2} + (W + w) \text{ or } \frac{V(W + w)}{gt} + (W + w) \text{ or }$$

$$\frac{(W + w)V^2}{2gs} + (W + w)$$

For inclined shafts :—

$$A = \left\{(W + w) \operatorname{Sin} \theta + \left(\frac{W}{40} + \frac{w}{20}\right) \operatorname{Cos} \theta\right\} \left(1 + \frac{V}{gt}\right)$$

Where θ = Angle of inclination of shaft with the horizontal.
$\frac{1}{40}$ = Coefficient of friction for load.
$\frac{1}{20}$ = Coefficient of friction for rope.

Capping of Winding Ropes.—Various methods of capping are employed, and a large variety of capels are used, apparently according to the individual ideas of the management. The most popular appear to be the White Metal Capel and the Reliance type, both of which are very efficient.

Recommended Method of Socketing.—The metal shall conform to the following analysis :—

Tin	5 per cent. ± 0·25 per cent.
Antimony	15 per cent. ± 0·50 per cent.
Lead	The remainder.

The metal shall be free from zinc and the total impurities shall not exceed 0·20 per cent.

The rope to be socketed should be securely bound with suitable soft iron seizing wire for a length of not less than two rope diameters commencing at a distance from the end of the rope equal to the length of the conical portion of the socket less one-half diameter of the rope (plus allowance for turning over the wire ends to form hooks).

After threading the rope through the socket the rope end should be unlaid, the fibre core, if any, removed, and each individual wire separated out so that the rope end resembles a brush. Care should be taken that the outer wires are not bent too sharply over the end of the binding.

The 'brush' end should be cleaned with petrol or other suitable solvent to remove all dirt and grease, care being taken to avoid saturating the remainder of the rope with solvent and thereby removing the internal dressing and exposing the wires to corrosion. The cleaning may be accomplished either by immersing the brush in solvent, or by wiping the wire with cloths or waste soaked in solvent. In any case the individual wires should afterwards be wiped dry with clean cloths or waste.

The cleaned 'brush' end should be drawn into position in the conical socket with a length of seizing equal to one-half the diameter of the rope projecting into the narrow end of the socket and the ends of the wires flush with the end of the basket. The ends of the wires should be even.

The socket complete with the rope in place should be clamped in a vertical position with the large end of the socket uppermost, and the rope should be in axial line with the socket for a distance of not less than 24 rope diameters.

The junction of the rope and socket should be tightly served with asbestos yarn to prevent escape of the molten white metal.

The socket should be gradually and evenly heated all round the outside circumference by a blow lamp, care being taken to avoid undue local heating, and particularly any heating of the rope outside the socket. The socket should be at a uniform temperature of about 212° F. immediately before pouring the molten white metal. Heating the socket is essential to the free flow of the white molten metal; undue heating may impair the strength of the rope wires.

When the socket is at the correct temperature, and immediately before pouring the molten metal, powdered rosin should be dusted among the wires in the socket basket.

The metal used shall be melted from new ingots of the composition laid down in British Standard Specification No. 643—1935. It shall be poured at a temperature of 660° F. ± 25°, *i.e.*, between 635° and 685° F. Dross should be removed from the surface of molten metal and clean bright fluid metal only poured into the prepared heated socket. The pouring ladle should be of sufficient capacity to hold the full amount of metal to fill the socket, and should be heated before use. The temperature of the metal should be taken when in the ladle immediately before pouring. Pouring should be continuous and uniform until the metal completely fills the socket, and when the surface of the metal sinks in the centre, a little metal should be poured in from the ladle. In no circumstances shall the metal stand proud of the top edge of the socket.

The temperature of the metal should be determined by a thermometer, and overheating of the metal must be avoided because excessive temperature will damage the rope wires and endanger the safety of the completed socket. On the other hand, the metal must not be too cold when poured or it will fail to penetrate between the wires and make a satisfactory solid cone.

After pouring the metal, the socket should be allowed to cool gradually and should remain undisturbed until the metal has fully set and the socket has cooled to air temperature. The rope adjoining the socket should then be carefully cleaned and treated with preservative dressing. (B.S.S. No. 643—1935.)

Note.—In the case of wire ropes of locked coil construction well fitting clamps, at least three in number, must be fixed as near as possible to the end of the rope, in order to prevent the outer covering wires slipping upon the core.

In addition, before the wires in the end of the rope are disturbed, the socket must be threaded on to the rope, for which purpose the extreme end of the rope must have a wire serving to prevent the wires spreading out when the end is cut.

Club End Method.

According to this method a cone is formed with turned back wires. A conical seizing of soft wire is wound at a distance from the end of the rope equal to the length of the socket, or alternatively a cone may be found by split conical wedge sleeves. The wires of the rope end are unstranded, one-third are cut to half the socket length, one-third to three-quarters, and the remainder are left uncut.

The wires are then bent back over the seizing or cone, the shortest wires first; next a seizing wire is wrapped over the whole of the cone so formed in order to produce a smooth cone fitting the socket. This method is frequently used with split sockets and rings. In this case the rings are threaded on to the rope before the rope is unwrapped and the socket is forced on after the cone is complete, the eye being heated only sufficiently to allow the jaws to be opened.

Thimble Splices.

Particulars of thimbles suitable for bare wire rope not previously served will be found in B.S.S. 464—1932, together with recommendations as to splicing procedure, which requires skill and experience. A splice for a six-strand rope should have at least five tucks, three with the whole strand and two with half the wires cut out of each strand. The tucks are made under and over against the lay of the rope. After completion the splice should be served throughout its length.

Clamps.

In some cases the eye round a thimble is formed by clamping. Special U-bolt clips are provided for this purpose with shaped bridge-pieces. Three to six of these clips should be employed and they should be fitted with the U-bolt against the dead end of the rope and the shaped bridge-piece against the working end, otherwise the rope may pucker.

Proportions of white metal sockets may be calculated from the following :—

Where A = External diameter of base of socket.
a = Internal ,, ,, ,, ,, ,,
B = External ,, ,, top ,, ,,
b = Internal ,, ,, ,, ,, ,,
C = Length of taper of basket.
d = Diameter of pin.
L = Distance between centres of jaws.
M = Bursting force.
N = Internal area.
P = Internal pressure on basket walls ignoring friction.
Q = Mean tensile strength of material.
W = Required breaking load of socket. (Ultimate strength.)
α = Angle of internal taper.

Then

$$\operatorname{Tan}\alpha = \frac{b-a}{2C}$$

$$M = \frac{W}{\operatorname{Sin}\alpha} \quad \text{tons.}$$

$$N = \pi\frac{(b+a)}{2} \times C \quad \text{sq. in.}$$

$$P = \frac{M}{N} \quad \text{per sq. in.}$$

$$A = a\sqrt{\frac{Q+P}{Q-P}}$$

$$B = b\sqrt{\frac{Q+P}{Q-P}}$$

Capel Pin.

Bending moment $= \dfrac{WL}{12}$ for an evenly distributed load.

Z = modulus of section of pin $= \dfrac{WL}{12Q}$ $Z = \dfrac{\pi d^3}{32}$

Cylindro Conical Drums.—Trouble with winding ropes is sometimes experienced in the adjustment of length of ropes on bi-cylindro conical drums when a new rope is installed. If this is too long or too short, the amount of adjustment is done more or less by rule of thumb methods, which are only approximate, often difficulty is experienced due to elongation of the new rope, which upsets the winding length.

The solution of the following equations give the necessary adjustment accurately :—

Suppose that when cage B (fig. 11) is on the pit bottom, cage A is a distance a above the bank level, and that when cage A is on the pit bottom, cage B is a distance b above the bank

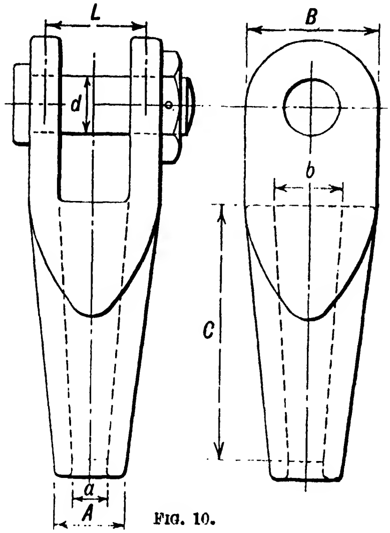


FIG. 10.

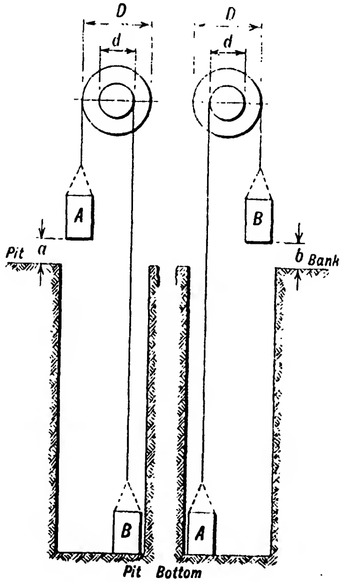


FIG. 11.

level. Then if $\frac{D}{d} = S$ and Z and Y are the alterations needed in length by ropes A and B respectively, we have the following simultaneous equations to determine Z and Y.

$$(1)\ a = Y\frac{D}{r} + Z.$$

$$(2)\ b = Y + Z\frac{D}{d}$$

This result may also be obtained by an alternative method as follows :—

$$Z = \frac{Sb - a}{S^2 - 1}\text{ for rope A.}$$

$$Y = \frac{Sa - b}{S^2 - 1}\text{ for rope B.}$$

Note.—If either cage A or cage B is below the bank level when the other cage is on the pit bottom, then a or b must be introduced in the formula with a negative sign.

If Z or Y comes out positive, rope A or B must be lengthened. If Z or Y comes out negative, then rope A or B must be shortened.

Guides and Rubbing Ropes.—These are made in half-lock and round wire construction. In the shallower pits with light loads round wire construction is usually used, or 7 wires up to about 1⅝ in. diameter, and above this they are usually constructed 8 or 9 outer wires over 7 smaller wires in mild steel rods, of a tensile strength of 28/32 tons per sq. in., or in Swedish charcoal iron.

Half-lock guides and rubbing ropes are usually constructed of a deep half-lock section and round wires over 7 wires. the section being of sufficient depth to withstand long wear. These are made in steel of 30/40, 40/50 or 50/60 tons per sq. in. in tensile.

The arrangement of guide ropes depends upon the layout of the shaft, clearance between the cages, design of cages, and so on. The diameter and breaking loads depending on the depth of the shaft. It is customary to hang weights on the lower end of the guides and rubbing ropes amounting to about 1 ton per 100 yds. of depth. In the deeper pits of 500 yds. or over, rather less is allowed.

The breaking load of a guide rope should not be less than five times the weight on the bottom, and the weight of the suspended length of the rope.

As guide ropes, the advantages of the lock coil construction may be readily understood when it is borne in mind that, in the event of a rod breaking in the ordinary twisted guide rope, it would stand out from the rope, and be caught by the cage gland with disastrous results; in a locked coil rope the broken rod would be securely held in position; this is also the case with Tru-lay ropes.

Aerial Carrying Ropes.—These are usually in locked coil or spiral strand construction, composed of 19, 37, 61 or 91 wires. Locked coil construction may have outer wires of rod and half lock, or full lock. In this country, however, half lock finds greatest favour, and deep sections are easier to produce than full lock sections.

Locked coil ropes possess many advantages for standing ropes or aerial ropeways, as their smooth surface affords easy running for the wheels of the carriage; consequently less haulage power is required and wear and tear is considerably reduced.

The load on an aerial carrying rope depends on the tension and sag. In practice, if the sag is too great, a high bending stress is set up on the saddles at the top of the towers, when the load is passing over, this causes rapid deterioration of the wires due to fatigue. The wires themselves should be of reasonably deep section, so as not to make the rope too flexible.

The stresses on aerial carrying ropes may be calculated from the following formula :—

The tension on an aerial carrying rope is that due to the rope itself plus the tension due to the load or loads added together.

Let T = Tension due to rope.
T_1 = Tension due to load or loads.
W = A single load.
V = Sags, usually 1/35 to 1/40 of the span.
L = Span.
w = Weight of rope per ft.

Then

Tension due to rope $T = \frac{wL^2}{8V}$

Tension due to load $T_1 = \frac{WL}{4V}$ for one load.

$\frac{2WL}{4V}$ for two loads.

$\frac{3WL}{6V}$ for three loads.

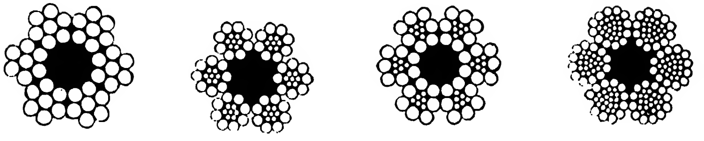
$\frac{\text{No. of loads} \times WL}{8V}$ for four or more loads.

The factor of safety should not be less than 4.

Traction cables are usually ordinary haulage ropes of six round strands or flattened strand construction.

HAULAGE ROPES.

Round Strand. Flattened Strand.



Single Construction. Compound Construction. Single Construction. Compound Construction.

FIG. 12.

Haulage systems vary considerably with the conditions under which they have to work, output required, gradients, and initial cost.

Under rope endless haulage.
Over rope endless haulage.
Main and tail haulage.
Direct haulage.
Self-acting incline haulage.

The choice of which will depend upon the conditions prevailing, gradients, output per shift, speed, and facilities for handling the output at the pit bottom and on the surface, and whether or not the expense is justified.

The usual conditions governing the selection of the various types are briefly as follows :—

Under or Over Rope may be a matter of convention at some collieries, depending on individual preference.

Undertub is used on roads where there is no difficulty in keeping the rope down, and the tubs are attached by means of clips when the gradients are such that there is no danger of slipping.

Overtub Haulage. is common on steep gradients and undulating roads, where it is difficult to hold an under rope down in a swilly. Usually lashing chains or clips are used. Chains are simpler and there is less danger of slip. The tubs, however, cannot be topped up, and the deflection pulleys are somewhat clumsy. They are often used on wet roads to keep the rope out of the water.

Main and Tail Haulage is used on gradients not steep enough for sets to gravitate and pull the rope. Also on undulating roads and level roads where it is not convenient to maintain a double road width.

Direct Acting Haulage is used where roads are narrow and the dip sufficient to gravitate the tubs and pull the rope out.

Self-acting Jigs are used where the gradient is in favour of the load, and sufficient for the full tubs to pull the empties up the incline.

Selection of Rope.

Notes on Rope Construction.

Haulage ropes are generally constructed 6 strands of 7 wires over a fibre core up to 4 in. circumference, and above this size a compound construction is used, in order that the wires are in reasonable proportion to the usual size of haulage drums and pulleys. The ratio of wire diameter to drum or pulley diameter should never be less than 1 to 450, but the aim should be 1 to 650, provided that the wire is of such a size to provide reasonable wear. The selection of the rope has an important bearing on its life. See fig. 12.

In many cases flattened strand ropes provide an improved life, due to the additional wearing surface, but greater care must be taken with splicing. The majority of haulage ropes are supplied in Lang's lay, but there are certain cases where ordinary lay has to be used. Haulage ropes are supplied in ranges of tensile strengths between 80 and 120 tons per sq. in., but 100/110 tons per sq. in. gives best all round service on heavy haulages.

Drum and pulley sizes are often inadequate for the size of rope in use, particularly diverting pulleys, at the top of a brow or round a curve. Extreme pressure and high bending stresses are caused to the detriment of the life of the rope.

The pressure on a single pulley at the top of a steep gradient can be obtained from the following calculation. See fig. 13.

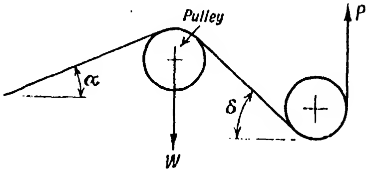


FIG. 13.

Pressure on pulley W = (P × sin α) + (P × sin δ)
Where P = Rope pull.
W = Pressure.
α and δ = Angles of inclination.

On some installations a jockey wheel is in use for tensioning the rope as in fig. 14.

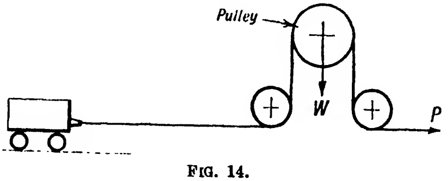


FIG. 14.

In this case W = 2P

O. D. Meals states that the unit radial pressure of a wire rope in a groove of a sheave or drum is given by :—

$U = \frac{T}{Rd}$ Where U = Unit radial pressure per sq. in. of projected area.
T = Total load stress in lbs.
R = Radius of sheave or drum at bottom of groove.
d = Diameter of rope in ins.

U is independent of the arc of contact and to consider this is erroneous.

SAFE UNIT RADIAL PRESSURE IN LBS. PER SQ. IN.

Material.	Ordinary Lay Ropes.			Lang's Lay Ropes.		
	6/7/1	6/19/1.	6/37/1.	6/7/1.	6/19/1.	6/37/1.
Cast iron . .	300	500	600	350	550	660
Cast steel . .	550	900	1,075	600	1,000	1,180
Manganese steel .	1,500	2,500	3,000	1,650	2,750	3,300

Where an ungrooved pulley is used, the pressure is concentrated on the crowns of the strands, and is greatly in excess of that when the rope is supported in a groove.

It is true that high pressures are likely to occur at such places as rounding a curve on diverting pulleys, too few and too small in diameter. Also at the top of an incline. The load should be distributed over a series of pulleys, rather than over one or two, as is the case in many installations, as the pressure varies in direct proportion to the diameter of the pulley.

Discarding of Ropes.—It is very difficult to assess the amount of remaining strength of a worn rope, unless one can examine a piece of rope. Measuring the outside diameter of a rope in service is not a true indication of wear, it might be that the fibre core has pulled down slightly under load, therefore the measurement does not necessarily indicate the amount of wear in the wires. If a rope is respliced at intervals a true examination can take place.

Preferably the wires in a new rope should be tested and the tests recorded, and comparisons made from time to time of samples taken from the rope during its lifetime. These should include tensile tests, number of bends, and number of torsions in 100 diameters.

Wear and corrosion are two frequent reasons for failure, and it is as well to examine the wires for pitting and cross-cutting of adjacent wires, and when tests reveal that the factor of safety is becoming low the rope should be discarded.

When a rope is respliced, the worn wires should be tested and the amount of deterioration in a specified period may be determined and recorded. Fig. 15 shows a chart from which the remaining area of worn wires may be readily obtained.

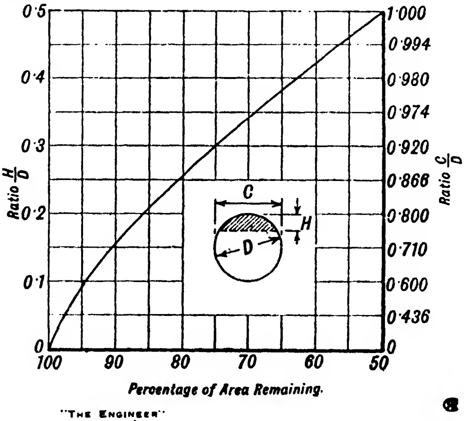


FIG. 15.

Deterioration.—The booklet entitled 'The Examination of Colliery Winding Ropes in Service,' published by S.M.R.B. in 1938, is deserving of careful study. On page 32 the limits of deterioration allowable are carefully set out, the following clauses being of utmost importance: 'As a general rule, no rope should remain in use after it has lost 20 per cent. of its strength by fatigue or corrosion fatigue, that is to say, a rope suffering deterioration by wear or corrosion is not likely to become dangerous until it has lost 20 per cent. of its strength provided fatigue is absent.' . . . 'No stranded rope should remain in use after the outer wires have lost 40 per cent. of their diameter by wear.'

Friction of Tubs.—Opinions differ widely as to the coefficient of friction for haulages, and 56 lbs. per ton of load, or 1/40, appears to be satisfactory for good average bearings efficiently lubricated.

The following table gives the pull per ton on various inclines including a coefficient of friction of 1/40.

TABLE FOR ASCERTAINING PULL ON HAULAGE ROPES.

Incline to Horizontal.	Pull in Lbs. per Ton including Friction 1/40.	Incline to Horizontal.	Pull in Lbs. per Ton including Friction 1/40.
1 in 1 = 45°	1624	1 in 21	162
1 in $1\frac{1}{2}$	1289	1 in 22	158
1 in 2	1042	1 in 23	153
1 in $2\frac{1}{2}$	884	1 in 24	149
1 in 3	761	1 in 25	146
1 in $3\frac{1}{2}$	670	1 in 26	142
1 in 4	598	1 in 27	139
1 in $4\frac{1}{2}$	541	1 in 28	136
1 in 5	494	1 in 29	133
1 in $5\frac{1}{2}$	456	1 in 30	131
1 in 6	423	1 in 31	128
1 in $6\frac{1}{2}$	396	1 in 32	125
1 in 7	372	1 in 33	123
1 in $7\frac{1}{2}$	352	1 in 34	122
1 in 8	334	1 in 35	120
1 in $8\frac{1}{2}$	318	1 in 36	118
1 in 9	303	1 in 37	116
1 in $9\frac{1}{2}$	290	1 in 38	114
1 in 10	278	1 in 39	113
1 in 11	258	1 in 40	112
1 in 12	244	1 in 45	105
1 in 13	227	1 in 50	100
1 in 14	215	1 in 55	97
1 in 15	205	1 in 60	93
1 in 16	196	1 in 70	88
1 in 17	187	1 in 80	84
1 in 18	180	1 in 90	80
1 in 19	173	1 in 100	78
1 in 20	168	1 in 110	76

Example.—Multiply the figure in the table by the gross load and weight of rope.

Load and rope	10 tons.
Incline	1 in $3\frac{1}{2}$.
Multiple	670
Load on rope	670 × 10 = 6,700 lbs.

Allow for factor of safety of not less than 4.

Min. B/L. = 26,800 or 12 tons (approximately).

The formulæ below give the pull on haulage ropes and covers load friction, rope friction, and friction of wheels and rollers:—

Where full tubs are ascending the incline.

W_f = Weight of full tubs and coal.

W_e = Weight of empty tubs only.

W_r = Weight of rope on incline.

$\frac{i}{h}$ = Incline to horizontal.

$$\text{Pull on ascending side} = \left(\frac{W_f + W_r}{h}\right) + 0\cdot0433\,W_f + 0\cdot05\,W_r.$$

$$\text{Pull on descending side} = 0\cdot0433\,W_e + 0\cdot05\,W_r - \left(\frac{W_e + W_r}{h}\right)$$

$$\text{H.P.} = \frac{\text{Nett pull at haulage wheel} \times 2{,}240 \times \text{speed in ft. per min.}}{33{,}000 \times 0\cdot8.}$$

Allowing for 80 per cent. efficiency.

Splicing.—The Endless Spice.—Haulage ropes may be made endless by means of a long splice. The length recommended depends on the size of the rope and the duty imposed on it.

A good guide for total length is 10 ft. for each ⅛ in. of diameter, for Lang's lay ropes. It is best to make the length divisible by 6, so that the length of the tuck can readily be cut off to an even length. This is to say, if a splice works out at 47 ft., make it 48 ft.

Wire Rope Splicing.

Long Splice.

This splice is used for haulage and where endless ropes are required.

Mark off both ends the length required for splicing, which depends on the size of the rope, say 48 ft., that is 24 ft. each end. Having marked each end, unlay three alternate strands from each end to within 6 ins. of mark, as shown in fig. 16, L.

Cut off the three remaining strands with the core and bend back at mark, as shown in fig. 16, R.

Lock both ends together, close up, so that the long strands pass alternately right and left, as shown in fig. 17.

Grip the rope on one side of the lock with tongs, unlay one short strand and lay up in its place the corresponding long strand until within 4 ft. of the end. To facilitate operations, the short strand may now be cut 4 ft. from the cross, the two ends measuring 8 ft. Lay up the other two long strands, the crosses being 8 ft. apart; next repeat the operations with other side, cutting the ends at 4 ft. and the crosses being 8 ft. apart, as shown in fig. 18.

The next operation is to run the ends in and substitute the core. It is advisable to serve the ends, this gives the outer strands a firm grip on them and prevents loose wires protruding through interstices of the strands. Grip the rope with tongs where indicated by arrow, drive the spike through centre of rope three strands in advance of the strand to be run in, and with the aid of short spike pick the core out, as shown in fig. 19.

Run spike back towards the tongs, insert needle over the end in the place of spike, hook spoon under the end, and in the same place as the needle, as shown in fig. 20.

Take two or three turns out of the strand and twist it round the rope as indicated by arrow, at the same time bring the handle of the needle and spoon together, as indicated. This forces the strand into centre of rope, and by twisting the needle round the lay the strand will enter the rope one side and force the core out on the other. Cut the core close up to the end of strand, and turn and run the opposite end in exactly the same way.

Run in the other five pairs of ends in exactly the same way as the first pair.

Care should be taken not to run the core too far back, otherwise flat places will occur.

This completes the splice, and if correctly executed there should be very little difference between the splice and any other portion of the rope.

With a Lang's lay rope the ends of the strands are run into the right of each other, as shown in fig. 21.

With the ordinary lay rope the ends of the strands are run into the left of each other.

Lubrication.—Wire ropes are thoroughly lubricated during manufacture to protect the fibre core against decay, and the internal wires against corrosion.

During service it is not easy (not possible in the case of locked coil ropes), and should not be necessary to replenish the internal lubricant, but the rope should be lubricated externally at such intervals and in such a way as to keep the wires covered. It is important that nothing should come into contact with the wires which is capable of producing chemical action. The rope should be cleaned as necessary with a light mineral oil and lubricated with a mineral oil compounded with petroleum jelly or graphite according to climatic conditions. If the lubricant is heated before application its chances of penetration are improved.

Factor of Safety.—The factor of safety to be used depends largely on the condition of service, and the following are the minimum for new ropes.

Dock cranes	5
Workshop cranes	6
Passenger lifts and hoists	12
Goods lifts	12

Colliery Winding Ropes.

Locked coil over 700 yds.	Not less than 7
,, ,, under 700 yds.	,, ,, ,, 8
Round strand	10
Flattened strand	9
Mineral haulage ropes	4
Man-riding haulage ropes	7 to 8

For colliery winding ropes the capacity factor is sometimes used to describe the factor of safety of a rope at the capel, that is the breaking load of the rope divided by the weight of the loaded cage and suspension gear. By this method the deeper the pit and the lower the static factor of safety of the rope. In a very long rope, the elastic elongation is greater than that of a

Wire Rope Splicing.

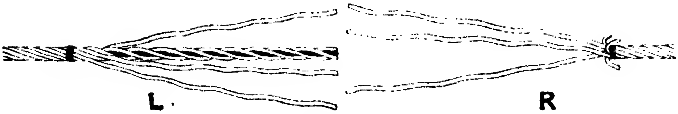


Fig. 16.

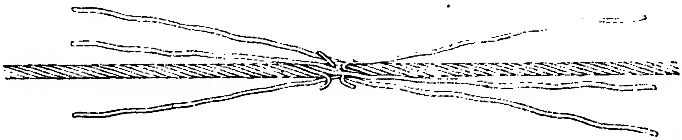


Fig. 17.

Fig. 18.

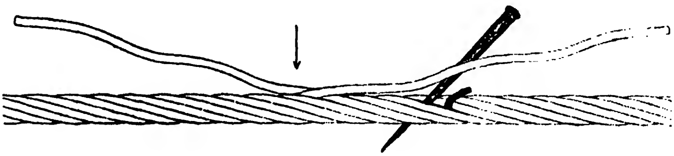


Fig. 19.

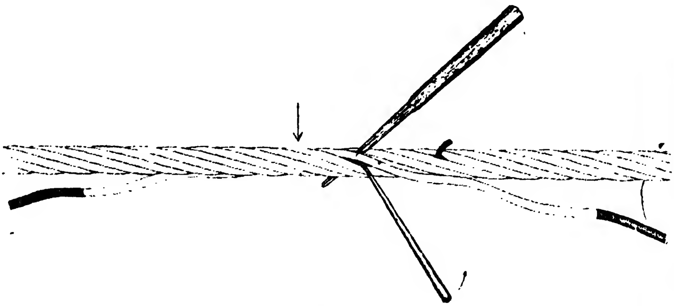


Fig. 20.

Fig. 21.

short rope, and it is more capable of absorbing shocks. Therefore the capacity factor takes care of the varying depths of shaft, and provides an adequate static factor of safety.

The following are minimum capacity factors recommended by Dr. Hogan.

Round strand	Not less than $13\frac{1}{2}$
Flattened strand	„ „ „ 12
Locked coil	„ „ „ 10

These figures are, however, subject to increase when the conditions are unfavourable.

Damping Dynamic Stresses in Mine Winding Ropes.

Investigation has shown that dynamic shocks have a marked influence on the life of steel winding ropes, and has led to the design of apparatus for the continuous control of the stresses in the ropes.

The elements of the shock-damping buffer are springs and pistons moving in hydraulic cylinders; the buffer (Koepe system) is preferably interposed at the lower end of the rope, as shown in fig. 22.

The buffer is shown in fig. 23. The main bar (*b*) passes through the roof of the cage (*a*) and is connected to the piston-rod (*c*) by a bolt (*d*). The buffer itself consists of the piston (*f*), the

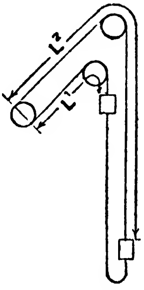


FIG. 22.

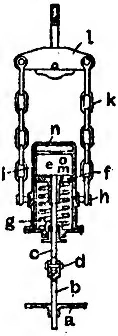


FIG. 23.

spring (*g*), and the cylinder (*e*), and is suspended by the two main trunnions (*h*), the chains (*k*), and the headpiece (*l*). The dynamic effect of the brake depends on the throttle valves (*m*) (of which there are four), through which the fluid (glycerine) contained in the cylinder is forced as the cylinder is displaced. A feather (*i*) prevents the pistons from turning. An expansion space (*n*) is provided above the main space (*e*) of the cylinder, and connected to it by a hole (*o*). This space (*n*), which is filled only partly with glycerine, serves to accommodate the expansion of the glycerine through heat and its varying displacement by the piston-rod.

From fig. 22 it will be seen that when the lower cage is stopped at the bottom of the pit the heaviest shocks come when the portion L1 of the winding rope is above the upper cage. Some of the shocks are taken up by the natural elasticity of the rope, but if L1 is only short it cannot reduce them very much. In the first pit in which the brake was applied the dimensions of the piston and of the spring were so calculated that the effect of the damping apparatus was equivalent to that of a wire rope 400 ft. long. As the length L1 was about 200 ft., the total elasticity was tripled by the arrangement. This holds for a new rope; old ropes have not the same elasticity as new ones, and with them the relative effect of the damping apparatus is therefore much greater. The first buffers were installed in 1924 at the Hannover pit (Germany), and have since been in continuous service. The springs were designed for a normal load of 21·5 tons which compresses them by about 6 ins. They may be compressed for 3 ins. more with a load up to 32 tons. The total height of the spring is 24 ins., and its section is 1·4 in. by 2·6 ins. There are about 12 coils in each spring.

The practical value of the buffers can be seen from the records of a winding-rope. At the Hannover pit 140 wires had broken in a rope without a buffer in two years, and the rope had to be removed after this time, but after the buffers has been fitted, only 20 wires were broken in two years of use; moreover, the breaks in the wires of the rope without buffers were concentrated at certain points, but were regularly distributed in the rope with buffers. The use of this rope has been permitted up to four years, so that its life is at least doubled. Another advantage is that the renewal of the rope, which causes delays, is required less frequently.

(*Times Engineering Supplement*, *Oct.* 1928.)

Various other methods are also employed for the above purpose.

Flat Ropes.

Flat ropes are composed of a number of individual ropes, alternative right and left lay, arranged side by side and stitched together with soft steel stitching wires. These ropes can be used with advantage when space for machinery is very limited, or where it is necessary in any winding operation that the rope should operate on a centre line without travelling across the drum.

Flat ropes make excellent balance ropes, and are extensively used for parallel drum installations where the cage centres are relatively close. They are also used on some installations for lifting heavy loads where an absolutely non-spinning rope is necessary.

The usual construction is 6 or 8 strands of 4 reddies each stitched together with 3-ply strand stitching or a series of single wires.

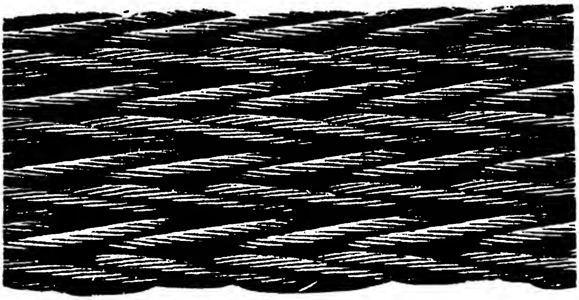


Fig. 24.—Eight strands of 4 Reddies of 7 wires each.

Coiling on Drums.—Many ropes give unsatisfactory service because they are not spooled properly on the drum. If a rope is wound from the wrong end of the drum for the direction of lay, lengthening of the lay can be caused or deformation due to local tightening up of the strands. In order to avoid these troubles the following rules should be observed for the four different possibilities which exist:—

(1) Overwinding on the drum from left to right, use a right-hand lay.
(2) Overwinding on the drum from right to left, use a left-hand lay.
(3) Underwinding on the drum from left to right, use a left-hand lay.
(4) Underwinding on the drum from right to left, use a right-hand lay.

Illustration in fig. 25 shows the correct winding for the four conditions.

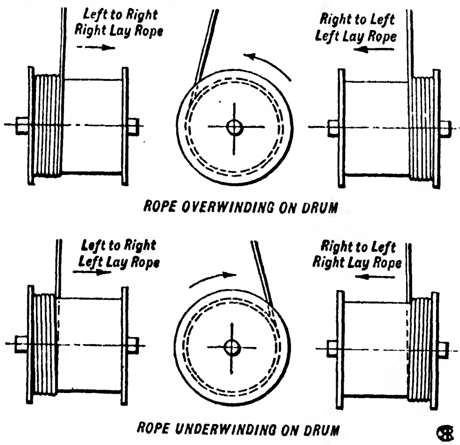


Fig. 25.

Slings.—The safe working load on a double wire rope sling may be calculated from :—

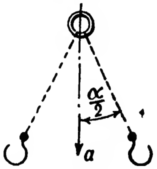


FIG. 26.

α = included angle.
a = working load of sling.
x = factor of safety.
c = breaking load of rope.

$$a = \cos\frac{\alpha}{2} \times \frac{C}{0 \cdot 5x}$$

Rings.—Strength of iron rings :—

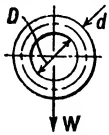


FIG. 27.

W = proof load.
D = internal diameter.
d = diameter of bar from which ring is made.

$$W = \frac{14 \cdot 8\, d^2}{D + 0 \cdot 3d}.$$

$\frac{D}{d}$ should not be less than 2 or greater than 7. (B.S.S. 781—1938.)

Small Bow Shackles.

d = diameter of material.
W = safe working load.
D = diameter of pin.
s = distance from pin to inside of bow.
w = width between jaws.

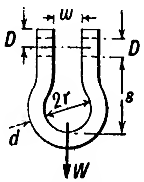


FIG. 28.

$$D = 0 \cdot 67\sqrt{W}$$
$$d = 0 \cdot 67\sqrt{W}$$
$$w = 0 \cdot 75\sqrt{W}$$
$$s = 2 \cdot 4\sqrt{W}$$
$$2r = 1 \cdot 6\sqrt{W}$$
$$\text{Pin, } W = \frac{6 \cdot 3\, D^2}{(w + d}$$

(B.S.S. 825—1939.)

Links.

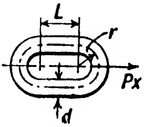


FIG. 29.

P = working load.
x = factor of safety.
d = diameter of material.

$$d = \sqrt[3]{\frac{P x r (L + 2 r)}{21 (L + \pi r)}}$$

Proof load on short link chains
$= 12 \times d^2$ tons.

See also Descriptive Section XXI, Part III.
Whitecross Co., Ltd.

SECTION XXI

PART IV

CHAIN DRIVING.

Precision chains present an efficient method of transmitting power at short and medium centre distances.

While the drive is positive, and so preserves a definite speed ratio and relationship between the driven and the driving shafts, a certain amount of resilience or flexibility is also present.

Differences in atmospheric conditions do not affect a chain drive, which is available for transmitting the full power of the electric motor or other prime mover immediately on starting up.

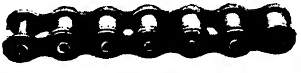
Static electrical effects are entirely absent.

Efficiency tests conducted at the National Physical Laboratory have demonstrated that the efficiency of a roller chain drive is in the neighbourhood of 98·5 per cent.

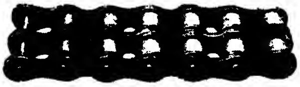
TYPES OF CHAIN.

Two types of chain are available for power transmission purposes: roller type and inverted tooth type.

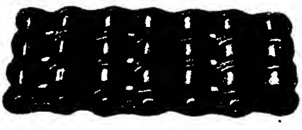
ROLLER CHAINS.



Simple.



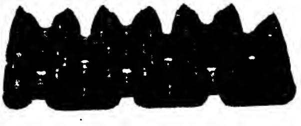
Duplex.



Triplex.

INVERTED TOOTH CHAINS.

Segmental Bush—Outside Guided.



Segmental Bush—Centre Guided.

The roller type of chain is generally used in preference to the inverted tooth type, since it is simpler, cheaper and requires less width for the transmission of a given power.

The inverted tooth type is furnished in two classifications, *i.e.* the round pin chain and the rocker-joint chain. It is claimed that in the latter there is a reduction in frictional losses, as the rocker-joint works on the principle of an oscillating pin on a plain surface. The roller type of chain is manufactured in the round pin type shown in our illustration, and also in the form of a semi-segmental joint construction, which gives more uniform wheel tooth wear and semi-automatic lubrication of the pin and bush elements.

The standardisation of roller chains and chain wheels has been investigated by the British Standards Institution, who have issued British Standard Specification No. 228—1934, to which reference may be made for further particulars.

Standardisation of inverted tooth chains has not been undertaken.

The following notes and formulæ apply to chains of the roller type only

Proportions of Roller Chains.

Although the dimensions of the standardised roller chains are not in strict relationship to the pitch throughout, it is generally recognised that a roller diameter 0·625 × Pitch represents the best compromise between the provision of adequate shearing and bearing areas in the circular parts and the maintenance of sufficient height and strength in the wheel teeth.

The width between the inner plates is a basic dimension and is, in general, 0·6 × Pitch or 0·4 × Pitch; chains with the first-mentioned proportion being known as 'wide,' and those with the second as the 'narrow' series. The former is preferred for general use.

Relationship of Chain Pitch to Shaft Speed.

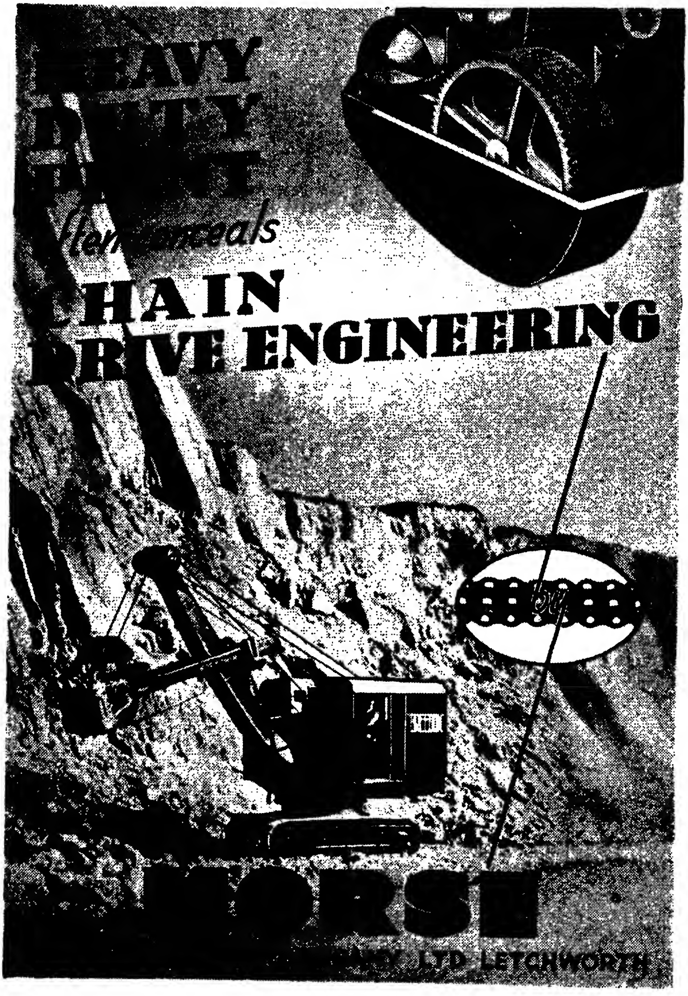
Experience has established maximum pinion speeds for each pitch of chain in normal drives; these speeds, which are those of the smallest pinion or wheel in the drive and relate to numbers of teeth from 19 to 30 inclusive, are given in the following table :—

Pitch of Chain. (Inches.)	Normal Maximum Pinion Speed. (Rev. per min.)
0·375	3,600
0·500	3,000
0·750	1,600
1·00	1,200
1·25	800
1·50	750
1·75	600
2·00	550
2·50	400
3·00	300

Numbers of Teeth in Wheels.

While there is no hard and fast dividing line for the mimimum number of teeth to be used in a pinion (cutters are available for cutting down to 9 teeth) it is not good practice to use less than 19 teeth for drives running at normal speeds, and for some applications, *e.g.* for the driving of electric generators, the pinion should have not less than 23 teeth.

It is not advisible to exceed 150 teeth in the large wheel—a more usual limitation is 114 teeth—which, in conjunction with a 19-tooth pinion, allows a maximum ratio of 6 to 1. For larger ratios than this, it is advisible to make the reduction, or increase, in two stages; the ratio on each stage then being the square root of the total ratio.



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The disadvantages of using large wheels are that an earlier limit is set to the life of the chain due to the fact that a given percentage of chain elongation, through wear, causes the chain to assume a correspondingly greater pitch circle diameter on the wheel, and that a large ratio drive calls for the centre distance to be increased so that the desirable minimum arc of contact of 120° on the pinion will be maintained.

In low-ratio drives it is good practice to have the sum of the teeth in the two pinions not less than 50, *e.g.* a 1 to 1 ratio drive should have 25 teeth in each pinion.

The shaft speeds in a chain drive are inversely as the numbers of teeth in the wheels.

Nature of Load—Bearing Pressures.

For all drives, except those running at very slow chain speeds, the bearing pressure in the chain joints is the important consideration: where this condition is observed the necessary factor of safety on the chain breaking load automatically follows.

The nature of the load and the service expected from the chain have a determining influence on the bearing pressure to be allowed.

For example, in drives working 50 hours per week, where the loads are steady and the conditions generally are favourable, bearing pressures up to 2,250 lb. per sq. in. of bearing pin projected area may be allowed.

On the other hand, for drives working continuously and where the conditions are more arduous, due to the presence of impulsive loads, the bearing pressure should be restricted to 1,250 lb. per sq. in.

For medium conditions, where loads may be unsteady but not impulsive, a bearing pressure intermediate between these two, 1,750 lb. per sq. in., may be allowed.

In general, where cyclic speed variation in excess of 3 per cent. is likely to be encountered, special consideration should be given to the conditions; it may be necessary to instal a shock-absorbing coupling or to provide additional flywheel effect.

Horse-power Transmitting Capacity of Roller Chains.

The horse-power transmitting capacity of a roller chain (wide series) can be conveniently expressed by the following formula:

$$\text{H.P.} = \frac{C.P^2 tn}{1000}$$

where C is a constant (see table below).
P is chain pitch in inches.
t is number of teeth in pinion.
n is revolutions per minute of pinion.

Values of C.

Conditions.	Simple.	Duplex.	Triplex.
Arduous—continuous running	1·0	2·0	3·0
Medium—not impulsive	1·4	2·8	4·2
Steady—favourable	1·8	3·6	5·4

Where one triplex roller chain is insufficient to transmit the required power, two duplex roller chains running side by side on integral wheels may be used. If the calculations show that these are insufficient, it is permissible to use up to four triplex chains running side by side on integral wheels. An essential condition for multiple drives, however, is that the chains fitted on them must be matched and the chain manufacturer's instructions regarding erection must be strictly observed.

Limitations on Pinion Size Imposed by Shafting.

The number of teeth in a pinion is frequently governed by the size of the shaft on which it is to be mounted : the following rough rule may be used to check the selection:

$$\text{Minimum teeth in pinion} = \frac{4 \times \text{shaft dia.}}{\text{Pitch}} + 7.$$

Centre Distance Between Shafts.

For normal conditions, the following expression may be used to determine a suitable centre distance between the driving and the driven shafts :—

$$\text{Centre distance (in.)} = 24 \times \text{Pitch} + 18.$$

Drives may, however, be run at centre distances such that the teeth of the wheels just clear one another, though it should be appreciated that due to the more frequent repetition of the chain in such drives its life is likely to be shorter.

Wherever possible, the centre distance should not be less than that required to preserve an arc of contact of 120° on the pinion.

Overall Dimensions of Drive.

Having determined the pitch of roller chain to be used and the numbers of teeth in the wheels, the approximate overall dimensions of the drive can be obtained from the following expressions, though for exact sizes reference should be made to the chain makers' catalogues or data sheets :—

Lateral space required for chain	Simple	1·75 × Pitch.
	Duplex	3·00 × Pitch.
	Triplex	4·25 × Pitch.

Clearance at sides of chain (per side) :—

0·375 in. + 0·125 in. per foot of centre distance.

For multiple strand drives, the total width will be the sum of the chain widths plus the side clearances plus 0·25 in. clearance between adjacent chains.

The radial dimension over the chain, measured from the shaft centre, exceeds the pitch radius of the wheel by approximately 1·3 × Pitch : this allows for the rise of the chain on the wheel due to wear. For clearance purposes, an amount equal to twice the pitch of the chain, with a maximum of 3 in., should be added.

Unit Pitch Diameter Table.

The following table gives pitch diameters of chain wheels in inches for roller or inverted tooth chains of 1 in. pitch.

Pitch diameters of wheels for other pitches of chain can be obtained from this table by multiplying the figures given by the pitch of the chain under consideration, *e.g.* for 1·25 in. pitch, multiply the figures in the table by 1·25 :—

Number of Teeth.	Pitch Diameter.	Number of Teeth.	Pitch Diameter.	Number of Teeth.	Pitch Diameter.
9	2·9238	57	18·1529	105	33·4275
10	3·2361	58	18·4710	106	33·7458
11	3·5494	59	18·7892	107	34·0640
12	3·8637	60	19·1073	108	34·3823
13	4·1786	61	19·4255	109	34·7006
14	4·4940	62	19·7437	110	35·0188
15	4·8097	63	20·0619	111	35·3371
16	5·1258	64	20·3800	112	35·6554
17	5·4422	65	20·6982	113	35·9737
18	5·7588	66	21·0164	114	36·2919
19	6·0755	67	21·3346	115	36·6102
20	6·3925	68	21·6528	116	36·9285
21	6·7095	69	21·9710	117	37·2467
22	7·0266	70	22·2892	118	37·5650
23	7·3439	71	22·6074	119	37·8833
24	7·6613	72	22·9256	120	38·2016
25	7·9787	73	23·2438	121	38·5198
26	8·2962	74	23·5620	122	38·8381
27	8·6138	75	23·8802	123	39·1564
28	8·9314	76	24·1985	124	39·4746
29	9·2491	77	24·5167	125	39·7929
30	9·5668	78	24·8349	126	40·1112
31	9·8845	79	25·1531	127	40·4295
32	10·2023	80	25·4713	128	40·7478
33	10·5201	81	25·7896	129	41·0660
34	10·8380	82	26·1078	130	41·3843
35	11·1558	83	26·4260	131	41·7026
36	11·4737	84	26·7443	132	42·0209
37	11·7916	85	27·0625	133	42·3391
38	12·1096	86	27·3807	134	42·6574
39	12·4275	87	27·6990	135	42·9757
40	12·7455	88	28·0172	136	43·2940
41	13·0635	89	28·3355	137	43·6123
42	13·3815	90	28·6537	138	43·9306
43	13·6995	91	28·9719	139	44·2488
44	14·0176	92	29·2902	140	44·5671
45	14·3356	93	29·6084	141	44·8854
46	14·6537	94	29·9267	142	45·2037
47	14·9717	95	30·2449	143	45·5220
48	15·2898	96	30·5632	144	45·8403
49	15·6079	97	30·8815	145	46·1585
50	15·9260	98	31·1997	146	46·4768
51	16·2441	99	31·5180	147	46·7951
52	16·5622	100	31·8362	148	47·1134
53	16·8803	101	32·1545	149	47·4317
54	17·1984	102	32·4727	150	47·7500
55	17·5166	103	32·7910	—	—
56	17·8347	104	33·1093	—	—

Wheel Materials.

The normal material for pinions from 16 to 29 teeth inclusive, is a 0·6 per cent. carbon steel, untreated.

For pinions having less than 16 teeth heat treatment is advisable to obtain maximum resistance to wear.

Wheels having 30 or more teeth can be made from a close grain cast iron for normal applications, but where shock loads are likely to be encountered and for certain specific applications, *e.g.* drives for marine work, steel forgings or castings are preferable.

Wheel Machining.

All wheels for use with power transmission chains should have machine-cut teeth. The machining may be done by rotary cutters, rack planing cutters or by hobs.

British Standard Dimensions for rotary tooth form cutters and for basic rack tooth shapes for roller chain wheels are contained in the British Standard Specification No. 228—1934, previously referred to.

When machining roller chain wheels, the bottom diameter is the important dimension: top diameter is relatively unimportant, and assuming the blank is of adequate size, is controlled by the cutter.

Casing and Lubrication.

The bearing surfaces in precision roller chains are of casehardened steel and efficient lubrication must therefore be provided for them. This is usually done by enclosing the drive in a sheet metal case and directing oil on to the inner edges of the plates by means of feed or spray pipes, the oil being supplied either from a drip feed lubricator, or by a simple type of pump.

Certain dispositions of drive can be lubricated on the oil bath principle, by maintaining oil in the case at such a level that the chain dips into it while running, but the pump method is in general the most satisfactory and is practically essential where powers in excess of 50 H.P. are concerned.

Chain Adjustment.

The cumulative effect of bedding-down and subsequent normal wear of the bearing surfaces in a chain makes it necessary to provide for chain adjustment. The simplest method is to arrange for the shaft carrying the driving or the driven wheel to be movable where this is practicable; otherwise a toothed jockey pinion, having at least three teeth in engagement with the non-driving strand of the chain, should be fitted.

Care should be taken when deciding on the initial centre distance of a drive to ensure that it will suit a chain having an even number of links when it is approximately at the commencement of the adjustment range (a small amount of negative adjustment is helpful during installation), as the use of cranked, or as they are sometimes termed 'half,' links is most undesirable.

The simplest method of checking chain length, particularly where more than two wheels are involved, is to make a scale layout of the drive and to step around the chain contact arcs of the pitch circles with dividers set accurately to the pitch; the free lengths of chain being measured directly between the terminal pitch points on the wheels.

Chain Drive Formulæ.

P = Pitch of chain in inches.
W = Weight of chain in lb. per foot.
t = No. of teeth in wheel.

Chain Speed (ft. per minute): $\frac{Pt}{12} \times$ r.p.m. of wheel.

Tension in Chain due to Drive (lb.): $33{,}000 \times \frac{\text{H.P. transmitted}}{\text{Chain speed}}$.

Tension in Chain due to Centrifugal Force (lb.): $8 \cdot 6\ W \times \left(\frac{\text{Chain speed}}{1{,}000}\right)^2$

Total Tension in Chain (lb.).—The sum of the tensions due to drive and centrifugal force.

Bearing Pressure (lb. per sq. in.).—Total tension divided by bearing area in sq. in.

Factor of Safety.—Breaking load of chain divided by total tension in chain.

See also Descriptive Section XXI, Part IV.
Morse Chain Co., Ltd.
Renold & Coventry Chain Co., Ltd.

SECTION XXI

PART V

SPUR GEARING—BEVEL GEARING—HELICAL GEARING—TURBINE GEARS—WORM GEARING.

(Contributed by H. T. Davey, M.I.Mech.E.)

Definitions.

Straight Spur Gear.—One having the faces of the teeth perpendicular to the plane containing the pitch circle.

Bevel Gear.—A gear in which the teeth are formed on a cone instead of on a cylinder, as in a straight spur gear, the pitch cylinder in the latter being replaced by a pitch cone in the former. See p. 988.

Helical Gear.—A type of gear in which the teeth are portions of helices, the number of helices being equal to the number of teeth in the gear. Such gears may be single, double or triple helical.

Rack.—A gear in which the pitch circle is of infinite radius, and is therefore a straight line.

Crown Wheel and Pinion.—A crown wheel is a bevel wheel in which the pitch cone angle is 90° (see fig. 11), a crown wheel being therefore a disc in which the tooth faces are perpendicular to, and radiate from, the axis of rotation. A crown wheel is in effect a circular rack. For a pinion to gear with a crown wheel the axis of its shaft must be inclined at an angle greater than 90° to the axis of the crown wheel.

Hyperboloid Gears.—The teeth of these are cut on a hyperboloid, the solid of revolution of a hyperbola. They are used for the transmission of power between skew shafts, but owing to difficulties of manufacture are not very common.

Hypoid Gears.—Similar to hyperboloid gears but manufactured along the lines of a helical bevel gear (see p. 989). Used for purposes similar to the hyperboloid gear.

Lantern Pinion.—A pinion, the teeth of which are formed by round bars or rods of equal lengths, the ends of which are fixed in and near the peripheries of two discs mounted centrally on a spindle with their planes parallel to one another. Within limits such a pinion will operate wheels having any number of teeth.

Straight Spur Gears—Nomenclature.

Pitch cylinder is a virtual cylinder. the cylindrical surface of which would transmit by friction when in contact with a similar cylindrical surface the required relative motion.

Pitch circle is the end elevation of the pitch cylinder.

Pitch surface lies virtually in the tooth space, and is the surface of the pitch cylinder bounded on each side by the lines of intersection of the faces and flanks of adjacent teeth.

Pitch line is the line of intersection of the face, and flank of a tooth. In the case of a straight spur gear it is the line perpendicular to the plane of the pitch circle at the point where the pitch circle is intersected by the tooth profile. Pitch line should not be confused with pitch circle.

Base Circle.—The generating circle or circle from which the involute curve for the tooth profile is developed in the case of involute teeth.

Diameter of a gear is the diameter of the pitch circle, and is often referred to as the pitch diameter.

Circular pitch is the length of the arc of the pitch circle measured from a point on one tooth to the corresponding point on the next tooth. It is equivalent to the circumference of the pitch circle divided by the number of teeth in the gear.

Diametral pitch is the number of teeth for each inch of pitch diameter, and is equivalent to the number of teeth in the gear divided by the diameter. It is a ratio and not a dimension. If a gear has a pitch diameter of 4 ins. and has 40 teeth, then it has a diametral pitch of 10.

Module.—This is the reciprocal of the diametral pitch and hence represents the pitch diameter per tooth. Measurement of pitch in this way is used mainly on the continent of Europe, in which case the diameter of the pitch circle is expressed in millimetres and the module stated accordingly.

Chordal pitch is the length of a chord of the pitch circle drawn from two exactly similarly placed points on two contiguous teeth. Convenient points are the centres of two contiguous teeth on the pitch circle. The chordal pitch is thus slightly less than the circular pitch.

Addendum is the distance measured radially from the pitch circle (or perpendicularly from the pitch line in the case of a rack) to the crest of a tooth. It is the difference between the radii of the gear blank and the pitch circle.

Dedendum is the distance measured radially from the pitch circle (or perpendicularly from the pitch line in the case of a rack) to the root of a gear tooth, or to the more definite surface, the bottom of the space between two contiguous teeth.

Clearance is the distance between the bottom of a tooth space and the crest of the mating tooth when in full mesh. It is the difference between the dedendum and addendum of the mating teeth. This clearance is quite independent of any side clearance which may be allowed between a tooth in full engagement and the corresponding tooth space.

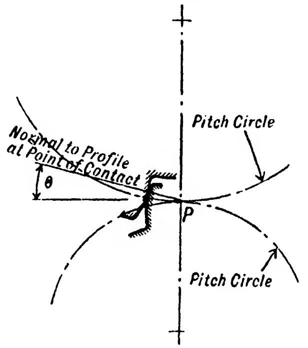


FIG. 1.

Whole depth is the total distance measured radially between circles containing the periphery of the teeth and the bottom of the tooth spaces.

Working depth is the distance which the teeth of one gear extend into the spaces of its mating gear. It is equal to the whole depth minus the clearance, and is also equal to the sum of the addenda of two mating gears.

Line of action.—In properly formed gear teeth a line drawn normal to the profile at the point of contact of two engaging teeth passes through the pitch point, P, fig. 1. For *involute teeth* all points of contact of engaging teeth should fall on this line in order to secure uniform transmission of motion. For other forms of tooth profile the path of contact is not a fixed straight line, but the normal to the profiles at point of contact must always pass through the pitch point.

Pressure angle or angle of obliquity of action is the angle of inclination of the line of action to the tangent at P. See angle θ, fig. 1.

Interference occurs when points of contact of mating teeth do not fall on the line of action. Motion cannot be uniform under these conditions.

Active profile is that part of the tooth which actually comes into contact with the profile of its mating tooth.

Arc of approach is the arc of the pitch circle through which two engaging wheels rotate from the time that contact is first made between two mating teeth, and the time that the point of contact reaches the line of centres. See p. 969 for line of centres.

Arc of recess is the arc of the pitch circle through which two engaging wheels rotate from the time they are in contact on the line of centres to the time when contact ceases.

Face of tooth is that portion of the tooth surface above the pitch cylinder.

Flank of tooth is that portion of the tooth surface below the pitch cylinder.

Width of tooth face is the length of the surface of contact of two mating gears. It is usually equal to the full width of the gear-wheel.

Gear blank is the prepared cylinder or disc or truncated cone in the case of a bevel gear) before the teeth are cut.

Crests of teeth are the peripheral surfaces of the blank which are left after the teeth have been cut to size.

Tooth tip is the intersection of the crest with the face.

Tip relief is sometimes referred to as *easing*. It is the amount by which the tooth face is sometimes modified toward the tip, a small amount of material being removed.

Rounding or chamfering are provided on sliding gears to facilitate engagement, a small amount of material being removed from the edges of the tooth profiles.

Clearance curve is the curve connecting the flanks of adjacent teeth and forming the bottom boundary of the tooth space.

Fillet is the portion of the clearance curve which joins the tooth flank to the bottom of the tooth space.

Line of centres is the straight line containing the centres of rotation of two engaging gears.

Centre distance is the linear distance between the centres of rotation of engaging gears measured along the line of centres.

Hunting Tooth.—When teeth could not be so accurately produced as at present it was common practice to introduce in the gear an extra tooth to equalise the wear on the teeth as a whole. If the number of teeth in the wheel is exactly divisible by the number of teeth in the pinion the same teeth are continually coming into engagement. This, in teeth not exactly of uniform shape and size, tends to produce uneven wear, and hence the introduction of the odd or hunting tooth. The high standard of production now achieved makes this practice unnecessary so far as machine cut gears are concerned.

PROFILES OF GEAR TEETH.

The involute tooth profile is now practically universal in engineering work, although large gears having cycloidal teeth are sometimes manufactured. The involute tooth profile can be produced very accurately with comparative ease, and is most suitable for gear manufacture on a first-class production basis. The modern method of gear manufacture is by the generating process. In this system the gear blank is made to rotate about its actual axis of rotation, and at the same time a cutter having a section of a rack is made to advance across the gear face a distance equal to the pitch of the finished gear, and in a direction at right angles to the axis of the rear. The rack cutter has properly 'raked' cutting edges, and reciprocates across the face of the blank as the latter oscillates about its axis of rotation. Such a system of production tends to produce uniform tooth profiles, since each tooth is formed under similar conditions.

Slight inaccuracy in the fixing of the wheel centres does not have such serious effect upon the efficient working of involute teeth as is the case with cycloidal ones in which accurate positioning is essential. Increase of centre distance results, however, in greater back lash with consequent liability to greater fatigue in the material, particularly if the tooth load varies frequently, when in such cases a minimum back lash is essential. The wear on involute teeth is rather greater than on cycloidal teeth, since the surfaces in contact are small in the case of the former profile owing to the convex curvature.

There are several tooth forms in use such as the British Standard, Brown and Sharpe, Sellers, and Logue. Generally the 20° pressure angle is favoured and it is common practice also to modify somewhat the involute profile to suit conditions of manufacture. (See sub-section on Gear Grinding.) Reference to the British Standard tooth form also exemplifies the alterations to the involute profile. (See also paragraphs on Interference.)

Interference.

This is an important phenomenon which occurs in all kinds of gearing when the number of teeth in mating gears and tooth proportions are unsuitable. (See also sub-section on Bevel and Helical Gears.)

Interference is accentuated in gear teeth having a 14½° pressure angle; a 20° pressure angle is more suitable in order to avoid this difficulty, although a slight increase in pressure on the bearings results from its use. Since the introduction of the generating system of gear production the former pressure angle has been much less used. In cutting teeth by this process, when interference exists due to too few teeth and unsuitable proportions, the flanks of the teeth may be undercut so much so that they are seriously weakened.

Interference occurs in 14½° gear teeth when the number of teeth is below 32, and 17 in the case of 20° teeth. In the case of gears having a very small number of teeth, say below 10, the whole of the involute portion forming the flank is cut away up to the pitch circle.

Corrections for Interference in Spur Gears.

There are three methods in general use for correcting this defect, namely, (*a*) addendum correction, (*b*) reduction of addendum, (*c*) increase of the pressure angle.

The first method (*a*) combined with a suitable pressure angle is generally regarded as the best. The reduction of the addendum gives what are commonly called stub teeth, and when this method is adopted in conjunction with an increase of pressure angle the best combination is obtained to meet the *most serious* cases of interference. Usually 20° stub teeth are adopted.

Method (*a*) enables a standard height of tooth to be retained, thus permitting the use of standard cutters which facilitates production. The general method of application of this system consists in reducing the dedendum of the pinion which is probably the more seriously affected by interference and increasing its addendum, applying reverse corrections of equal amount to the engaging wheel teeth. Such corrections produce stronger pinion teeth and more efficient tooth contact. There are no standardised corrections in vogue. Each manufacturer makes his own particular allowances to suit requirements.

In the stub tooth system the addendum is commonly reduced to 0·8/*d.p.*, and the dedendum to 1/*d.p.* or 0·8 module and 1·0 module respectively.

Back Lash.

In general engineering work a certain amount of back lash is essential to allow for errors in the tooth form and in the centre distance. The complete elimination of back lash is only desirable in special cases. When the loading on the teeth fluctuates and is thus liable to produce a hammering effect between the teeth, then the back lash should be reduced to a minimum. David Brown & Sons, Ltd., Huddersfield, recommend for general engineering work a back lash of 0·03 in. to 0·05 in. for 1 diametral pitch teeth, and proportionately for other pitches.

Face Width.

The width of face of a gear depends largely upon the conditions under which it is required to work. If the power transmission is small and the speed not excessively high then the face width can be comparatively narrow. A width of from two to four times the circular pitch is common, but the correct estimate of this dimension should be made by considering the speed of the gear, pressure between teeth, materials in contact, and method of lubrication. The strength of the teeth is also affected by this dimension, the strength to resist bending being directly proportional to it. (See section on Zone Factors, etc., B.S.I. Report, No. 436.) For heavy wear the face width is often very great as in rolling mill gears where it is sometimes equal to the pitch diameter.

Setting out Involute Teeth.

Let ABCD (fig. 2) be the pitch circles of two mating gears, and EF their line of centres. Through the point of contact of the two pitch circles on EF, draw MN such that angle θ is equal to the angle of obliquity of action desired. MN is the line of action of the two gears, and is therefore the common tangent to the two base circles from which the teeth profiles are developed. Hence, with centres on EF draw the base circles *nn* and *ss*, just touching the line MN. The involute curve of which the tooth profiles are a portion is generated from these base circles.

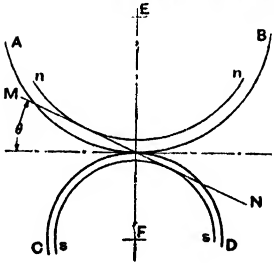


FIG. 2.

A convenient way of setting out an involute curve is to roll a notched straight edge round the base circle, marking successive positions of the notch as the straight edge rolls around the circle, until a sufficient length of the involute curve has been derived.

Proportions of Involute Gear-Teeth for Spur Gearing.

p = circular pitch ; P = diametral pitch.

BRITISH STANDARD TOOTH FORMS FOR SPUR GEARS.*

Proportions for high-class or commercial cut gears, 20° pressure angle.

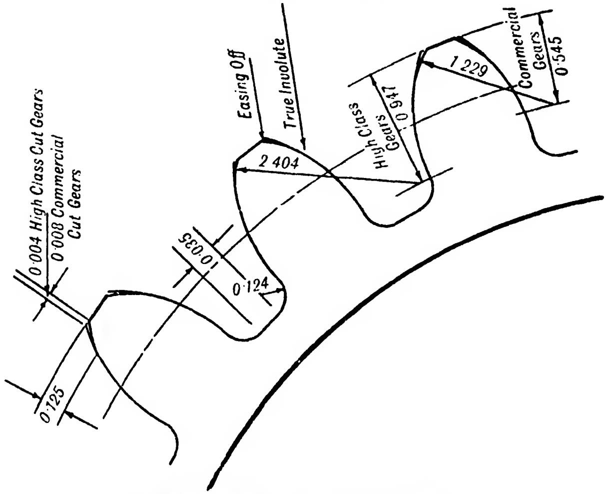
Thickness of tooth and width of tooth space at pitch circle	0·5p or 1·571/P.	Addendum	0·3183p or 1/P.
		Dedendum	0·3979p „ 1·2500/P.
		Total height of tooth	0·7162p „ 2·2500/P.

Proportions for precision ground or cut gears 20° pressure angle.

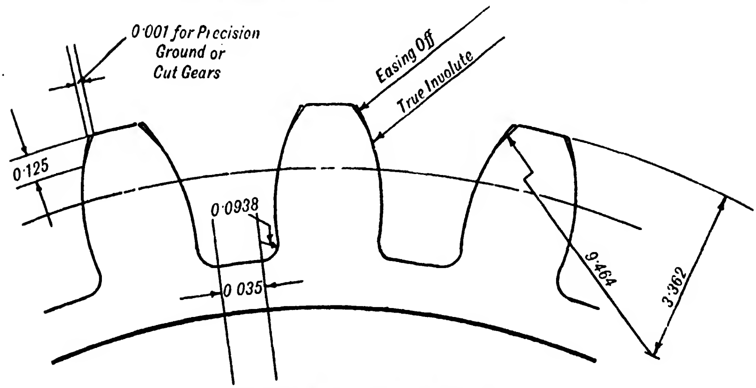
Thickness of tooth and width of tooth space at pitch circle	0·5p or 1·571/P.	Addendum	0·3183p or 1/P.
		Dedendum	0·4583p „ 1·4397/P.
		Total height of tooth	0·7766p „ 2·4397/P.

* By permission of the British Standards Institution.

Detailed proportions and tooth form are shown in figs. 3 and 4.



All dimensions are in terms of the Circular Pitch.
FIG. 3.—British Standards 20° Tooth Form for Commercial Cut Gears.



All dimensions are in terms of the Circular Pitch.
FIG. 4.—British Standard 20° Tooth Form for Precision Ground or Cut Gears.

(Machine-cut gears of 14½° and 20° obliquity.)

(Brown and Sharpe.)

*Thickness of tooth at pitch circle and width of tooth space	0·5p or 1·571/P.	Dedendum	0·3683p or 1·157/P.
Addendum	0·3183p or 1/P.	Total height of tooth	0·6866p „ 2·157/P.
		Bottom clearance	0·05p „ 0·157/P.

* Nominal thickness in each case.

Proportions of Short Involute Teeth, Logue System.

(Stub teeth, 20° obliquity.)

*Thickness of tooth at pitch circle and width of tooth space	$0 \cdot 5p$ or $1 \cdot 571/P$.	Dedendum	$0 \cdot 3p$ or $0 \cdot 942/P$.
Addendum	$0 \cdot 25p$ or $0 \cdot 785/P$.	Total height of tooth	$0 \cdot 55p$ „ $1 \cdot 728/P$.
		Bottom clearance	$0 \cdot 05p$ „ $0 \cdot 157/P$.

Proportions of Short Involute Teeth, Sellers System, 20° Obliquity.

*Thickness of tooth at pitch circle and width of tooth space	$0 \cdot 5p$ or $1 \cdot 571/P$.	Dedendum	$0 \cdot 35p$ or $1 \cdot 099/P$.
Addendum	$0 \cdot 3p$ „ $0 \cdot 942/P$.	Total height of tooth	$0 \cdot 85p$ „ $2 \cdot 041/P$.
		Bottom clearance	$0 \cdot 05p$ „ $0 \cdot 157/P$.

NOTE.—The above proportions in terms of the circular pitch are approximate. The error is very small.

In the Fellows stub tooth system two pitches are used, the tooth proportions being calculated from one and the pitch diameter of the gear from another.

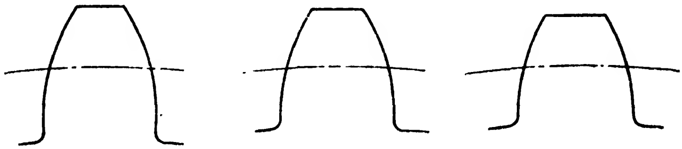


FIG. 5.—Forms of Gear Teeth.

Machine-Moulded Teeth.

Proportions for these vary. The following are typical values:—

Thickness of tooth, $0 \cdot 485p$.	Addendum, $0 \cdot 25p$ to $0 \cdot 35p$.	Total height of tooth, $0 \cdot 6p$ + $0 \cdot 8$ in. to $0 \cdot 75p + 0 \cdot 08$ in.
Width of tooth space, $0 \cdot 515p$.	Dedendum, $0 \cdot 35p + 0 \cdot 08$ in. to $0 \cdot 4p + 0 \cdot 08$ in.	

Rolling-Mill Gears.

The shape of the teeth for this purpose varies somewhat from the standard involute form, although the involute profile is closely followed. Good radii are provided at the root of the teeth, and a 22½° pressure angle is used. Usually the tooth face is exceptionally wide, often being made equal to the pitch diameter of the gear. This makes provision for the heavy wear to which these gears are subjected.

Typical tooth proportions are as follows:—

Addendum, $0 \cdot 275p$; dedendum, $0 \cdot 325p$; width of tooth at pitch circle, $0 \cdot 42p$.

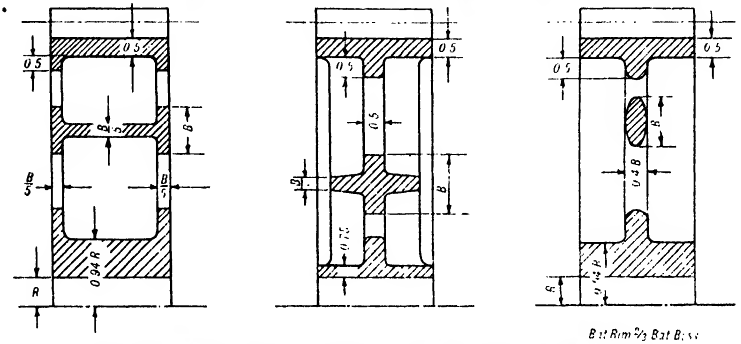
Proportions of Gear Wheel Bosses, Arms and Rims.

The British Standards Specification No. 436 contains a number of charts which enable the dimensions of the boss rim and arms to be calculated. Proportions are given in fig. 6 for heavy types of cast iron gear wheels. In the case of cast steel gears of good quality the dimensions obtained from these proportions can be reduced on the grounds that in tension steel castings are approximately 4½ times as strong as iron castings.

The dimensions of the arms of a gear wheel should, when the load to be transmitted is appreciable, be determined by calculating the bending moment to which they are subjected. Assuming the loading between the teeth to be steady then if HP = horse power transmitted, V = linear velocity of a point on the pitch circle in ft. per min., and W = tangential load between the teeth in lb. at the pitch circle then HP = W.V/33000 and W = 33000 HP/V. This load may be increased due to the starting torque being greater than the mean running torque and also it may be greater due to external influences. Assuming that the arms are straight and take an equal

* Nominal thickness in each case.

share of the load then the bending moment on each arm at its root = Wl/n where l = length of arm between the boss and rim and n = No. of arms, n is usually 4, 6 or 8 according to the size of the wheel. Then $Wl = fz$ where f is the allowable bending stress in lbs. per sq. in. which may be up to 2,500 for good cast iron and 9,000 or 10,000 for cast steel. Z is the modulus of section (see



Proportions are in terms of Circular Pitch except where otherwise stated.

FIG. 6.—Proportions for Cast Iron Gear Wheel Arms and Rims. See also B.S.I. Report No. 436.

appropriate tables) involving the dimensions of the cross-section of the arm at its root and depending upon the shape of the cross-section. One of the dimensions of the cross-section will in most cases be governed by the width of tooth face and hence the other may be calculated. If the arms are curved the actual stress produced by a given load at the pitch circle may be 60 to 70 per cent. greater than that calculated from the formulæ given for straight arms. The dimensions of the cross-section of the arms may be reduced at the rim where the bending moment is obviously much less than at the boss.

Involute Gear Cutters.

Since the dimensions of gear teeth are proportional to the circular or diametral pitch in the standard tooth forms, it follows that, theoretically, a different cutter is required for gears having different numbers of teeth.

In actual practice, however, a set of 8 cutters is employed for the 14½° system, and another set of 8 for the 20° system.

The cutters are employed as follows and are numbered from 1 to 8, each one being known by its particular number:—

No. 1	for cutting	gears	having	135	teeth to	a rack	
No. 2	,,	,,	,,	55	,,	134	teeth
No. 3	,,	,,	,,	35	,,	54	,,
No. 4	,,	,,	,,	26	,,	34	,,
No. 5	,,	,,	,,	21	,,	25	,,
No. 6	,,	,,	,,	17	,,	20	,,
No. 7	,,	,,	,,	14	,,	16	,,
No. 8	,,	,,	,,	12	,,	13	,,

The shape of the cutter conforms to that of the gear having the smallest number of teeth.

The above cutters are satisfactory for gears working under general conditions where special attainments are not needed, and for gears working under steady loading conditions.

For more accurate work and more precise conditions, cutters bearing half numbers are made as follows :—

No. 1½	for cutting	gears	having	80	teeth to	134	teeth
No. 2½	,,	,,	,,	42	,,	54	,,
No. 3½	,,	,,	,,	30	,,	34	,,
No. 4½	,,	,,	,,	23	,,	25	,,
No. 5½	,,	,,	,,	19	,,	20	,,
No. 6½	,,	,,	,,	15	,,	16	..
No. 7½	,,	,,	,,	13 teeth only.			

Cycloidal Teeth.

Small amount of wear occurs with this form of tooth profile due to large surface of contact. To ensure quiet running and no back lash it is essential that the wheel centres be accurately located, since the path of contact consists of two curves which change direction at the pitch point.

These gears are difficult to cut accurately in a machine. and if the profiles of pinions of 12 teeth to a rack are generated with rolling circles greater in diameter than half the diameter of the smallest pinion, then the flanks cannot be cut by the ordinary rotating cutter.

Gears of this form have good wearing qualities, but the advent of higher speeds of rotation have made the defects of these teeth very pronounced, and involute gears have superseded them.

Epicyclic Gears.

In epicyclic gearing the arm rotates, and hence its motion must be added to that of the wheels, In general, the action of epicyclic gear trains can be summarised as follows : The number of revolutions of any gear-wheel in the train for each single revolution of the arm is equivalent to the number of revolutions that the gear-wheel would make if the arm was fixed and the first gear-wheel turned through 1 revolution + 1 for wheels rotating in the same direction, and − 1 for wheels rotating in the reverse direction to that of the arm.

With epicyclic gears high velocity ratios can be obtained in a compact and convenient form.

To Determine Relative Motion of Components in a Simple Epicyclic Gear.

In fig. 7 the number of teeth in each member of the system is denoted by N_a, N_b, N_c, and A represents the arm rotating about the principal centre and carrying pinion, N_c. To compile the following tables the driver is given one revolution, and the corresponding motion of the other wheels and arm are tabulated. The negative sign indicates motion in the opposite direction to that of the driving wheel.

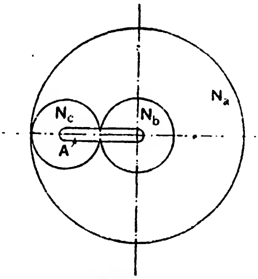


FIG. 7.

Member Fixed.	Member Driving.	Member Driven.	No. of Revs. of A.	No. of Revs. of N_b.	No. of Revs. of N_c. About own Axis.	No. of Revs. of N_a.
A	N_b	N_a	0	1	$-\frac{N_b}{N_c}$	$-\frac{N_b}{N_a}$
A	N_a	N_b	0	$-\frac{N_a}{N_b}$	$\frac{N_a}{N_c}$	1
N_a	A	N_b	1	$\frac{N_a+N_b}{N_b}$	$-\frac{N_a}{N_c}$	0
N_b	A	N_a	1	0	$\frac{N_b}{N_c}$	$\frac{N_a+N_b}{N_a}$
N_a	N_b	A	$\frac{N_b}{N_a+N_b}$	1	$-\left(\frac{N_b}{N_a+N_b}\right)\frac{N_a}{N_c}$	0
N_b	N_a	A	$\frac{N_a}{N_a+N_b}$	0	$\left(\frac{N_a}{N_a+N_b}\right)\frac{N_b}{N_c}$	1

Relation between Circular Pitch and Diametral Pitch.

$p.$ = circular pitch; P = diametral pitch; $p.c.d.$ = pitch circle diameter. N = number of teeth; π = 3·1416.

$$p. = \frac{\pi \times p.c.d.}{N}; \quad P = \frac{N}{p.c.d.}$$

Hence,

$$p. \times P = \pi; \quad p. = \frac{\pi}{P}; \quad P = \frac{\pi}{p.}$$

To Calculate the Linear Velocity of the Pitch Circle.

Multiply the pitch circle diameter in inches by π (3·1416) and this product by the number of revolutions per minute of the gear. Divide the result by 720 which gives the linear velocity of the pitch circle in feet per second.

To Calculate the Chordal Pitch of a Straight Spur Gear.

Divide 180° by the number of teeth in the gear. Find the sine of the resulting angle (from trigonometrical tables) and multiply the sine by the pitch diameter of the gear in inches, thus obtaining the chordal pitch in inches.

Tooth Load.

The load on the teeth at the pitch circles of two gears in engagement due to the power transmitted may be calculated from the following: H.P. = horse-power transmitted: V = linear velocity in *ft. per min.* of a point on the pitch circle of one of the gears: D = the diameter in feet of the pitch circle of one of the gears and N = its number of revs. per min: W = load in lbs. between the teeth in engagement.

$$V = \pi DN. \qquad W = \text{H.P. } 33{,}000/V.$$

To Compute the Horse-Power transmitted by a Gear.

Multiply the tangential load in lbs. at the pitch line by the pitch circle velocity in ft. per min. and divide the product by 33,000.

Example.—A gear 8 ins. *p.c.d.* runs at 400 r.p.m. and carries a pitch line load of 800 lbs. What horse-power is transmitted?

$$\text{Pitch circle velocity} = \frac{\pi \times p.c.d. \times N}{12} = \frac{3{\cdot}1416 \times 8 \times 400}{12} = 837\ \textit{ft. per. min.}$$

$$\text{H.P.} = \frac{800 \times 837}{33{,}000} = 20{\cdot}3.$$

The load W referred to above is not necessarily the max. load on the teeth since its calculation is based on the mean torque transmitted. It is, however, common practice for example to make allowances for possible dynamic increases in the tooth load by modifying the working stress allowed when designing the teeth. The tooth strength is calculated on the principle that each tooth acts as a cantilever and carries the load W (referred to above) at the apex of the tooth, and in a direction perpendicular to its radial centre line.

Thus if h = height of tooth in ins. Wh = bending moment in in. lb. Also if the face width in ins. = f, b = the breadth of the tooth at its root in ins. and s = the allowable working stress in lb. per sq. in. which will depend on the material used and the speed of the gear, then

$$Wh = \frac{b^2 fs}{6} \text{ or } f = 6Wh/b^2 s.$$

Generally b and h are fixed from general proportions. The above forms the basis of the well-known Lewis formula for gear teeth.

Lewis Formula.

$$\text{As above } Wh = \frac{b^2fs}{6} \text{ or } W = \frac{b^2fs}{6h}$$

Let p = circular pitch of teeth in ins. Then if c and k are constants, $b = cp$ say, and $h = kp$ say,

$$\therefore W = \frac{c^2p^2fs}{6kp} = \frac{c^2pfs}{6k}.$$

$\frac{c^2}{6k}$ is usually denoted by 'y,' and is called the Lewis strength factor.

$$\therefore W = ysfp, \text{ which is the Lewis formula.}$$

When calculations are made in terms of the diametral pitch, Y is used to denote the Lewis factor. In this latter case $W = \frac{Ysf}{P}$ and $Y = \pi y$, and P = diametral pitch.

The values assigned to s, the safe working stress, are based on a basic value of the stress, which is actually the safe static stress suitable for the particular materials in use. This basic value is then modified to allow for the pitch circle velocity of the gear. In the D.B.S. system the modifying factor is based on the revolutions per minute, and not on the pitch circle velocity.

A common formula in use connecting basic stress and pitch circle velocity is

$$S_v = S_o\left(\frac{600}{600 + V}\right)$$

where S_v = safe stress at given pitch circle velocity.

S_o = safe basic stress. V = pitch circle velocity in ft./min.

Modern methods in vogue for producing accurate tooth profiles and the increased velocities made possible by these conditions, have resulted in the above formula being modified thus :—

$$S_v = S_o\left(\frac{1{,}200}{1{,}200 + V}\right)$$

and in this form it meets the case of accurate machine-cut teeth working at high velocity. The American Gear Manufacturers' Association recommends the following formula for gears working with pitch circle velocities of 4,000 ft. per min. and over :

$$S_v = S_o\left(\frac{78}{78 + \sqrt{V}}\right).$$

The following are suitable basic stress values for use in the above formulæ :—

Material.	*Basic Bending Stress.*
Nickel chrome steel, 100 tons U.T.S.	55,000 lb. per sq. in.
,, ,, ,, 60 tons ,,	33,000 ,, ,,
3½ per cent. nickel steel, case-hardened and heat treated	25,000 ,, ,,
0·5 per cent. carbon steel, 50 tons U.T.S.	27,000 ,, ,,
0·4 ,, ,, 40 ,, ,,	22,000 ,, ,,
0·3 ,, ,,	18,000 ,, ,,
*Mild steel	15,000 ,, ,,
Phosphor bronze	12,000 ,, ,,
Cast iron	6,000 ,, ,,
Raw hide	3,000 ,, ,,

* See page 981.

Lewis Strength Factors.

No. of Teeth.	Involute 20° Obliquity. y	Involute 20° Obliquity. Y	Involute 14½° Obliquity and Cycloidal. y	Involute 14½° Obliquity and Cycloidal. Y	Radial Flanks. y
12	0·078	0·245	0·067	0·211	0·052
13	0·083	0·261	0·070	0·220	0·053
14	0·088	0·276	0·072	0·227	0·054
15	0·092	0·289	0·075	0·236	0·055
16	0·094	0·296	0·077	0·242	0·056
17	0·096	0·302	0·080	0·252	0·057
18	0·098	0·307	0·083	0·261	0·058
19	0·100	0·314	0·087	0·274	0·059
20	0·102	0·321	0·090	0·283	0·060
21	0·104	0·326	0·092	0·289	0·061
23	0·106	0·333	0·094	0·296	0·062
25	0·108	0·340	0·097	0·305	0·063
27	0·111	0·349	0·100	0·314	0·064
30	0·114	0·358	0·102	0·321	0·065
34	0·118	0·371	0·104	0·326	0·066
38	0·122	0·384	0·107	0·336	0·067
43	0·126	0·396	0·110	0·346	0·068
50	0·130	0·408	0·112	0·352	0·069
60	0·134	0·421	0·114	0·358	0·070
75	0·138	0·434	0·116	0·365	0·071
100	0·142	0·446	0·118	0·371	0·072
150	0·146	0·459	0·120	0·377	0·073
300	0·150	0·472	0·122	0·384	0·074
Rack	0·154	0·484	0·124	0·390	0·075

In the D.B.S. system (David Brown & Sons) higher values of 'y' are taken in the Lewis formula and the speed coefficient is based on the r.p.m. and not on the pitch-circle velocity. This latter substitution makes allowance for repetition stresses.

Thus

$$y = 0{\cdot}124 - \frac{0{\cdot}648}{T} \text{ for } 14\tfrac{1}{2}^\circ \text{ pressure angle.}$$

and

$$y = 0{\cdot}154 - \frac{0{\cdot}912}{T} \quad ,, \quad 20^\circ$$

where T is the number of teeth in the gear.

The British Standards Institution Report* gives the formula for tooth strength in a somewhat different form although fundamentally the same as that quoted previously. It can be written thus :—

$$W/f = \text{strength factor} \times \text{working stress}/P.$$

The value of the working stress is calculated from considerations of the speed and material of the gear. Tables and charts are given with the report to enable suitable working stresses, etc., to be selected.

Davey's Modification of Lewis Formula.

For the application to gearing problems where it is intended to use circular pitches,

$$W = ysf/p.$$

For similar problems where it is intended to use diametral pitches,

$$W = \frac{Ysf}{P}$$

* B.S.I. Specification No. 436.

Most practical problems consist of determining a suitable pitch of teeth to transmit a given horse-power, together with the width of face, which in order to solve the equation for any particular case, is given some value in terms of the pitch.

In a practical problem the approximate pitch circle diameter is usually known, or can be fixed according to the conditions ruling.

In applying the Lewis formula to the problem of finding the pitch, we have two unknowns—(a) Lewis factor; (b) pitch of teeth. The Lewis factor depends upon the number of teeth, as also does the pitch for a given pitch circle diameter. It is common to *assume* a number of teeth and select from a table the corresponding Lewis factor. If the number of teeth assumed is incorrect as shown by the application of the formula, then fresh values for the number of teeth have to be selected until a suitable pitch is obtained consistent with strength. The method is, therefore, one of trial and error, and requires considerable judgment and experience in order to obtain suitable results.

In order to obviate this, the formula can be written as follows :—

Let,

Width of face $f = 3p$ (which agrees with average practice)

then, for circular pitch gears,

$$W = 3ysp^2.$$

Also,

$$p = \frac{\pi D}{N};$$

where,

N = number of teeth; D = pitch circle diameter

and,

$$W = \frac{3ys\pi^2D^2}{N^2}; \quad \frac{y}{N^2} = \frac{W}{3\pi^2D^2s}.$$

Multiplying the right-hand side of equation by 10,000, adjusting values of $\frac{y}{N^2}$ accordingly we get

$$\frac{y}{N^2} = \frac{337\cdot7W}{D^2s},$$

which permits easier tabulation and facilitates reading.

For diametral pitch gears,

$$W = \frac{sfY}{P} = \frac{sfYD}{N}; \quad W = \frac{3\pi^2D^2sY}{N^2}; \quad \frac{Y}{N^2} = \frac{W}{3\pi D^2s}.$$

Or, multiplying the right-hand side by 10,000,

$$\frac{Y}{N^2} = \frac{1061W}{D^2s}.$$

If other values for the width of face are required the formulæ become

$$\frac{y}{N^2} = \frac{1013\cdot2W}{KD^2s} \text{ for circular pitches,}$$

and

$$\frac{Y}{N^2} = \frac{3183W}{KD^2s} \text{ for diametral pitches.}$$

$$K = \frac{\text{width of face}}{\text{circular pitch}}, \text{ a value to be decided upon initially.}$$

If values of $\frac{y}{N^2}$ and $\frac{Y}{N^2}$ are tabulated, then it becomes unnecessary to assume the number of teeth, since this can be seen at a glance from a table when the above calculation has been made, and the required pitch of teeth can then be found for the given pitch circle diameter.

Some slight adjustments may be necessary in order to make the pitch such as to enable the gear to be cut by the usual standard cutters.

No. of Teeth.	20° Involute Teeth.		14½° Involute and Cycloidal Teeth.	
	$\frac{y}{N^2}$	$\frac{Y}{N^2}$	$\frac{y}{N^2}$	$\frac{Y}{N^2}$
12	5·42	17·0	4·65	14·6
13	4·91	15·4	4·14	13·0
14	4·49	14·1	3·67	11·6
15	4·41	12·8	3·33	10·5
16	3·67	11·5	3·00	9·45
17	3·32	10·5	2·77	8·69
18	3·02	9·47	2·56	8·05
19	2·77	8·69	2·41	7·59
20	2·55	8·02	2·25	7·07
21	2·35	7·39	2·09	6·55
23	2·00	6·29	1·77	5·59
25	1·72	5·44	1·55	4·88
27	1·52	4·78	1·37	4·31
30	1·26	3·98	1·13	3·56
34	1·02	3·21	0·90	2·82
38	0·845	2·66	0·741	2·32
43	0·681	2·14	0·595	1·87
50	0·520	1·63	0·448	1·40
60	0·372	1·17	0·316	0·994
75	0·245	0·771	0·206	0·648
100	0·142	0·446	0·118	0·371
150	0·0644	0·204	0·0533	0·168
300	0·0166	0·0524	0·0135	0·0426

Tooth Strength.

The continued increase in speed and the more exacting conditions demanded in gearing practice together with the wider application to engineering problems, has necessitated a closer study of the behaviour of materials under working conditions, and a more exact method of determining the actual loading to which the teeth are subjected.

The Lewis formula is based entirely upon the assumption that the load due to the power transmitted is applied at the tooth point, and that each tooth in turn carries the full load at this point. Such an assumption is in practice seldom justified. The load is more often transmitted from *two* teeth to two mating teeth. Actually the whole of the load is not often carried by any one tooth until the point of contact has travelled some distance along the profile.

In the D.B.S. system and as recommended in the B.S.I. Report, No. 436, a *tooth strength factor* is employed in determining the load carrying capacity of teeth.

The *tooth strength factor* is the tangential load which will produce a bending stress of 1 lb. per sq. in. on teeth of 1 D.P. 1 in. wide when the teeth are assumed to act as cantilevers.

It is well known that a pinion with few teeth is stronger when gearing with a rack than when gearing with a pinion having an equal number of teeth. Hence the latter is a more reliable method of comparing tooth strength than the former, which assumes the load to be applied at the tooth point.

An examination of the foregoing equation will at once reveal a similarity to the Lewis formula. The important points about this latter equation are (*a*) the method of calculating a suitable working stress to meet the given conditions, and (*b*) the method of determining the tooth strength factor '*y*.'

Surface Stress.—In addition to the bending stress it is important that the surface stress to which the tooth faces are subjected should not be excessive. If the surface stress is too intense failure may occur from crushing or flaking of the material. The basic value * of the surface stress, which varies according to the materials in contact, when multiplied by a speed coefficient gives the allowable surface stress S_c suitable for the given conditions.

* See British Standard Specification No. 436—1940.

Zone Factor* 'Z.'—The permissible tooth load for any pair of gears depends upon the face width, surface stress, relative curvature of tooth surfaces, pressure angle, and the pitch diameters of the gears. These variables are combined to give what is known as the *zone factor*. Thus the value assigned to Z depends upon the number of teeth in a combination, and varies accordingly. The permissible tooth load per in. of face width $= \frac{S_s Z}{D.P.}$. To obtain the values of S_s and Z charts are used.

Increment Load and Static Load.

A report of researches conducted in America was presented to the American Gear Manufacturers' Association toward the end of 1930. The report deals with experimental work conducted on the Lewis testing machine to verify or modify the method by which the stress on gear teeth is determined.

Taking the Lewis formula, $W = ysfp$, and the stress formula, $S_v = S_o \left(\frac{600}{600 + V}\right)$ and putting $yfp = K$, we have $W = K\left(\frac{600}{600 + V}\right)S_o$, which can be rewritten thus:

$$S_o = W\left(\frac{600 + V}{600K}\right) = \frac{1}{K}\left(W + W\frac{V}{600}\right).$$

The first item, W within the brackets, is the static load which can be applied to the gear, while the second item is that usually known as the increment load, which arises due to errors in tooth form, inaccuracy of tooth spacing, etc.

The equation shows the latter to be directly proportional to the static load W, a fact which in the light of these researches is erroneous in the case of high-speed gearing. The increment load depends upon inaccuracies in gear cutting, speed, and nature of material. The most common materials used seem to make the increment load practically independent of the static load. The former is apparently dependent upon the modulus of rigidity of the material used, since in the case of cast-iron gearing the increment load is more dependent on the applied load than is the case with steel gears. This is probably accounted for by the fact that the material with the lower modulus of rigidity is better able to absorb shock.

The increment loads given by the old formula are in many cases much in excess of the new values obtained in these researches, particularly in the case of the highly rigid materials such as the steels now employed.

Internal Gears.

When space is limited convenient use can often be made of internal gears, since with these gears the centre distance is reduced owing to the pinion being inside the wheel. The arrangement of the bearings may be complicated, however, by the use of this form of gearing. The teeth have in general greater strength than a spur gear for the same purpose.

For ratios greater than 2 to 1 the teeth can be of standard proportions if the number of teeth is sufficient to overcome the difficulty of undercutting. Thus, if 14½° pressure angle is employed there must not be less than 32 teeth and 17 for 20° pressure angle.

Exceptional interference occurs in this form of gearing when the velocity ratio is less than 2 to 1. It can, however, be ignored if the difference between the number of teeth in the wheel and pinion is not less than 14 for 14½° teeth, and 8 for 20° teeth.

Gearing on Armature Shafts.

In gearing, large revolving masses—as, for example, armatures of motors—should be separated from adjoining masses, such as large gear-wheels of the second motion, by some elastic medium, such as a length of shafting or a flexible coupling. This is intended to allow for slight variations in the angular velocity, due to unavoidable errors in cutting the teeth, to take place without giving rise to destructive loads between the teeth. The stresses due to inaccuracies in the spacing of the teeth may exceed the working stress. With an error of only 0·2 per

* See British Standard Specification, No. 436—1940.

cent. in the pitch the excess load due to acceleration and retardation may amount to as much as 28 times the load transmitted. Such loads increase as the square of the mean pitch-circle velocity and as the mass of the wheel considered.

Gearing Noises.

During the last few years considerable attention has been given to noise produced by gears. The kind of noise produced by a pair of gears will often give some indication of defects present.

High-class gearing running in perfect order produces a low-pitched humming sound with absence of beats, the sound being quite 'smooth' and continuous.

The greater the accuracy in machining the teeth and the better-class finish put upon the tooth faces, all tend to reduce noise. Irregular spacing of teeth is a common defect, and can be recognised by an intermittent clicking or a pulsating howl. Similar sounds occur when teeth are mis-shaped.

A pulsating howl may also arise due to improper setting of the gears upon their spindles, torsional vibrations, or lack of static or dynamic balance.

High-pitched screeching often occurs, due to rough tooth faces. A common defect, particularly in heavy gearing, is to use either too little support for the shafts or too small a shaft diameter. Such errors set up spring in the shafts, resulting in pulsating sounds of varying pitches according to the pitch-circle velocities.

A good condition of a set of gears, particularly those running at high speed, is indicated by a more or less musical sound, or sounds of constant pitch.

In the case of gears which are required to run in either direction, the fact that noise prevails more in one direction than another indicates that the teeth are not radial.

Generally, fine-pitched teeth and wide face widths tend toward noise reduction.

Critical Speeds of Gearing.

Certain conditions of gear-teeth surfaces after running for some time cannot be directly traceable to general causes. Results of investigations go to show that the running of gear shafts at speeds too near that of the critical speeds results in the development of rough tooth profiles and general deformation of the surfaces in contact. Vibrations are often manifest in gearing, particularly in the case of high speeds. These will be considerably increased if they harmonise with those of the shaft at its critical speed, and such conditions may produce fracture.

In cases where gears are running at or near the critical speeds of their shafts, unsatisfactory running will invariably result. Hence dimensions of shafts must be arranged so that this defect is avoided.

Approximate rule for determining critical speed :—

$$S_c = \frac{188}{\sqrt{\delta}}$$

where,

S_c = critical speed of shaft in r.p.m.; δ = max. deflection of shaft in ins.

If a gear is required to run above the first critical speed, then when starting up the gear must be allowed to pass quickly over this critical speed period. In general, the working speed of a gear should not be within 28 per cent. or 30 per cent. of the critical speed.

Non-Resonant Pinions.

In order to reduce noise when gearing is to run at high speed, it is common practice to employ pinions made of such materials as raw-hide, fibre, 'fabroil,' and 'bakelite.' Such pinions are usually made with metal end re-enforcing plates, which clamp together the softer non-resonant material, and thus prevent spreading at the ends of the tooth faces. For raw-hide pinions the maximum tooth pressure per in. width of tooth face is about 250 lb., the pressure being decreased as the pitch line velocity increases. It is important to keep oil and grease away from raw-hide pinions, as mineral oil proves most destructive. A dressing of raw linseed-oil occasionally keeps the material in good condition and preserves it.

Cast Iron and Mild Steel Gears.

During the past few years the quality and strength of cast iron have been much improved. Whereas 13 to 16 tons per square inch ultimate strength used to be considered good, 17 to 21 tons per square inch ultimate strength is now procurable without alloy. Alloy cast irons with ultimate strength up to 30 tons per square inch can now be obtained at increased cost. For certain gears cast iron has advantages. For example, free graphite in the iron tends to provide for smooth running while phosphorous increases the length of life by resisting wear without prejudice to the other qualities if in correct proportion. Cast-iron blanks should be as hard as possible consistent with machining requirements. A Brinell number between 170 and 220 is most suitable, while for sand casting the carbon content should not exceed 3·4 per cent. A phosphorous content up to 0·8 per cent. gives good wearing quality.

Common commercial mild steel having a carbon content of 0·1 per cent. to 0·25 per cent. is not recommended for gears required to carry appreciable tooth loads. Such material does not possess good enough wearing properties. For gears which do not require to be of very great strength a 0·35 per cent. to 0·45 per cent. carbon steel is more satisfactory and costs little more than mild steel ones. Pinions of this material working with cast-iron wheels give very satisfactory results under low loading conditions.

Gear Tooth Failure.

Failure of a gear tooth may be due to fracture or a breakdown of the tooth face due to excessive wear. Either may be caused by defective material or use of unsuitable materials for the particular purpose. Improper design of teeth and insufficient accuracy consistent with speed of gears may cause failure, while misalignment caused by bad workmanship, distorted mountings, and inadequate bearings will also cause or contribute to failure. Tooth wear on an excessive scale may be due to lack of or bad lubrication and unsuitable lubricant. These latter defects may cause excessive running temperatures. Overloading caused directly or by torsional vibrations or by inefficient shaft couplings may contribute seriously to tooth failure.

Running Temperatures of Gear Units.—Temperatures up to 200° F. are satisfactory. A rise of 100° F. is safe if units are well designed, provided always that lubrication is adequate and efficient and that correct lubricant is used. Excessive rise of temperature may be due to faulty design of casing resulting in insufficient dissipation of heat due to improper circulation of lubricant. If a cooler is in use this may be defective. Excessive depth of lubricant with consequent loss of power by churning of lubricant may produce appreciable rise of temperature. Insufficient clearance between gears and casing increase risk of churning action.

The Hardening of Gears.

The more exacting requirements demanded from gears in modern engineering practice has led to a very close study of gear hardening within the last few years. Modern systems of gear-tooth grinding as a finishing process, and the careful selection of suitable materials for gear manufacture has enabled the difficulty of warping during the hardening process to be largely overcome.

Case Hardening.—The selection of suitable steels for case hardening is of primary importance, while it is hardly less important to use a suitable carburising compound for the hardening process. In regard to the latter, considerable effort has been expended in the production of suitable compounds since those used in general engineering work are not usually of sufficiently high merit for first-class precision gear practice. Firms specialising in gear manufacture are in most cases prepared to advise on the question of selecting suitable carburising compounds.

The most suitable steel for gear manufacture, when case hardening is required is a carbon steel with low carbon content. The *absence* of dissolved oxides in the steel is an essential feature, since the heat generated during the grinding process may be sufficient to impair the hardness if such are present.

Steel manufactured by the acid open-hearth process is regarded by most authorities as the best, although under exceptional circumstances in manufacture basic open-hearth steel may be used.

The following specification for steel for case hardening is recommended by David Brown & Sons Ltd., for all work except the very lightest.

	Minimum. Per cent.	Maximum. Per cent.
Carbon	0·1	0·2
Silicon	—	0·2
Sulphur	—	0·04 (0·06 or acid)
Phosphorus	—	0·04 „ „ „
Manganese	0·5	1·00

The normal heat treatment for such steel after the carburising process is a quench in water at 900° C., followed by a water quench at 750° C. Such treatment should give on a ¾ in. to 1½ in. bar the following values:—

	Minimum.	Maximum.
Yield point .	18	28 tons per sq. in.
Maximum stress	30	40 „ „ „
Izod Impact .	60 foot lb.	—
Brinell hardness No.	143	180

For work of delicate nature the maximum carbon content should be 0·12 per cent., and manganese between 0·35 and 0·60 per cent.

In many cases it is necessary to provide a stronger core than that given by the foregoing steel, in which case an alloy steel is employed. The most common is a 3 per cent. nickel steel, while 5 per cent. nickel, nickel chrome, and chrome-vanadium steels are in demand to meet high stresses. Messrs. David Brown & Sons recommend the following specification for 3 per cent. nickel steel.

	Minimum. Per cent.	Maximum. Per cent.
Carbon	0·10	0·20
Silicon	—	0·30
Sulphur.	—	0·05
Phosphorus . .	—	0·05
Manganese . .	0·20	0·70
Nickel	3·00	3·50

With this steel oil-hardening at 760° C. is recommended, and the following values on ¾ in. to 1½ ins. bars should be obtained:—

	Minimum	Maximum.
Yield point . . .	24 tons per sq. in.	
Maximum stress . .	36	50 tons per sq. in.
Izod impact . . .	40 foot lb.	—
Brinell hardness No. .	156	229

For 5 per cent. nickel steel heat treat as above but at 740° C.

Tempering.—In order to secure best results in case-hardened gears, particularly when the teeth are to be ground, a low-temperature tempering process is employed. The treatment consists of immersing the gears in an oil bath of liquid tallow or oil at about 140° - 150° C. The size of the job will govern the time of immersion, which should not be less than 1 to 1½ hours.

Local Hardening.

Warping is almost inevitable when case hardening is employed, but is almost entirely eliminated by use of the oxy-acetylene blow-pipe. Earlier methods of this type depended too much upon the individual capability and knowledge of the operator for consistent results. The Shorter * process now being extensively used and developed combines effectiveness and consistency, made possible by mechanically-controlled flame movement and special burner. The process is suited to steel which can be effectively heat-treated. The carbon content for steel forgings and castings being 0·4 to 0·7 per cent. and 0·48 to 0·55 per cent. respectively; manganese in each case should be 0·4 to 0·8 per cent. The majority of alloy steels are also suited to the process. Local hardening is applied only to the surfaces of the teeth in actual contact, and the depth of hardness is controllable, ranging from 30 to 125 thousandths. Thus the portion of the teeth and gear not influenced by wear in the same way are left in their natural condition.

Nitrogen Hardening Process.

One of the most recent advances in gear-hardening is that in which steel is heated in contact with nitrogen. It is particularly suited to gearing since distortion is extremely small and except in special cases is insufficient to need correction. For absolute accuracy the grinding allowance need only be very small, and hence from this point of view the method is economical. Another important feature is the retention of the hardness at temperatures up to 500° C., which is impossible with ordinary case-hardening methods. No quenching is needed after heating, and the process is carried out at low temperature, hence the reason for the small amount of distortion.

The material used for treatment by this process is known as Nitralloy * steel, and after treatment possesses a hardness and wear resistance which surpasses anything yet known. The alloy is a chromium-aluminium-molybdenum steel, and in order that the material shall possess the required physical properties it is usually heat treated to the maker's specification before the final machining operation is carried out. These properties are not affected by the nitrogenising process. Other steels do not harden effectively by this process. The heating and exposure of the work to the ga is carried out in high nickel stainless steel boxes. The work is heated to 500° C., and maintained at that temperature for 30 to 100 hours, and at the same time is acted upon by ammonia gas. The time of treatment varies according to the depth of hardening required. The parts are then

* See page 1189.

allowed to cool (without quenching), and the resulting thin film of soft matter is removed by emery cloth or polisher. The maximum depth of the case obtainable by this process is about 32 thousandths.

Other Methods of Hardening Gears.

A successful process of hardening automobile gears is given by the *American Machinist*. It is claimed that a minimum amount of warping occurs, and the shrinkage of the shaft holes is inappreciable. The average case thickness is approximately 25 thousandths of an inch.

The gears are preheated in a large gas-oven or forge until a dull red heat is obtained; temp. about 700° to 1000° C. The gears are then boiled in a solution of cyanide of potassium for about 1 hr. After this treatment the gears are plunged into a bath of fish oil which is kept constantly in circulation to ensure uniform temperature throughout.

The hardening and tempering of alloy steels by quenching and drawing is sometimes employed. Chrome-nickel and chrome-vanadium alloy steels are used for the purpose. The method of heat treatment, etc., should in all cases be carried out according to the steel-makers' instructions for hardening and tempering.

Grinding of Gear Teeth.

The necessity of running gears at high pitch circle velocities has given rise to the need for finer and more accurate finishing of tooth profiles. Hardened steel gears used under severe conditions are ground. At high pitch line velocities small defects in the tooth faces are very detrimental to efficient and continuous working.

Defects in tooth faces produce noise, and also reduce the load that can be transmitted by a tooth.

As compared with cut gears, those which are ground produce a characteristic high-pitched note when running under load and at high pitch line velocity. Even fine cut grinding-wheels do not produce a sufficiently smooth surface to entirely eliminate noise, but after ground gears have been in service for a short time they should, if accurately cut, run almost noiselessly.

When gears are to be ground, errors due to the wearing of the grinding-wheel must be guarded against. The material from which the abrasive wheel is made must be carefully selected. No specific recommendation can be given as details affect each particular case, as also does the material to be ground.

Ground gears are now being used extensively in automobile work, for power transmission in electric locomotives, steam lorries, and also in high-class machine-tools. Aero-engine reduction gears and those for the timing on oil engines are also invariably ground to a high degree of accuracy. In all cases where heavy loads have to be transmitted at high speed the grinding of the tooth profiles is an essential feature in order to secure high efficiency of transmission, silence, and lasting wear. The actual grinding process and the accuracy to be obtained is comparatively easy when modern machines are employed. The operation is certainly not beyond the capability of any gear manufacturing shop, although specialists in the process are often employed or the work sent to them for this final operation.

Methods of Gear Grinding.

At the present time there are two principal methods in vogue for the grinding of gear teeth, and in each the basic principle is quite different from that of the other. The two methods are known as the *Formed Wheel* method and the *Generating* method.

The latter is *geometrically* the correct method of grinding the teeth, but so far as the finished products are concerned there is apparently little to choose between the two methods. In either case the correction of the wear of the abrasive wheel is the all important feature, and in the two representative types of machines in general use, this point has undoubtedly received most careful consideration in order to eliminate every possible error.

The Formed Wheel Process.

The best example of a machine working on this principle is the Orcutt gear grinder, in which an abrasive wheel shaped to the tooth profiles is made to traverse the tooth spaces.

Correction for wear of the abrasive wheel is effected by a pantagraph system in which three diamonds (fig. 8, A, B, C) are employed to 'true' the wheel. Two diamonds serve to correct the involute profiles of the wheel, while a third mounted above the wheel corrects its periphery, (see fig. 8). The two profile diamonds are automatically operated by power, but the third trimmer is hand operated. In the latest design of this machine the abrasive wheels are cut to a radius on the periphery instead of being made to the usual shape of the bottom of the tooth spaces, as was the case in the earlier design of the machine. Machines, however, can be obtained with correcting mechanism for retaining the normal tooth space if so desired. In these latter the third trimming diamond is arranged below the wheel. The concave bottom to the tooth spaces strengthens the teeth at the roots. From four to six thousandths of an inch are allowed for grinding on both the tooth flanks and the bottom of the tooth spaces.

During the grinding process the abrasive wheel is at intervals withdrawn from the work, and the truing diamonds are brought into action by the operator. This operation, made at the will of the operator, is always effected just before the finishing cut is to be made.

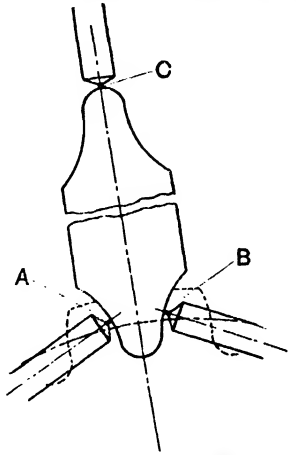


FIG. 8.—Principle of Orcutt Gear Grinder.

The abrasive wheels used are about 12 ins. diameter, and are in effect the media between the diamonds and the work, the former moving as directed by an accurately-shaped form plate. Fig. 8 shows the type of wheel used and the position of the correcting diamonds.

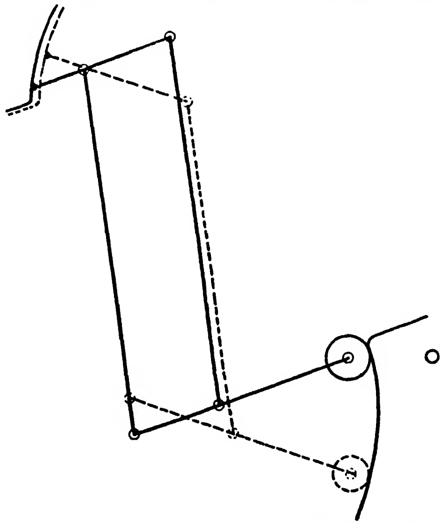


FIG. 9.—Pantagraph Mechanism for Truing Abrasive Wheels in Formed Wheel Gear Grinder.

Wheel-Trimming Mechanism.—The movements of the correcting diamonds are governed by a form plate which is made six times as large as the actual teeth to be ground, and an accuracy of one ten-thousandth of an inch in the tooth profile is secured. The motion of the diamonds is obtained through a pantagraph, in which a roller attached to the long lever works on the profile of the form plate (see fig. 9). The rollers are held in contact with the form plate by springs. To set the mechanism a gauge is used which fits over the diamond spindles. The diamonds are then brought up to bear on a flat surface, which sets them at the correct distance from the centre of rotation. The wear of the trimming diamonds is allowed for by using different diameter rollers in contact with the form plates (see dotted lines on fig. 9).

Another important feature of this machine is to be found in the indexing plate, which provides the means of setting the successive gear teeth in position for grinding. The accuracy of the profile is obviously dependent upon this, and hence a special arrangement has been devised for checking the accuracy of this plate. The checker consists of a suitably arranged silver scale divided into 5 minutes of arc, and further sub-division is effected by a micrometer and microscope system enabling one second of arc to be read. In addition the silver scale is checked by a system of three microscopes arranged 120° apart.

Internal gears can be ground on the same machine as that referred to above.

The Generating Process.

The well-known Maag-Sulzer gear grinder represents a class of machine working on the generating principle, and is extensively used in gear manufacture.

It has been recognised by the designers of this machine that in order to obtain the high degree of uniformity necessary for interchangeability of gears it is necessary to grind both profiles of the teeth at the same setting. In order to achieve this object two grinding wheels are used and are

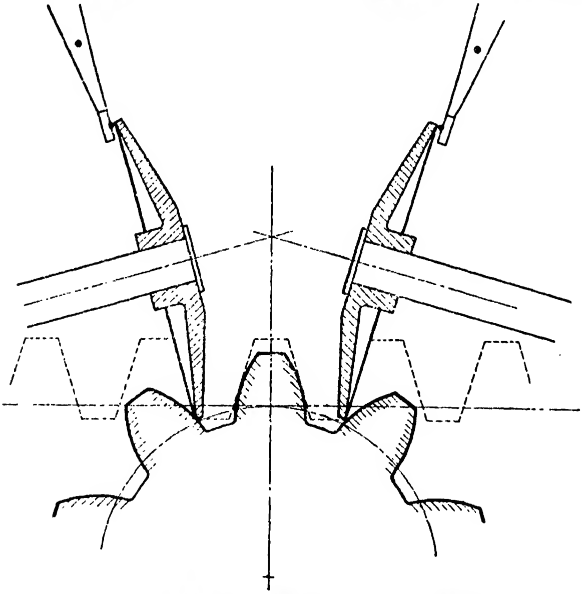


FIG. 10.—Principle of Maag Gear Grinding Process.

mounted at angles equivalent to the sides of the teeth of a rack cutter. In earlier machines the two abrasive wheels operated in the same tooth space at the same time, but in recent designs the wheels have been arranged to work in adjacent tooth spaces.

The abrasive wheels are mounted on separate sliding carriages, which allow independent movement to enable a variety of pitches to be ground. In the earlier type of machine flat abrasive wheels were employed, but these have now been superseded by the saucer-shaped abrasive wheel. shown in fig. 10. Trouble was experienced with the former type of wheel in retaining a flat grinding *surface* due to the wear of the truing diamond. In the case of the latter the cutting *edge* of the abrasive saucer-shaped wheel revolves in one plane. Thus when the cutting edge wears, the plane of rotation is maintained.

The difficulty referred to in regard to flat grinding wheels has been overcome in some cases by using larger diameter abrasive wheels, but when accuracy of one ten-thousandth of an inch is demanded, the liability of the larger wheels to springing is apt to impair this accuracy.

Motion of Gear in Generating Grinder.—The gear-wheel to be ground is mounted on a mandrel, which is given two motions, one performed quickly and the other slowly. The gear is fed slowly longitudinally under the abrasive wheels, and at the same time reciprocated across the machine rapidly. Thus by giving a rolling motion to the gear, which combined with the latter reciprocating motion, gives a motion equivalent to that which the gear would have when in engagement with a rack. The reciprocating motion may be as high as 200 per min., while the longitudinal feed may be one of eight rates, from 1 mm. to 3 mm. per stroke.

Correction of Abrasive Wheels for Wear.—Compensation for wear of the grinding wheels is made by an automatic electrical mechanism. The principles of the original design first used in 1914 are still in vogue, slight modifications only having been made in the constructional details to facilitate manufacture.

Lapping Gear Teeth.

Lapping is an old process. In view of the high degree of finish and accuracy now produced this method is becoming obsolete, and cannot be regarded as of much value in the light of modern precision gear cutting.

The idea of running hardened gears together with lubricant and abrasive material has been tried. For producing a highly finished surface to the tooth faces the method is not satisfactory, but it serves to correct slight errors in tooth profiles. From a production point of view the method is also unsatisfactory.

Good results as regards finish and accuracy can be obtained by running a hard steel gear in mesh with one made of softer material and having a wide face. Both gears should be run at high speed and with load on while abrasive material and lubricant are applied to the tooth faces. The hard gear should be made to traverse the face of the wider soft gear.

In some cases cut gears are run with load on with two or three hardened and ground gears. This process tends to produce a well-burnished tooth surface.

Involute Measuring Machine.

Elimination of tooth form errors is a subject which is being rapidly pursued, and in consequence new devices and modifications to existing ones are almost continuously appearing.

The most recent machine produced is an Orcutt involute tooth profile-measuring machine (see *The Engineer*, April 21, 1933), designed for the examination of involute profiles, those of helical teeth, internal gears, and teeth produced by the Fellows' gear-cutter. The machine also tests the radius of the base circle.

Important Facts to be Considered when Setting Up Gearing.

It is essential when setting up a system of gears to be sure that the correct tooth meshing is not impaired by faulty or unsuitable fitments.

Tooth meshing may be affected by errors such as improper foundation, temperature effects on shaft and casing, defective coupling arrangements between shafts, inaccurate shaft alignment and eccentricity. All these points should receive close attention when designing a system of gears, with a view to avoiding defective meshing of the teeth due to errors or unsuitable equipment in the 'external' arrangements. Gear manufacturers are always prepared to advise in these matters.

Bevel Gearing.

Bevel gears are used to transmit power between two shafts whose axes of rotation are inclined to one another.

A pair of bevel wheels which gear together having equal diameters, and having axes of rotation inclined at 90 degrees to each other are often referred to as *mitre wheels*.

Bevel wheels can be manufactured by the generating process and may have straight or spiral teeth. Either form of teeth are capable of production by the generating process.

Double helical bevels are sometimes specified, and can be obtained from a number of gear manufacturers, but are not in general recommended since the advantages obtained from double helical spur gears are not readily secured from double helical bevel gears. The axial load is not balanced in the latter as is the case in the former, and extreme care is needed in setting up the bearings etc. for satisfactory working.

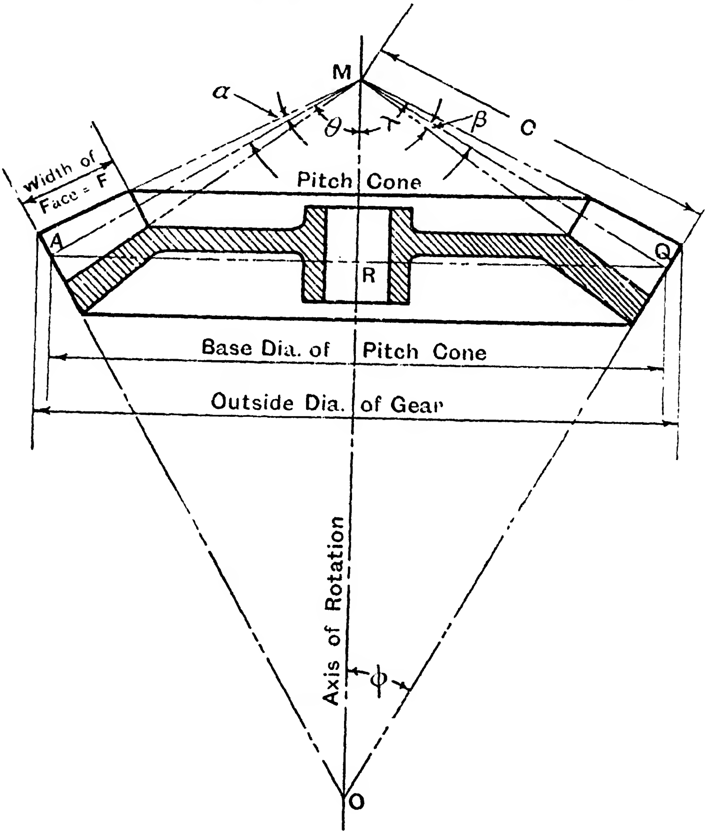


FIG. 11.

NOMENCLATURE.

Pitch cone.—This is an imaginary cone represented by AMQ which, when in contact with a similar cone (the apices meeting in a common point) would transmit by friction the requisite power with the given velocity ratio. It therefore bears a similar relation to the bevel gear that the pitch cylinder bears to the spur gear.

Back cone.—A cone represented by AOQ formed by producing the lines AO and OQ at right angles to the pitch surface, to intersect the axis of rotation of the bevel gear in O.

Pitch cone angle.—Represented by the angle θ, and is the apex angle of the triangle AMR which, when rotated about MR, sweeps out the pitch cone surface. Some authorities call this angle the *semi-pitch cone angle*. The former definition is more usual.

Back cone angle.—Angle ϕ.

Addendum angle.—Angle α.

Dedendum angle.—Angle β.

Face angle.—Angle λ.

Circular and diametral pitch.—These terms are used in connection with bevel gears and have the same meaning as in spur gearing. In the case of a bevel gear, however, the pitch of the teeth becomes smaller as the apex of the pitch cone is approached. In all cases the pitch is measured on the face of the back cone at the base of the pitch cone. Thus the circular pitch is the circumference of the base of the pitch cone divided by the number of teeth in the gear. Similarly the diametral pitch is the number of teeth per inch of diameter of pitch cone base.

Equivalent number of teeth or Virtual number of teeth.—These terms are synonymous. If the surface of the back cone is developed, and teeth set out on the developed surface, having the *same pitch* as the actual teeth in the bevel gear, then the number of teeth in this *equivalent spur gear* is called the '*equivalent*' or '*virtual*' number of teeth.

If N = number of teeth in bevel gear and N^1 = number of teeth in equivalent spur gear

$$N^1 = \frac{N}{\cos\theta}$$

Proportions of Tooth and Pressure Angles.

The involute profile is now generally used in bevel gears and the tooth proportions are commonly the same as in spur gears, *i.e.*

$$\text{addendum} = 1/P \; ; \; \text{dedendum} = 1 \cdot 157/P.$$

Generally the pressure angles used are $14\frac{1}{2}°$ and $20°$ although higher values are employed in special cases.

Face width.—The face width F should not be greater than one-third of the length of the pitch cone surface, since the strength of the teeth diminishes as the apex of the pitch cone is approached. The face width is seldom less than *twice the circular pitch* or greater than *five times* the *circular pitch.*

British Standard Tooth Form for Bevel Gears.*

The standard tooth form recommended is of 20° pressure angle, involute, and of full depth. The proportions of the teeth are similar to those recommended for straight spur gears except that the radius of the clearance space is $0 \cdot 082p$. the centres of the two fillets being $0 \cdot 095p$ apart at the largest end of the tooth. At the small end the surface of the tooth space is almost semi-circular. It is claimed that by making the surface of the tooth space approximately semi-circular the resistance to fatigue of hardened and heat-treated gears is much improved. The static strength is reduced by this form of tooth space due to the greater height of the tooth. The 20° pressure angle tooth also possesses superior strength and wear-resisting qualities and interference in the smaller wheels is much reduced.

Helical Bevels.

These are used for drives in machine tools, and are suitable for general mechanical work, and combine quiet running with increased pitch line velocity. High pitch circle velocity is made possible with this type of tooth.

Two forms of helical teeth are commonly made, one the 'Gleason' type, or curved tooth helix, and the other the common single helical; both are capable of production by the generating process. The former is generated by a cutter moving in a circular orbit, the tooth conforming very closely to a circular arc. The latter is generated by a tool constrained to move in a straight line across the tooth face and offset from the apex of the pitch cone. The curved tooth helical bevel is used in automobile drives.

Helical angles.—Suitable spiral angles for the curved tooth bevel range between 30° and 35° with a $14\frac{1}{2}°$ pressure angle. In the plain single helical type, helical angles vary up to 20°. (For definitions of helical angle, see notes on helical gearing.)

Tooth proportions of helical bevels.—These vary somewhat. Typical values are as follows :

$$\text{addendum } 0 \cdot 85/P \; ; \; \text{dedendum } 1 \cdot 05/P.$$

* See B.S.I. Specification, No. 545.

Interference in Bevel Gears.

Bevel gears suffer from this defect in the same way as spur gears, when the number of teeth is too small and the pressure angle unsuitable. Interference occurs when the 'equivalent number of teeth' is below 32 in conjunction with a pressure angle of 14½°. By increasing the pressure angle to 20° the minimum number of teeth possible in the equivalent gear is decreased to 17 without interference.

Correction for interference.—As in the case of spur gears, addendum correction is commonly applied to remedy the defect of interference. For purposes of design, charts are used from which suitable values of the correcting constants may be selected. See B.S.I. Specification, No. 545.

Strength of Bevel Gears.

Since the pitch of the teeth diminishes as the apex of the pitch cone is approached, hence the size of the tooth becomes smaller, so the strength of the teeth diminishes.

The method of calculating the strength of bevel gear teeth is similar to that used for spur gearing except that a correcting factor is required to allow for the varying strength of the tooth across the face.

Thus the allowable load

$$W = \frac{S.F.Y.(O-F)}{P.O.}$$

S = safe working stress in lb. per sq. in. Y = tooth strength factor, and P = the diametra pitch. The factor $\frac{(O-F)}{O}$ approximately expresses the ratio of the strength of a bevel gear tooth to that of a spur gear tooth of similar size. The dimensions O and F are indicated on fig. 11.

The *tooth strength factor* Y is based on the *number of teeth in the equivalent spur gear.*

Safe bending and surface stresses.—These are determined as in the case of spur gearing.

Zone factor.—This value for bevel gears is the same as for a spur gear having the same number of teeth as the virtual or equivalent number of teeth in the bevel gear.

VALUES OF 'Y' FOR BEVEL GEARS.

N′.	Y. 14½° Involute Teeth.	Y. 20° Involute Teeth.	N′.	Y. 14½° Involute Teeth.	Y. 20° Involute Teeth.
12	0·210	0·245	27	0·314	0·349
13	0·220	0·261	30	0·320	0·358
14	0·226	0·276	34	0·327	0·371
15	0·236	0·289	38	0·336	0·383
16	0·242	0·295	43	0·346	0·396
17	0·251	0·302	50	0·352	0·408
18	0·261	0·308	60	0·358	0·421
19	0·273	0·314	75	0·364	0·434
20	0·283	0·320	100	0·371	0·446
21	0·289	0·327	150	0·377	0·459
23	0·295	0·333	300	0·383	0·471
25	0·305	0·339	Rack	0·390	0·484

The horse-power which can be safely transmitted by a pair of bevel gears may be calculated in the same way as that for spur gears when the tooth load W has been determined from the foregoing formula.

HELICAL GEARING.

Helical gears are used to transmit motion between two non-intersecting shafts, the axes of which may be inclined to one another at angles varying between 0° and 90°. When the shafts are at 90° to one another the drive is similar to that of a worm and worm wheel. Although the term 'spiral' is often used to indicate this type of gearing, actually each tooth is part of a helix, and the correct description of such gearing is helical rather than spiral.

Helical gears may be single, double or triple. The single helical gear is open to the objection that an axial thrust is introduced which requires special arrangements for taking up this end load. By using double helical teeth a form of gear is obtained which surpasses in efficiency, load carrying capacity, wear, and quietness of operation anything yet produced.

Helical gears, either single or double, may be produced in any one of three ways: (*a*) by generating process in which a rack cutter is employed, (*b*) by a hobbing process, (*c*) by use of an end mill.

Interference in Helical Gears.

As in spur and bevel gears interference occurs in helical gearing and is most serious when a $14\frac{1}{2}°$ pressure angle is employed.

To obviate the difficulty addendum correction is applied in a similar way to that in spur gears. The correction is, however, based on the *virtual* numbers of teeth and not on the actual numbers. (For definitions of virtual number of teeth and virtual diameter, see Nomenclature.)

NOMENCLATURE.

Normal circular pitch is the distance between the centres of two consecutive teeth measured round the pitch cylinder and normal to the tooth profile.

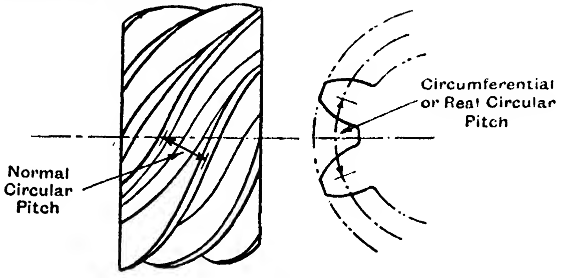


FIG. 12.

Circumferential or real circular pitch is the distance between the centres of two consecutive teeth measured round the pitch cylinder and in a plane perpendicular to the axis of rotation. See fig. 12.

Helix angle θ.—The angle which the tooth helix makes with the axis of rotation. Fig. 13.

Lead Angle λ.—The complement of the helix angle, or the angle between the tooth helix and a plane perpendicular to the axis of rotation. Fig. 13.

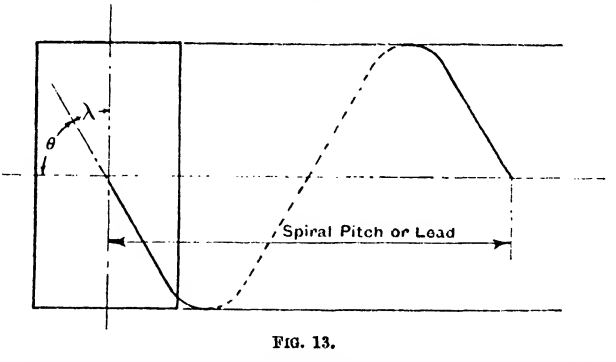


FIG. 13.

Helical pitch or lead.—A helical gear can be considered as a portion of a multiple-threaded screw If the gear was made wide enough each tooth would make a complete turn on the blank. The angle of the teeth controls the pitch of this complete turn, known as the *helical pitch or lead*. In order to cut the teeth at any desired angle the lead is required. See fig. 13.

$$\text{Lead} = \frac{\text{Pitch circumference}}{\text{Tan } \theta}, \text{ where } \theta = \text{helix angle.}$$

Increase of helix angle increases the circular pitch, and hence the pitch diameter for the same number of teeth.

Normal pressure angle is the angle between a tangent to the tooth profile at the pitch cylinder and a radial, the angle being measured normal to the helix. Definitions given previously for pitch cylinder, pitch diameter, etc., are similarly applied to helical gears.

Equivalent or virtual diameter of a helical gear.—The terms are synonymous. The virtual diameter is that of a circle having a radius equal to that of the pitch cylinder on a section normal to the profile of the teeth.

Equivalent or virtual number of teeth.—This is the number of teeth in a gear of a diameter equal to the *virtual* diameter of the helical gear, the pitch being equal to the normal pitch of the latter.

Base diameter.—The diameter of the circle from which the involute curves for the tooth profiles are generated.

Centre distance.—The distance between the axes of rotation of two engaging gears, measured perpendicularly.

Other definitions applicable will be found under spur gearing.

Let d_v = virtual diameter of helical wheel, d = actual diameter of pitch cylinder, and θ = spiral angle,

$$\text{then } d_v = \frac{d}{\cos^2\theta}$$

Also, if N_v = virtual number of teeth in helical gear, and N = actual number of teeth in helical gear,

$$N_v = \frac{N}{\cos^2\theta}$$

Relation between circular and diametral pitch as applied to helical gearing.—Actually it is more convenient to work on the diametral pitch as this obviates the introduction of π. In modern practice either the diametral pitch or the module is always used.

θ = helix angle; *r.c.p.* = real circular pitch; *n.c.p.* = normal circular pitch;

N = no. of teeth; *r.d.p.* = real diametral pitch; *n.d.p.* = normal diametral pitch.

Normal circular pitch = real circular pitch × cos θ.

$$n.c.p. = \frac{\pi}{n.d.p.}; \quad r.c.p. = \frac{n.c.p.}{\cos\theta}$$

$$\text{Pitch dia.} = \frac{r.c.p. \times N}{\pi} = \frac{N}{r.d.p.}; \text{ Outside dia.} = \text{Pitch dia.} + 2\,(\cdot 3183\ n.c.p.$$

$$r.d.p. = n.d.p. \times \cos\theta.$$

A common example of single helical drive is that in which two helices are used to transmit motion between two shafts at right angles to each other.

Consider two gears A and B meshing under the conditions stated above.

Let

D = pitch dia. of A;	d = pitch dia. of B;
N = no. of revs. of A;	n = no. of revs. of B;
α = helical angle of A;	β = helical angle of B.

Velocity ratio = $\frac{N}{n}$. The number of revolutions of the wheels is inversely proportional to the number of teeth.

Number of teeth in A = $r.d.p._A \times D$; Number of teeth in B = $r.d.p._B \times d$.

Also,

Number of teeth in A = $n.d.p._A \times \cos\alpha \times D$; Number o teeth in B = $n.d.p._B \times \cos\beta \times d$.

Since the wheels are in mesh the normal pitches must be the same, hence $n.d.p._A = n.d.p._B$.

Therefore,

$$\frac{\text{No. of teeth in A}}{\text{No. of teeth in B}} = \frac{D\cos\alpha}{d\cos\beta}$$

$$\frac{N}{n} = \frac{d\cos\beta}{D\cos\alpha} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

since $\alpha = 90-\beta$, and $\cos\beta = \sin\alpha$, therefore

$$\frac{N}{n} = \frac{d}{D}\tan\alpha \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The above relations are true for right- or left-handed helical gears, and hold good for any two wheels in mesh, whether axes are at right angles or not. The only difference will be in the relation between α and β.

From equation (2) the required helix angles can be obtained for a given velocity ratio and given diameters.

EXAMPLE.—The camshaft for a 4-stroke cycle gas-engine is to be driven by a single right-angled helical drive. Determine the helix angles of the two gears.

The gear diameters are usually equal in cases of this sort. The speed reduction is 2 to 1, the camshaft running at half engine speed.

$$\frac{N}{n} = 2 = \frac{d}{D} \tan \alpha. \quad \frac{d}{D} = 1,$$

therefore,

$$\tan \alpha = 2.$$

Hence,

$$\alpha = 63^\circ\ 26', \text{ and } \beta = 26^\circ\ 34'.$$

When the helix angles have been calculated, the actual diameters must be obtained, so that with a standard normal pitch an integral number of teeth is given.

In the case of single helical drives the provision of some means to take up end-thrust is essential. Ball thrust bearings are commonly used, while for light duty a plain collar on the shaft, bearing against a bronze bush or end-pad is satisfactory.

Let

H.P. = horse-power transmitted by a pair of gears; V = pitch circle velocity in ft. per min.

P = tangential load on the teeth; $\text{H.P.} = \frac{PV}{33{,}000}$; $P = \frac{33{,}000\,\text{H.P.}}{V}$

If P_a = axial thrust, then $P_a = P \tan \theta$.

Tooth face width.—This should not be less than about 2½ to 3 times the normal circular pitch, to ensure proper continuity of action. The normal tooth profile is involute.

The cutter for milling spiral gears is not selected according to the number of teeth as is the case with spur gears. The number of teeth in the helical gear is divided by the cosine of the helix angle cubed, *i.e.* $\frac{N}{\cos^3 \theta}$. When this calculation has been made, the resulting number is the number of teeth for which the cutter is selected.

Addendum $= \frac{1}{n.d.p.}$; Whole depth of tooth $= \frac{2 \cdot 157}{n.d.p.}$; Normal tooth thickness at pitch line $= \frac{1 \cdot 571}{n.d.p.}$

Double Helical Gears.

When accurately designed and well made this type of gearing has many advantages compared with other forms of gearing. The chief advantages are :—

(*a*) The engagement of the teeth is continuous when proper face width is used, and hence greater loads can be transmitted with high degree of efficiency.

(*b*) Shock is eliminated as the teeth are in continuous engagement, hence the load is transferred from tooth to tooth smoothly. Wear is thereby reduced, and also noise.

(*c*) Vibration is practically eliminated.

There are two forms of double helical teeth, one known as the continuous tooth type tooth, and the other as the staggered tooth type. In the latter, one half of the wheel is moved round relative to the other half a distance equal to a tooth thickness, thus the teeth on the one half are opposite the tooth spaces on the other half. This construction was mainly to facilitate the cutting of the teeth. With this latter form of tooth the permissible load is reduced owing to the discontinuity of the teeth. Methods of production have now advanced to an extent which renders the staggered type of gear teeth almost obsolete.

Tooth profiles and proportions.—The standard normal section of double helical teeth is involute. Proportions vary somewhat according to different manufacturers. The following are typical values:

addendum 0·88/P; dedendum 1·005/P; bottom clearance 0·125/P.

The above are the standards adopted by David Brown & Sons (Hudd) Ltd.

Other manufacturers use such values as: addendum 0·7/P; dedendum 0·85/P.

Face Width.

The total face width should not be less than 5 times the normal circular pitch. High ratio gears have face widths up to 12 times the normal circular pitch.

Pressure angles.—Usually a 20° pressure angle is adopted although occasionally $14\frac{1}{2}°$ is used. To meet special requirements pressure angles above 20° are used.

Helix angles.—These vary according to the work for which the gear is designed. Angles of $22\frac{1}{2}°$ are common for general purposes. Fine pitched turbine gears often have 30° helix angles. For rolling mill gears 32° 9′ is adopted by David Brown & Sons, Ltd., as this angle is convenient in setting up the gear cutting machine (tan of this angle $= \frac{\pi}{5}$). Gears for heavy duty, such as for haulage apparatus, including triple helical gears have helix angles as high as 45°.

Back-lash.—To secure efficient working of double helical gears, a certain amount of back-lash is essential. In general work David Brown & Sons, Ltd., recommend 0·005 in. to 0·01 in. per in. of circular pitch, the teeth of each gear in mesh being reduced in thickness by half this amount.
Gears working in high temperatures may require greater back-lash, and for satisfactory working, fine pitched gears, lubricated by spraying nozzles, also require increased back lash.

Interference.—This is corrected in a similar way to that in spur gears, a chart being used for the purpose. (See B.S.I. Specification, No. 436.) David Brown & Sons, Ltd., recommend the following addenda corrections.
Diametral pitch gears correction applied in increments of 0·04 in. for a 1 P. tooth giving 22 divisions (addendum $= \frac{0\cdot 88}{\text{D.P.}}$).

Circular pitch gears 0·01 in. for a tooth 1 in. pitch, giving 28 divisions (addendum $= 0\cdot 28p$).

Strength of Double Helical Gears.

Surface stress.—It is important to keep this within certain limits, depending upon the materials of the two gears in mesh. This stress depends upon such factors as the modulus of rigidity of the gear material, intensity of applied load, and radius of curvature of teeth. Suitable values of surface stress are found by multiplying basic values by a speed coefficient determined from charts.* The following table of basic values of surfaces stress is reproduced from David Brown's technical data.

Material.	Basic Value.	When in contact with
Cast Iron	150	Cast Iron
,, ,,	100	Steel
Phosphor bronze, sand cast	125	Steel
,, ,, centrifugally cast	385	Steel
Mild Steel 28 tons U.T.S.	450	Cast iron
,, ,, ,, ,,	385	Bronze
,, ,, ,, ,,	260	Steel
0·3 carbon steel 32 tons U.T.S.	550	Cast iron
,, ,, ,, ,, ,,	465	Bronze
,, ,, ,, ,, ,,	310	Steel
0·4 ,, ,, 40 tons ,,	765	Cast iron
,, ,, ,, ,, ,,	660	Bronze
,, ,, ,, ,, ,,	440	Steel
0·5 ,, ,, 50 tons ,,	865	Bronze
,, ,, ,, ,, ,,	575	Steel
Nickel steel 60 tons U.T.S.	885	Steel
,, chrome steel 100 tons U.T.S.	1,240	Steel
,, steel, case hardened	1,860	Bronze
,, ,, ,, ,,	1,240	Steel
Rawhide	80	Cast iron or steel.

Zone factor.—As in spur gears a zone factor is employed. This depends upon number of teeth in wheel and pinion, the spiral angle, and pressure angle. The product of the zone factor and the allowable surface stress gives the allowable load per in. width of tooth face for teeth of unit pitch.†

* See B.S.I. Specification, No. 436.
† Charts are used for determining zone factors.

Bending stress.—To obtain the stress due to bending on double helical teeth, the tooth is considered to act as a cantilever with a load at the line of contact. The safe allowable stress is found by determining the product of the basic stress and the speed coefficient obtained from charts.* The speed coefficient used in this latter case is the same as that used in calculating the safe surface stress.

BASIC VALUES OF BENDING STRESS ALLOWED IN DOUBLE HELICAL GEARS.

The following table of safe stress is abstracted from David Brown & Sons, Ltd., technical data.

Material.	Basic Bending Stress.
Nickel chrome steel, 100 tons per sq. in. U.T.S.	12,200 lbs. per sq. in.
,, steel, 60 tons per sq. in. U.T.S.	8,900 ,, ,, ,,
0·5 per cent. carbon steel, 50 tons per sq. in. U.T.S.	8,350 ,, ,, ,,
0·4 ,, ,, ,, ,, 40 ,, ,, ,, ,,	7,750 ,, ,, ,,
Nickel steel, case hardened	7,750 ,, ,, ,,
0·3 per cent. carbon steel	7,000 ,, ,, ,,
†Mild steel	6 700 ,, ,, ,,
Phosphor bronze	4,700 ,, ,, ,,
Cast iron	2,000 ,, ,, ,,
Rawhide	1,850 ,, ,, ,,

Tooth strength factor.—This factor, usually denoted by Y, is again obtained from charts; its value depending upon the number of teeth in the wheel and mating pinion. It expresses the tooth load in lbs. per in. of face width, and the bending stress produced by it for teeth of 1/P. Hence, the allowable load per in. width of face in lbs. $= fb\mathrm{Y/P}$.

fb = bending stress at the required speed, Y is the tooth strength factor, and

P. = diametral pitch.

Similarly, considering the allowable working surface stress fc at required speed and the zone factor Z,

allowable load per in. width of face in lbs. $= fc\mathrm{Z/P}$.

Horse-power transmitted.—The safe horse-power that can be transmitted depends on the surface stress and the bending stress that can be allowed on either the wheel or pinion. The weaker of the two obviously limits the power that can be transmitted, and hence the following calculations when applied to each of the units, will give the power which the pair of gears will safely transmit.

Let

fc = safe surface stress in lbs. per in. width of face at required speed.

$_b$ = safe bending stress in lbs. per in. width of face at required speed.

F = face width of wheel or pinion in ins.

T = number of teeth in wheel or pinion.

N = number of revs. per min. of wheel or pinion.

P = diametral pitch.

Y = tooth-strength factor.

Z = zone factor.

π = 3·1416.

D = dia. of wheel or pinion in ins.

Safe horse-power based on surface stress

$$= \frac{f_c.\mathrm{Z.F.}\pi \mathrm{DN}}{33{,}000.\ \mathrm{P.}\ 12} = \frac{f_c.\mathrm{Z.F.}\ 0\cdot262\ \mathrm{D.N}}{33{,}000\ \mathrm{P.}} \qquad \mathrm{D} = \frac{\mathrm{T}}{\mathrm{P.}}$$

$$\therefore \text{ Safe horse-power} = \frac{f_c.\ \mathrm{Z.F.}\ 0\cdot262\ \mathrm{T.N}}{33{,}000.\mathrm{P}^2} = \frac{f_c.\mathrm{Z.F.T.N}}{126{,}000.\mathrm{P}^2}$$

Similarly the safe horse-power based on bending stress $= \dfrac{f_c.\mathrm{Y.F.T.N}}{126{,}000.\mathrm{P}^2}$

The factors in the foregoing are multiplied together, the . being a multiplication sign. The basic values given for the stresses are for gears having 22½° helix angles.

* British Standard Specification No. 436, 1940.
† See page 981

Mounting of Double Helical Gears.

It is of first importance that the axes of rotation of a double helical gear reduction should be *parallel and in the same plane.* Uneven load distribution will result if this is not so. It is important that the centre distance be such that the requisite amount of back-lash is obtained. Back-lash must not be excessive, but must be sufficient to ensure proper lubrication. In special cases, such as in gun-sights, back-lash is undesirable. Excessive back-lash is liable to create hammer-blow.

In order to ensure that there shall be no end thrust, it is necessary for the gears to take up a position of equilibrium, and hence flexible couplings between shafts should be used.

Triple Helical Gears.

A gear having two changes in the hand of the helices is called a triple helical gear. Such gears are sometimes used in mining machinery. The idea of the triple helical gear apparently evolved from the notion that helical gears gave best results when the apex of the teeth pointed in the directions of rotation, and hence the triple gear provided the best form of gear for reversing. Such a contention does not, however, appear to have much to support it in the light of modern gear technology. They are, however, still in some demand, and engineers occasionally specify them.

Turbine Reduction Gears.

The fact that large horse-powers are transmitted at high pitch-line velocities calls for special consideration in manufacture. Pitch-line velocities up to 18,000 ft. per minute are dealt with. Speed reductions from turbine shafts to electric generator shafts, mill shafts, and ships' propeller shafts need specially designed and accurately made gearing. Similar kind of gearing is needed to increase speed from internal combustion engines, electric motors, etc., required to work blowers, compressors, rotary pumps, and the like.

Turbine Gear Arrangements.—Figs. 14 to 20 are intended to show diagrammatically various turbine gear arrangements. In regard to single reduction gear, ratios up to 1 to 20 or 1 to 26 are possible, while reductions up to 1 to 50 are possible with double reduction gearing. Double helical gears are commonly used while the left- and right-hand portions are often separated. Axial thrusts may be balanced by the use of double helical gearing and high efficiencies of transmission often up to 98 per cent. are possible.

In respect to marine installations special precautions are necessary in design since uneven stressing may be set up due to rough weather and other causes. If not properly catered for, these stresses produce pitting in the teeth and excessive wear. The straining actions in the larger wheels require careful treatment to prevent these defects arising.

TURBINE GEAR ARRANGEMENTS.

Fig. 14.—Single casing turbine with single reduction gear.
Fig. 15.—Twin turbine casings, single reduction.
Fig. 16.—Three turbine casings, single reduction.

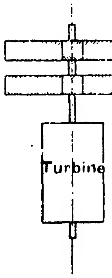


FIG. 14.

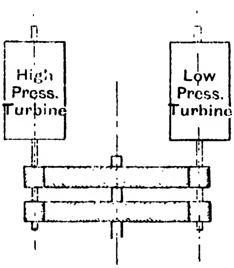


FIG. 15.

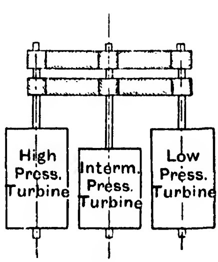


FIG. 16.

To allow the condenser to be placed below turbine the L.P. casings are arranged so that the pinions mesh near the top of the wheel, as in fig. 20.

Fig. 17.—Four turbine casings in groups.

Fig. 18 shows a twin turbine double reduction.

Fig. 19 shows an arrangement for securing a more symmetrical stressing of pinions and reduction in twist than with the previous arrangement. The intermediate gears are arranged in the centre of the divided main gears.

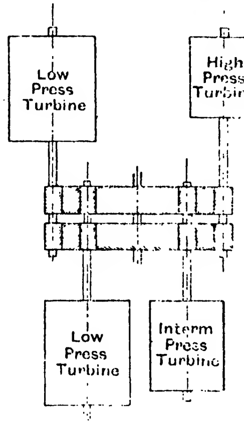


FIG. 17.

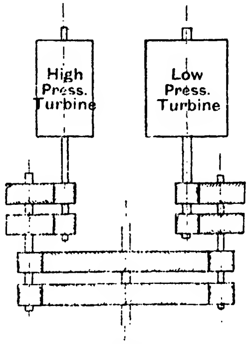


FIG. 18.

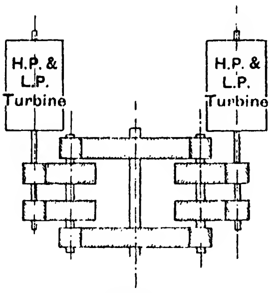


FIG. 19.

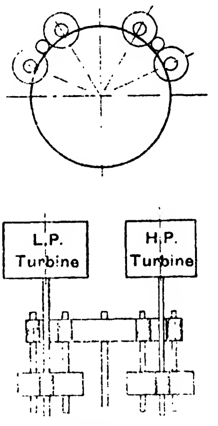


FIG. 20.*

Fig. 20 shows a single helical reduction gear by Brown Boveri et Cie.

* Nozzles 0·10 in. to 0·15 in. diameter are employed, spaced 1½ ins. to 4 ins. apart. Oil pressure is governed so that the rate of flow is about ·33 to ·67 gals. per minute per inch width of tooth when peripheral speed is 33 ft. per second. When peripheral speed reaches 130 ft. per second, 1·0 to 1·32 gals. per minute per inch width are used. Nozzles having too small a bore should not be used, since clogging is liable to occur. It is a further advantage to have the oil reservoir above the level of the gear wheels. Too high a jet velocity should not be used as atomisation of the oil is likely.

Fig. 21 is a diagrammatic arrangement by Parsons' Marine Steam Turbine Co., Ltd., of a double reduction gear for a single screw cargo boat.

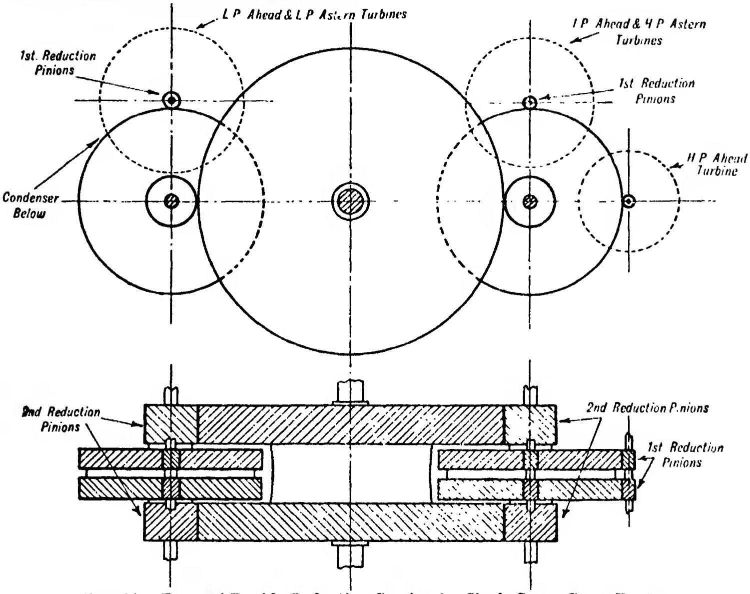


FIG. 21.—Parsons' Double Reduction Gearing for Single Screw Cargo Boat.

***Double Helical Turbine Reduction Gear Pinions.*—These are commonly made of 3 per cent. to 5 per cent. nickel or nickel-chrome oil hardened steel, the pinion being an integral part of the shaft. The left- and right-hand portions of the gear are commonly separated to allow for the provision of a centre bearing, to obviate excessive deflection.**

***Double Helical Turbine Reduction Gear Wheels.*—Small wheels are commonly manufactured from forged steel and forced on to the shaft by hydraulic pressure. Larger wheels are generally made in two parts. The wheel centre made either of cast iron or cast steel has a rolled steel rim shrunk on to it. The turning to size and the cutting of the teeth are carried out when the wheel has been hydraulicly pressed on to the shaft. Perfect balancing of turbine gears is essential in order to reduce vibration to a minimum. The production of this type of gearing calls for specially designed hobbing machinery and precision ground hobs. Heat treatment also forms a most important part of an organisation dealing with this class of work.**

Turbine Gear-Tooth Design.

An important factor in tooth design is the limiting pressure which can be allowed on the teeth. Points to be considered in fixing the pressure are wear, type of lubrication, and peripheral speed.

Gears for cargo steamers limiting pressure per inch width, 350 to 450 lbs., calculated on the length projected on to the shaft. In naval turbines pressures up to 850 lbs. per in. are used.

Double reduction gears tooth pressure is limited on the first pinion on account of high peripheral speed. About 450 lbs. is allowed on the first pinion, and 650 lbs. to 1,450 lbs. per in. width on gears running at 164 ft. per second. Under these latter conditions 10,000 horse-power has been transmitted effectively.

The general treatment of design of turbine reduction gear teeth of the double helical type is given under the notes on helical gearing and may be applied when suitable stress values, consistent with the spiral angles to be adopted, have been decided upon.

The height of the teeth is usually about 0·6 times the circular pitch. The teeth are inclined usually at 30°.

Pitches vary. For small gears, values of $\cdot 12\pi$ to $\cdot 16\pi$ are used, while in the slower running gears, pitches of $\cdot 32\pi$ are used.

One of the factors to be taken into account when deciding upon the tooth pressure is the danger of the oil film failing if this pressure is too high. According to H. M. Martin (*Engineering*, August 11, 1916), the stress on the oil film obeys some such law as $p = bd^n$, where, p = load on tooth per inch run in lbs.; d = pitch diameter of pinions in inches; and b = constant; n is given as ½, and b is taken by Messrs. Parsons as 175, but values of 250 have been used.

In most of the examples given the pinions are located above the gear wheels with which they engage. The pinions are divided, each pinion or wheel forming one-half of the double helical gear.

Pitch.

Fine pitches give silent running at high speeds, and a normal pitch, that is, the pitch measured at right angles to the teeth, of 0·583 in. (or $\frac{7}{12}$ in.) has been largely used in this country. For small pinions, or where extreme silence is of importance, 0·4 in. in pitch has been used. For turbine gears the pitch for a pinion having at least thirty teeth may be found from the expression:

$$p = \frac{WK}{P},$$

where,

p = pitch in inches; P = total tooth load in lbs.; W = width in inches; K = constant corresponding to the velocity. (See fig. 22.)

The expression applies to teeth with normal addendum and dedendum.

The materials in turbine gearing are never highly stressed; gears are designed primarily for wear, and this gives a much greater factor of safety than would be employed were they to be designed for considerations of strength alone.

Thickness of tooth at pitch line is usually about ½ circular pitch. Tooth clearance is allowed to compensate for heating of gears and to ensure proper lubrication. The centre distance is usually increased by this amount to provide clearance.

In order to secure best results the highest degree of accuracy is needed in the manufacture of turbine gears, together with accurate lining up and careful attention while at work.

The teeth are usually cut by generating process or special relief ground hobs. The end-mill process is an older method, but is still used where hobbing process is inapplicable.

For turbine gears a 20° pressure angle is used, the normal tooth profile being involute.

The actual length of tooth carrying the load depends upon the deformation, due to bending and twisting. The amount allowed is limited to about 0·0008 in. or 0·0012 in.

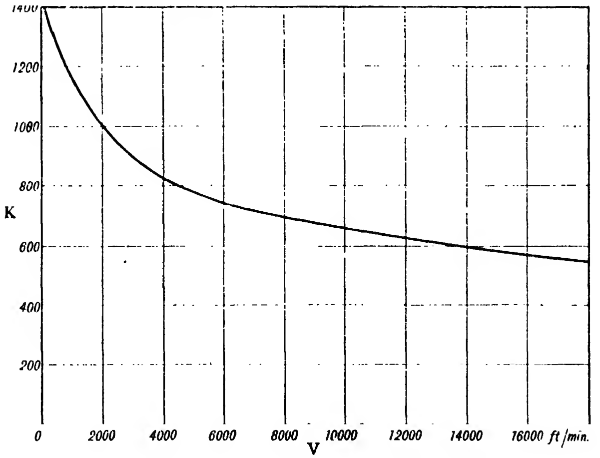


Fig. 22.

No. of Teeth in Pinions.

The number of teeth is not often less than 30 or 35. For cargo steamers and the first reduction gear, pitch-line velocities of 80 ft. per second are employed. For second reduction gear, 28 to 40 ft. per second are used, while naval gears run at 150 ft. per second. Greater pitch-line velocities are sometimes used.

$$\text{Tooth length} = \frac{\text{tangential load on teeth}}{\text{bearing pressure per inch width}}.$$

The Power Plant Co., Ltd., of West Drayton, use for the pinions, alloyed steels (nickel or chrome-nickel steel), and for the wheels above velocities of 3,000 ft. per min., forged steel rims.

The calculation must start from the requisite pinion diameter to transmit a given load without exceeding the very narrow limits within which bending and twisting are allowed.

With gears where the pinion face exceeds three times the pitch diameter, three bearings are invariably required for the pinion, to resist bending. The slighest inaccuracies in pitching and setting of the wheels, and comparatively small bending and twisting strains are liable to affect turbine gears very adversely, but these effects can be counteracted by special designs, such as the wheels with the pressure compensating device as manufactured by the above company.

An ample supply of oil is imperative, not only to lubricate the teeth, but also to carry off the heat generated. For this purpose it is usual to have $\frac{1}{4}$ in. to $\frac{3}{16}$ in. nozzles of about 5 ins. pitch squirting oil under a pressure of 10 to 20 lbs. per sq. in. on to the pinion, some distance in advance of the engagement, as this allows some of the oil to disperse and it is not carried forward and pumped out by the gears themselves. The quantity of oil is about 1 gal. per minute per 100 to 150 h.p. transmitted. The case must be well drained, otherwise if the wheel dips into the oil at the bottom much heat will be generated by the churning of the oil.

Turbine pinion distortion.—A discourse on the subject will be found in the Transactions of the Manchester Association of Engineers, 1919, by Prof. Gerald Stoney, F.R.S., D.Sc.

Gauging of Gear Teeth.

Straight spur teeth.—Generally gear teeth are measured by taking two measurements at the same time. This process is effected by using a special vernier caliper provided with two sets of scales, thus enabling the two readings to be made at the same setting of the instrument. See fig. 23.

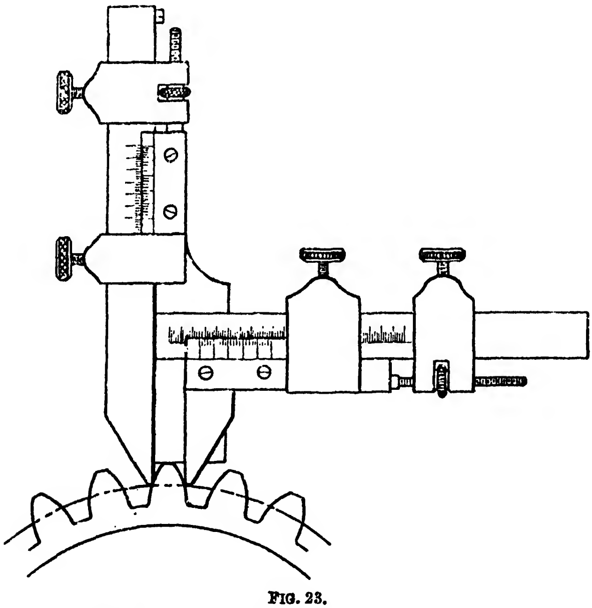


FIG. 23.

The instrument measures the chordal thickness of the tooth and at the same time the height of the tooth above the chord. If the length of the arc of the pitch circle is required between the two profiles of a tooth then a correction to the caliper measurement has to be made. The same applies to the addendum measurement; the nominal addendum must be increased by the height of the arc, to make it equivalent to the caliper reading.

For straight spur gears if D = the pitch dia. N = number of teeth.

then $$\frac{2.\text{chord}}{D} = \frac{\sin 180}{N} \text{ from which the chordal thickness}$$

$$= D.\sin\left(\frac{90}{N}\right).$$

Similarly, the height of the arc $= \frac{D}{2}\left[1 - \left(\cos\frac{90}{N}\right)\right].$

For calipering helical teeth the instrument must be set normal to the tooth profile. The actual caliper readings are then the same as for the equivalent or virtual spur gear having a diameter equal to the equivalent or virtual diameter. Thus, if D^1 = equivalent or virtual pitch dia. and N^1 = equivalent number of teeth, α = spiral angle,

$$\text{Chordal thickness} = D \sin\left(\frac{90}{N^1}\right)$$

$$= \frac{D}{\cos^2\alpha} . \sin\left(\frac{90 \cos^3\alpha}{N}\right)$$

$$\text{Height of arc} = \frac{D^1}{2}\left[1 - \left(\cos\frac{90}{N^1}\right)\right]$$

$$= \frac{D}{\cos^2\alpha}\left[1 - \cos\left(\frac{90 \cos^3\alpha}{N}\right)\right].$$

To facilitate the use of these formulæ tables are often used from which chordal distances can be read off quickly.

WORM GEARING.

There are two types of worm gear—namely, one in which the worm is parallel throughout its whole length and the other in which the worm is curved along its face to suit the curvature of the worm wheel. The former is known as the 'parallel' worm gear, and the latter as the 'Hindley' or 'globoidal' worm gear.

The introduction of machine-cut worm gear of high precision and the increased knowledge of the nature of tooth contacts has resulted in the use of worm gear for the transmission of large horse-powers, with very high efficiencies.

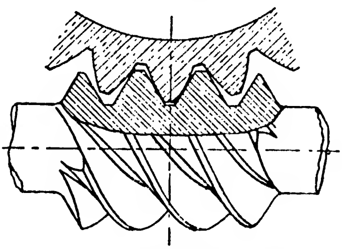


FIG. 24.—Parallel Worm Gear.

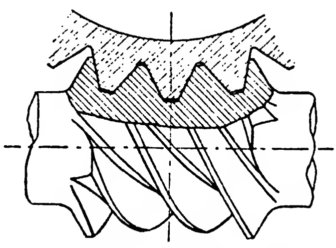


FIG. 25.—Hindley Worm Gear.

Formerly it was considered correct to make the threads of all worm gear straight-sided on a linear section through the axis of the worm, and to take the pitch line midway down the working depth of the threads; thus, for the parallel gear this section showed the threads to be a pure rack, and for the globoidal gear a similar rack formation but with the threads pointing radially to the centre of the worm wheels. Actually these two types are very closely related because a parallel worm can be regarded as being a globoidal worm which has been made to suit a worm wheel of infinite diameter.

The construction of the threads of the parallel worm to correspond to that of a rack gives involute profiles to the worm-wheel teeth on the central section, as shown in fig. 24, and with the Hindley gear, shown in fig. 25, a shape corresponding to the threads of the worm, consequently giving teeth in simultaneous contact. It would therefore appear as if these two types were quite correct in every way: involute tooth engagement for the parallel worm and a complete tooth engagement for the Hindley worm. Actually these conditions only apply on the central plane. Neither is the involute tooth profile nor the attractiveness of the Hindley type maintained; their value is entirely lost on all other planes owing to the curvature of the face of the worm wheel. Whenever straight-sided threads are used for worms, whether for the parallel or the Hindley type, the teeth on other planes than the central one are not, and cannot be, wholly conjugate to the threads of the worms, because the hob during cutting unavoidably removes too much metal from the wheel teeth. The result is that in actual practice neither of these worms transmits uniform velocities to the worm wheels, and instead of the contact being mainly on the leaving side it is almost wholly on the entering side, thus giving a higher coefficient of friction and a lowering of the overhaul efficiency of the gears. Worm gear, unless correctly designed, does not transmit a uniform velocity from one member to another. A close approximation is obtained with gears of comparatively fine pitch, and in those of a high lead angle any error is sometimes concealed by the adjacent thread being in action in such a manner that by sustaining an extra proportion of the load distribution the variable velocity period is tided over. Even the inertia of the parts may prevent a change in the velocity of the engaging pair, but this is detrimental to the gears. To consider a worm gear as a kind of a development from the screw is fundamentally wrong. The pitch surfaces of the worm gear are cylinders lying at right angles to each other, hence they only touch at one point, whereas in a screw and nut the pitch surfaces

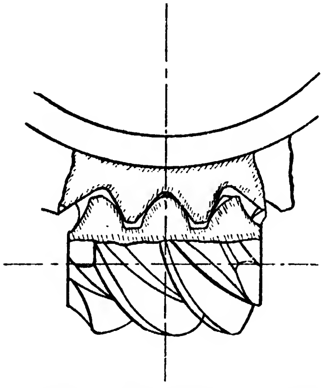


FIG. 26.—'D.B.S. Patent Worm Gear.'

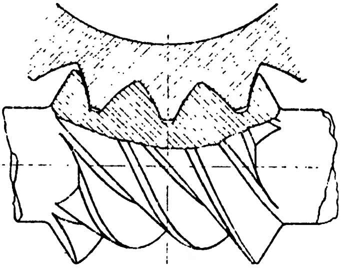


FIG. 27.—Enveloping Worm Gear.

coincide. Expressed in simple language, one can say that, consistent with the transmission of uniform velocity, the normal to the surfaces from the point of contact must always cut the central plane which contains the axis of the wheel, at a point on the straight line lying on the wheel pitch cylinder, and consequently tangential to the pitch cylinder of the worm. Thus, for every point of intersection between a worm thread and the zone of contact, there is a definite relation between its angular and longitudinal positions. If contact takes place at any other point than this intersection, it does so to the detriment of the transmission of uniform velocity.

The mathematical investigations carried out by Mr. F. J. Bostock have shown that the usual rack formation of the threads of the parallel and the globoidal worms is entirely wrong. Fig. 26 shows the correct form of worm threads for the parallel worm, and is known as the 'D.B.S. Patent Worm Gear,' manufactured by David Brown & Sons (Huddersfield), Ltd. In this gear, instead of the threads being straight-sided on a section containing the axis of the worm, they are straight-sided on a section taken at a predetermined distance away from the axis, and the pitch line, instead of being half-way down the working depth, is taken at the bottom of the working depth or at the throat diameter of the worm wheel. This gear has been largely used for the propulsion of buses and lorries in this country and America, and upon test by the N.P.L. gave an efficiency of 97·3 per cent.

The corresponding correct form of threads for the globoidal type is shown in fig. 25, which was also the result of Mr. Bostock's investigations, and is manufactured by Bostock & Bramley, Leeds.

Although the ordinary involute type of parallel worm gear has certain inherent defects, it is still largely used for drives in which maximum load-carrying capacity and efficiency are of little importance.

Nomenclature.

Gear Ratio :—

$\dfrac{\text{No. of teeth in wheel}}{\text{No. of starts on worm}}$. If worm is single threaded, no. of starts is 1 ; if double threaded no. of starts 2, and so on.

Worm-pitch cylinder and worm-wheel pitch circle, fig. 28.—The worm-pitch cylinder is a virtual cylinder which makes contact with the worm-wheel pitch circle at a point lying on a line through the axis of the worm wheel and perpendicular to the worm axis.

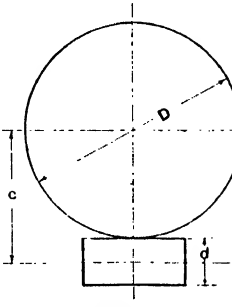


FIG. 28.

Centre distance is the distance between the axis of the worm and the centre of the worm wheel.

Axial pitch of worm threads.—The distance between the centre lines of two adjacent threads taken on an axial section.

Normal circular pitch for worm threads is the distance between the centres of adjacent threads measured round the pitch cylinder and normal to the helix.

Worm lead angle is the slope of the helix, and is the angle between the helix and a plane perpendicular to the worm axis, fig. 29 ; lead angle = α.

Worm helix angle is the complement of the lead angle = θ, fig. 29.

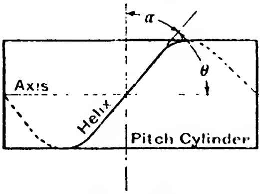


FIG. 29.

Axial pressure angle is the angle of inclination of worm profile at any point on an axial section to a perpendicular to the worm axis = ϕ, fig. 30.

Normal pressure angle is the angle between a tangent plane at any point on tooth profile, and a perpendicular to the axis at that point.

Lead of worm is the linear distance of advancement parallel to the axis of each thread of the worm per revolution of the worm.

Circular pitch of wheel teeth is the distance between the centres of adjacent wheel teeth measured round the pitch circle.

Clearance at worm wheel tooth base as for spur gears.

Worm thread addendum.—Height of thread above pitch cylinder.

Worm thread dedendum.—Depth of thread below pitch cylinder

Worm crest diameter.—The overall dia. of the worm threads.

Worm root diameter.—The dia. of the worm at the root of the threads.

Wheel throat diameter.—The dia. of wheel rim at the bottom of the curved tooth face.

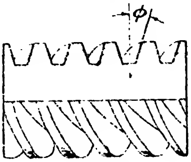


FIG. 30.

Wheel tooth throat addendum.—The distance between the pitch circle and top of teeth on a centre section normal to the wheel axis.

Wheel tooth overall addendum.—The addendum measured to overall dia. of rim.

Wheel teeth dedendum.—The distance between the pitch circle and the bottom of the tooth spaces measured radially and on a central plane.

Wheel overall diameter.—The greatest dia. of wheel rim.

Wheel root diameter.—The dia. at the base of wheel teeth.

DESIGN.

In the common type of worm gear, in which the straight-sided thread on the axial section is used, the $14\frac{1}{2}°$ axial pressure angle is generally adopted and 15° by some makers. The cross-sectional dimensions of the worm thread are commonly made the same as the $14\frac{1}{2}°$ rack teeth—*i.e.* addendum $= 1/P.$; dedendum $= 1/P. +$ clearance; clearance $= \cdot 155$ module. If the lead angle used is greater than 20°, then the proportions are based on the normal pitch. If a 20° lead angle is adopted, then $14\frac{1}{2}°$ pressure angle is too small to give satisfactory results. For automobile drives a 30° pressure angle is usual.

If the worm is to be cut on a shaft direct, then the pitch dia. must be such as to give sufficient strength to make the shaft rigid, and strong enough to carry the torque with the requisite margin of safety. The root dia. must be such as to prevent serious deflection when carrying the load.

The width of face of the worm wheel should be as wide as is reasonably possible, compatible with general design of gear.

The throat should not envelop root of worm by more than about 1·5 times the module, otherwise the wheel teeth will be cut to a sharp edge.

CALCULATIONS.

To compute gear ratio, revs. of worm and wheel being given :—

$$R = \text{revs. per min. of worm}; \quad r = \text{revs. per min. of wheel.}$$

$$\text{Gear ratio} = \frac{R}{r}.$$

To compute no. of wheel teeth N, when gear ratio is given and no. of worm starts, n,

$$N = \text{Gear ratio} \times n.$$

To compute worm pitch dia., given centre distance and worm lead angle, a,

$$d = \frac{2 \text{ centre distance}}{1 + [\text{gear ratio} \times \tan a]}, \text{ approx.}$$

If centre distance and wheel pitch circle dia. is given,

$$d = 2 \text{ centre distance} - p.c.d. \text{ of wheel.}$$

To compute the circular pitch of wheel teeth (p), given $p.c.d.$ of wheel D and no. of teeth N

$$= \frac{\pi D}{N}.$$

To compute the pitch dia. of wheel when circular pitch and no. of teeth in wheel are given,

$$D = \frac{pN}{\pi}.$$

To compute the centre distance when the pitch dia. of worm is given and the pitch circle dia. of wheel,

$$C = \frac{D + d}{2}.$$

To compute the lead of a worm, given the pitch and number of starts of worm (n),

$$\text{Lead} = \text{pitch} \times n.$$

To compute lead of worm, given the pitch dia. of wheel and the gear ratio,

$$\text{Lead} = \frac{\pi D}{\text{Gear ratio}} = \frac{\pi D n}{N}.$$

To compute the worm lead angle, given the pitch dia. of worm and worm lead,

$$\tan \alpha = \frac{\text{Lead}}{\pi d}.$$

To compute normal circular pitch, given the pitch and lead angle α,

$$n.c.p. = p \cos \alpha.$$

To compute worm thread addendum and dedendum on wheel throat addendum, given the pitch of the wheel teeth,

$$\text{Add.} = \cdot 3183\,p\,; \quad \text{Ded.} = \cdot 3683\,p.$$

If a exceeds 20°, Add. = $\cdot 3183\ n.c.p.$; Ded. = $\cdot 3683\ n.c.p.$

To compute worm crest dia., given worm pitch dia. and addendum,

Worm crest dia. = $d + 2$ addendum ; Worm root dia. = $d - 2$ dedendum.

To compute wheel throat dia., given pitch dia. of wheel and wheel tooth throat addendum,

Wheel throat dia. = D + 2 wheel tooth throat addendum.

To compute overall dia. of wheel, given pitch circle dia. of wheel and wheel tooth throat addendum,

Overall dia. of wheel = D + 3 wheel tooth throat addendum.

To compute wheel root dia., given pitch dia. of wheel and wheel tooth dedendum.

Wheel root dia. = D − 2 wheel tooth dedendum.

Worm-wheel peripheral velocity, $Vw = \cdot 262\ DN'$ ft. per min.

Worm peripheral velocity, $vw = \cdot 262\ dn'$ ft. per min.

where N' = no. of revs. per min. of worm wheel, and n' = no. of revs. per min. of worm.

Rubbing velocity at pitch point,

$$V_R = \sqrt{Vw^2 + vw^2} = \frac{vw}{\cos \alpha}.$$

Let,

T = torque on shaft transmitting H.P. horse-power at N revs. per min.,

Then,

$$T = \frac{63{,}025\ \text{H.P.}}{N}\ \text{lbs. ins.}$$

Also,

$$T = \frac{\pi d^3}{16} fs,$$

where d = shaft dia. in ins.; fs = safe shear stress for the material in lbs. per sq. inch.

Neglecting bending and other stresses,

$$= \sqrt[3]{\frac{16\,T}{\pi fs}}; \quad \text{H.P.} = \frac{2.\pi.N.T.}{33{,}000 \times 12}.$$

Interference in Worm Gears.

When it is necessary to have a small number of teeth in a worm wheel interference may occur as in other types of gearing. The difficulty may be overcome by increasing the centre distance. This is equivalent to increasing the pitch diameter of the worm and reducing its addendum and is therefore a similar correction to that applied to spur gears. For the proportions given the correction for the centre distance is as follows :—

$$\text{Increase in centre distance} = A\left(1 - \frac{30}{N}\right)$$

where A = worm addendum, N = number of teeth in wheel.

Irreversible worm gears.—To secure efficient results an auxiliary friction device is preferable to the use of a small lead angle, in order to make a worm gear irreversible. Theoretically, the lead angle must be equal to the corrected angle of friction ; in this connection the actual value of the coefficient of friction is always a doubtful factor. The efficiency of irreversible worm gears is never above 50 per cent. and hence the power lost in wear and tear is abnormal.

Worm and wheel proportions.—These vary with different makers, each having their own particular systems to suit special conditions. The following proportions are recommended by David Brown & Sons, Ltd., who work on the module principle.

Worm addendum = 1·5 times normal module.
Worm dedendum = 0·6 of worm addendum.
Clearance = 0·1 of worm addendum.
Wheel throat addendum = 0·5 worm addendum.
Wheel overall addendum = 0·8 worm addendum.
Normal module = $M \cos \alpha$

$$M\sqrt{\frac{W^2}{W^2 + n^2}}$$

M = module = circular pitch of wheel teeth/π ; W = pitch dia. of worm ÷ module
n = number of starts on worm.

The width of face should not exceed $2\sqrt{A^2 + Ad}$.

A = worm addendum ; d = pitch dia. of worm.

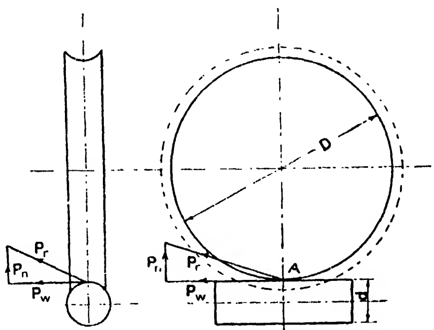


FIG. 31.

It is assumed that the resultant reaction between the worm and wheel passes through the pitch point A.

Resultant reaction = P_r ; resultant force acting parallel to wheel axis = P_W ; resultant force acting parallel to worm axis = P_w ; force acting perpendicular to both axes = P_p ; R = Gear ratio. T = torque due to H.P. in lb. ins.

Approximately,

$$P_w = \frac{2RT}{D} = \frac{2TN}{Dn}; \quad P_W = \frac{2T}{d}; \quad P_p = P_w \tan \phi$$

$$\text{The efficiency is approximately} = \frac{\tan \alpha}{\tan(\alpha + \lambda)},$$

where λ = angle of friction.

The maximum efficiency occurs when the lead angle is about 45°.

Location.

The parallel worm is not seriously affected by slight inaccuracy in the centre distance, and does not need special precaution with respect to endwise movement.

The globoidal form, however, needs accurate location in all directions in order to work correctly. Further, when dismantling the worm and shaft cannot be removed bodily as is the case with the parallel worm.

***Finishing.*—It is essential that worm threads have a very hard case, and must be ground to produce a highly polished surface. Hence the selection of suitable material is of prime importance. Generally worms machined from solid steel forgings give best results. Particular care is also required n the hardening processes in order to avoid, as far as possible, serious distortion.**

EXAMPLE.—Obtain the dimensions of a worm for a gear ratio of 10 : 1 when the centre distance is 8 ins.

Assume number of worm threads = 5 = n, then no. of teeth in wheel = 50 = N. Assuming lead angle = 45°.

$$\text{Let } \frac{\text{pitch dia. of worm}}{\text{module}} = m. \quad \text{Then } \frac{n}{m} = \tan 45° = 1.$$

$$m = n = 5. \qquad \text{Also } N + m = \frac{2c}{\text{module}}.$$

$$\therefore \quad \text{module} = \frac{16}{55} = \cdot 29, \text{ for which a module of } \cdot 3 \text{ would be used.}$$

Pitch dia. of worm = 1½ ins.; dia. of worm wheel = 6½ ins.

Worm Gear Application.

Worm reducing gear units are with great advantage coupled direct to electric motors, enabling reductions in speed from 4 : 1 to 100 : 1 in a silent and efficient manner, and when both of them are mounted on the common baseplate they represent a simple, sound, and economical proposition.

On account of their ability to withstand considerable overload for short periods they lend themselves admirably for intermittent work, because for the work they are called upon to do they can be made relatively small; therefore they are of great use in crane work and for starting purposes of electric motors, steam, gas, and oil engines.

Horse-Power Formula.

An authority gives the following method of calculating the horse-power that can be transmitted by a worm gear.

$$\text{H.P.} = \text{K} \times \text{A} \times \text{T} \times r^{\frac{1}{2}} \times \text{R}^{\frac{1}{2}} \times \text{S},$$

where,

K = constant; A = projected area of worm wheel in sq. ins.; T = No. of teeth in contact; r = pitch radius of worm in ins.; R = revs. per minute of worm; S = function of lead angle.

For K, use:

Hardened mild steel worm and phosphor-bronze wheel						**K = 0·14**
Soft	,,	,,	,,	,,	,,	**K = 0·1**
Hardened	,,	,,	**cast-iron wheel**			**K = 0·08**
Soft	,,	,,	,,	,,		**K = 0·07**

For intermittent work multiply the above by 1·3.

For T, use:

Worm Threads	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
T	**1·4**	**1·5**	**1·66**	**1·8**	**1·9**	**2**	**2·1**	**2·2**

For S, use:

Lead Angle	**5°**	**10°**	**15°**	**20°**	**25°**	**30°**	**35°**	**40°**	**45°**
S	**0·09**	**0·18**	**0·27**	**0·36**	**0·45**	**0·55**	**0·65**	**0·77**	**0·9**

Lubrication.

Efficient lubrication and use of suitable lubricants are important factors in gearing design and working. Heavy tooth pressure and high velocity renders the selection of proper lubricant and means of lubrication highly important, and in consequence considerable attention has lately been given to this problem. Such questions as churning loss, viscosity, oxidation, and method of supplying oil to the surfaces concerned are factors which require careful consideration. Generally a fairly light oil is recommended for most purposes

The following tables relating to lubricants are reproduced from David Brown & Sons' technical data, and are supplied by them for use of gear users.

TABLE 1.

APPROXIMATE VISCOSITY OF OIL AT 140°F (SECS. REDWOOD I.)

Material or Ultimate Tensile Strength of Wheel	PITCH LINE SPEED (Feet per Min.)						
	Slow Speed up to 100	100 200	200 500	500 1,000	1,000 2,000	2,000 5,000	Over 5,000
Fabric, Cast-Iron or Bronze ...	400	275	200	140	110	90	—
Steel : 30—40 tons/in²	600	400	275	200	140	110	90
" 40—50 tonsi/n²	600	400	275	200	140	110	90
" 50—65 tons/in²	600	600	400	275	200	140	110
" 65—80 tons/in²	600	600	400	275	200	140	110
" 80—100 tons/in²	1000	600	600	400	275	200	140
Casehardened...	1000	600	600	400	275	200	140

TABLE 2.

APPROVED LUBRICANTS

Approximate Viscosity Required at 140°F.	MAKER					
	Alexander Duckham & Co. Ltd.	Shell-Mex & B P. Ltd.	Vacuum Oil Co. Ltd.	Valvoline Oil Co. Ltd.		C. C. Wakefield & Co. Ltd.
90	H.2 (87)	B.C.8 (95)	Etna Heavy Medium (95)	V.L.M. (82)	V.P.R. 206 (81)	"C" Engine Oil (95)
110	H.3 (110)	B.C.9 (110)	D.T.E. Heavy Medium (110)	S.M.R. (106)	V.P.R. 306 (107)	"MB" Engine Oil (110)
140	H.4 (150)	C.Y.1 (135)	D.T.E. Heavy (130)	A.A.M. (138)	V.P.R. 506 (156)	Deusol "G" (135)
200	N.P.D.3 (200)	C.Y.3 (210)	D.T.E. Extra Heavy (200)	X.R.M.E. (207)	V.P.R. 707 (213)	Heavy "M" Engine Oil (220)
275	S.W.S. (300)	C.5 (360)	D.T.E. B.B. (275)	V.H.E. (277)	V.P.R. 808 (280)	Alpha Engine Oil (270)
400	N.P.D.5 (350)	C.5 (360)	D.T.E. A.A. (420)	M.H.M. (410)	Eboline Cylinder Oil (497)	Extra Alpha Engine Oil (420)
600	D. (600)	B.4 (550)	600W. Cylinder Oil (550)	S.R.C. (590)		No. 1 Red Cylinder Oil (600)
1000	N.4 (920)	B.6 (1100)	Extra Hecla (1100)	A.A.E.C. (985)		"CMP" Cylinder Oil (1050)

The number in brackets gives the actual viscosity at 140°F (Secs. Redwood 1.)

The following table, also by the same authority, gives oils suitable for worm drives.

Maker.	Grade of Oil.
Alexander Duckham & Co., Ltd.	N. 4.
Shell Mex and B.P. Ltd.	Spirax Gear Oil.
Vacuum Oil Co., Ltd.	Mobiloil 'C' Gargoyle. 600 W, or Extra Hecla.
Valvoline Oil Co., Ltd.	Edgewater.
C. C. Wakefield & Co., Ltd.	'D.'

Methods of Lubrication.—Except in the case of worm gears, splash lubrication is efficient for pitch-circle velocities up to 2,000 or 2,500 ft. per min. With this system it is important that the correct oil level be maintained in order to prevent over-heating and excessive loss due to the churning action produced. The oil level should be arranged and maintained so that the gear wheel is immersed to a depth not usually in excess of 1½ in. or less than ¾ in.

Systems of forced lubrication are essential when very high pitch-circle velocities are employed and also for high tensile steel units. Specially arranged oil tanks and oil coolers may be necessary, with circulating pumps, etc. Many double helical reduction gears are provided with spraying nozzles, and oil is supplied at pressures ranging from 1 lb. per sq. in. to 10 lbs. per sq. in., according to ruling conditions and manufacturers' own ideas on the subject.

For worm drives mineral oil is not generally recommended, and the splash system should be replaced by a pick-up attached to the worm shaft and arranged so as to deposit the oil, after dipping, on to the surfaces of the worm.

TABLE OF DIAMETRAL AND CORRESPONDING CIRCULAR PITCHES.

Diametral Pitch *P*	Circular Pitch *p*	Diametral Pitch *P*	Circular Pitch *p*	Diametral Pitch *P*	Circular Pitch *p*
0·5	6·2832	6·5	0·4832	12·5	0·2513
1·0	3·1416	7·0	0·4459	13·0	0·2416
1·5	2·0944	7·5	0·4188	13·5	0·2327
2·0	1·5708	8·0	0·3927	14·0	0·2229
2·5	1·3566	8·5	0·3696	14·5	0·2166
3·0	1·0472	9·0	0·3490	15·0	0·2094
3·5	0·8918	9·5	0·3306	15·5	0·2027
4·0	0·7854	10·0	0·3142	16·0	0·1963
4·5	0·6980	10·5	0·2992	17·0	0·1848
5·0	0·6283	11·0	0·2855	18·0	0·1745
5·5	0·5710	11·5	0·2732	19·0	0·1653
6·0	0·5286	12·0	0·2618	20·0	0·1571

TABLE OF PITCH CIRCLE DIAMETERS.

Number of Teeth.	CIRCULAR PITCH.								
	3/8"	1/2"	5/8"	3/4"	7/8"	1"	1 1/8"	1 1/4"	1 3/8"
10	1.194	1.592	1.989	2.387	2.785	3.183	3.581	3.979	4.377
11	1.313	1.751	2.188	2.626	3.064	3.501	3.939	4.377	4.814
12	1.432	1.910	2.387	2.865	3.342	3.820	4.297	4.775	5.252
13	1.552	2.069	2.586	3.104	3.621	4.138	4.655	5.173	5.690
14	1.671	2.228	2.785	3.342	3.899	4.456	5.013	5.570	6.127
15	1.790	2.387	2.984	3.581	4.178	4.775	5.371	5.968	6.565
16	1.910	2.546	3.183	3.820	4.456	5.093	5.730	6.366	7.003
17	2.029	2.706	3.382	4.058	4.735	5.411	6.088	6.764	7.440
18	2.149	2.865	3.581	4.297	5.013	5.730	6.446	7.162	7.878
19	2.268	3.024	3.780	4.536	5.292	6.048	6.804	7.560	8.316
20	2.387	3.183	3.979	4.775	5.570	6.366	7.162	7.958	8.754
21	2.507	3.342	4.178	5.013	5.849	6.685	7.520	8.356	9.191
22	2.626	3.501	4.377	5.252	6.127	7.003	7.878	8.754	9.629
23	2.745	3.660	4.576	5.491	6.406	7.321	8.236	9.151	10.067
24	2.865	3.820	4.775	5.730	6.685	7.639	8.594	9.549	10.504
25	2.984	3.979	4.974	5.968	6.963	7.958	8.952	9.947	10.942
26	3.104	4.138	5.173	6.207	7.242	8.276	9.311	10.345	11.380
27	3.223	4.297	5.371	6.446	7.520	8.594	9.669	10.743	11.817
28	3.342	4.456	5.570	6.685	7.799	8.913	10.027	11.141	12.255
29	3.462	4.615	5.769	6.923	8.077	9.231	10.385	11.539	12.693
30	3.581	4.775	5.968	7.162	8.356	9.549	10.743	11.937	13.130
31	3.700	4.934	6.167	7.401	8.634	9.868	11.101	12.335	13.568
32	3.820	5.093	6.366	7.639	8.913	10.186	11.459	12.732	14.006
33	3.939	5.252	6.565	7.878	9.191	10.504	11.817	13.130	14.443
34	4.058	5.411	6.764	8.117	9.470	10.823	12.175	13.528	14.881
35	4.178	5.570	6.963	8.356	9.748	11.141	12.533	13.926	15.319
36	4.297	5.730	7.162	8.594	10.027	11.459	12.892	14.324	15.756
37	4.417	5.889	7.361	8.833	10.305	11.777	13.250	14.722	16.194
38	4.536	6.048	7.560	9.072	10.584	12.096	13.608	15.120	16.632
39	4.655	6.207	7.759	9.311	10.862	12.414	13.966	15.518	17.069
40	4.775	6.366	7.958	9.549	11.141	12.732	14.324	15.915	17.507
41	4.894	6.525	8.157	9.788	11.419	13.051	14.682	16.313	17.945
42	5.013	6.685	8.356	10.027	11.698	13.369	15.040	16.711	18.382
43	5.133	6.844	8.555	10.265	11.976	13.687	15.398	17.109	18.820
44	5.252	7.003	8.754	10.504	12.255	14.006	15.756	17.507	19.258
45	5.371	7.162	8.952	10.743	12.533	14.324	16.114	17.905	19.695
46	5.491	7.321	9.151	10.982	12.812	14.642	16.473	18.303	20.133

TABLE OF PITCH CIRCLE DIAMETERS.

Number of Teeth.	CIRCULAR PITCH.								
	1½"	1⅝"	1¾"	1⅞"	2"	2¼"	2½"	2¾"	3"
10	4.775	5.173	5.570	5.968	6.366	7.162	7.958	8.754	9.549
11	5.252	5.690	6.127	6.565	7.003	7.878	8.754	9.629	10.504
12	5.730	6.207	6.685	7.162	7.639	8.594	9.549	10.504	11.459
13	6.207	6.724	7.242	7.759	8.276	9.311	10.345	11.380	12.414
14	6.685	7.242	7.799	8.356	8.913	10.027	11.141	12.255	13.369
15	7.162	7.759	8.356	8.952	9.549	10.743	11.937	13.130	14.324
16	7.639	8.276	8.913	9.549	10.186	11.459	12.732	14.006	15.279
17	8.117	8.793	9.470	10.146	10.823	12.175	13.528	14.881	16.234
18	8.594	9.311	10.027	10.743	11.459	12.892	14.324	15.756	17.189
19	9.072	9.828	10.584	11.340	12.096	13.608	15.120	16.632	18.144
20	9.549	10.345	11.141	11.937	12.732	14.324	15.915	17.507	19.099
21	10.027	10.862	11.698	12.533	13.369	15.040	16.711	18.382	20.054
22	10.504	11.380	12.255	13.130	14.006	15.756	17.507	19.258	21.008
23	10.982	11.897	12.812	13.727	14.642	16.473	18.303	20.133	21.963
24	11.459	12.414	13.369	14.324	15.279	17.189	19.099	21.008	22.918
25	11.937	12.931	13.926	14.921	15.915	17.905	19.894	21.884	23.873
26	12.414	13.449	14.483	15.518	16.552	18.621	20.690	22.759	24.828
27	12.892	13.966	15.040	16.114	17.189	19.337	21.486	23.635	25.783
28	13.369	14.483	15.597	16 711	17.825	20.054	22.282	24.510	26.738
29	13.846	15.000	16.154	17.308	18.462	20.770	23.077	25.385	27.693
30	14.324	15.518	16.711	17.905	19.099	21.486	23.873	26.261	28.648
31	14.801	16.035	17.268	18.502	19.735	22.202	24.669	27.136	29.603
32	15.279	16.552	17.825	19.099	20.372	22.918	25.465	28.011	30.558
33	15.756	17.069	18.382	19.695	21.008	23.635	26.261	28.887	31.513
34	16.234	17.587	18.939	20.292	21.645	24.351	27.056	29.762	32.468
35	16.711	18.104	19.496	20.889	22.282	25.067	27.852	30.637	33.423
36	17.189	18.621	20.054	21.486	22.918	25.783	28.648	31.513	34.377
37	17.666	19.138	20.611	22.083	23.555	26.499	29.444	32.388	35.332
38	18.144	19.656	21.168	22.680	24.192	27.215	30.239	33.263	36.287
39	18.621	20.173	21.725	23.276	24.828	27.932	31.035	34.139	37.242
40	19.099	20.690	22.282	23.873	25.465	28.648	31.831	35.014	38.197
41	19.576	21.207	22.839	24.470	26.101	29.364	32.627	35.889	39.152
42	20.054	21.725	23.396	25.067	26.738	30.080	33.423	36.765	40.107
43	20.531	22.242	23.953	25.664	27.375	30.796	34.218	37.640	41.062
44	21.008	22.759	24.510	26.261	28.011	31.513	35.014	38.515	42.017
45	21.486	23.276	25.067	26.857	28.648	32.229	35.810	39.391	42.972
46	21.963	23.794	25.624	27.454	29.285	32.945	36.606	40.266	43.927

TABLE OF PITCH CIRCLE DIAMETERS.

Number of Teeth.	CIRCULAR PITCH.								
	3/8"	1/2"	5/8"	3/4"	7/8"	1"	1 1/8"	1 1/4"	1 3/8"
47	5.610	7 480	9.350	11.220	13.090	14.961	16.831	18.701	20.571
48	5.730	7.639	9.549	11 459	13.369	15.279	17 189	19.099	21.008
49	5.849	7.799	9.748	11.698	13.648	15.597	17.547	19.496	21.446
50	5.968	7.958	9.947	11.937	13.926	15.915	17.905	19.894	21.884
51	6.088	8.117	10.146	12.175	14.205	16.234	18.263	20.292	22.321
52	6.207	8.276	10.345	12.414	14.483	16.552	18.621	20.690	22.759
53	6.326	8 435	10.544	12.653	14.762	16.870	18.979	21.088	23.197
54	6.446	8.594	10.743	12.892	15.040	17.189	19.337	21.486	23.635
55	6.565	8 754	10.942	13.130	15.319	17.507	19.695	21.884	24.072
56	6.685	8.913	11.141	13.369	15.597	17.825	20.054	22.282	24.510
57	6.804	9.072	11.340	13.608	15.876	18.144	20.412	22.680	24.948
58	6.923	9.231	11.539	13.846	16.154	18.462	20.770	23.077	25.385
59	7.043	9.390	11.738	14.085	16.433	18.780	21.128	23.475	25.823
60	7.162	9.549	11.937	14.324	16.711	19.099	21.486	23.873	26.261
61	7.281	9.708	12.136	14.563	16.990	19.417	21.844	24.271	26.698
62	7.401	9.868	12.335	14.801	17.268	19.735	22.202	24.669	27.136
63	7.520	10.027	12.533	15.040	17.547	20.054	22.560	25.067	27.574
64	7.639	10.186	12.732	15.279	17.825	20.372	22.918	25.465	28.011
65	7.759	10.345	12.931	15.518	18.104	20.690	23.276	25.863	28.449
66	7.878	10.504	13.130	15.756	18.382	21.008	23.635	26.261	28.887
67	7.998	10.663	13.329	15.995	18.661	21.327	23.993	26.658	29.324
68	8.117	10.823	13.528	16.234	18.939	21.645	24.351	27.056	29.762
69	8.236	10.982	13.727	16.473	19.218	21.963	24.709	27.454	30.200
70	8.356	11.141	13.926	16.711	19.496	22.282	25.067	27.852	30.637
71	8.475	11.300	14.125	16.950	19.775	22.600	25.425	28.250	31.075
72	8.594	11.459	14.324	17.189	20.054	22.918	25.783	28.648	31.513
73	8.714	11.618	14.523	17.427	20.332	23.237	26.141	29.046	31.950
74	8.833	11.777	14.722	17.666	20.611	23.555	26.499	29.444	32.388
75	8.952	11.937	14.921	17.905	20.889	23.873	26.857	29.842	32.826
76	9.072	12.096	15.120	18.144	21.168	24.192	27.215	30.239	33.263
77	9.191	12.255	15.319	18.382	21.446	24.510	27.574	30.637	33.701
78	9.311	12.414	15.518	18.621	21.725	24.828	27.932	31.035	34.139
79	9.430	12.573	15.717	18.860	22.003	25.146	28.290	31.433	34.576
80	9.549	12.732	15.915	19.099	22.282	25.465	28.648	31.831	35.014
81	9.669	12.891	16.114	19.337	22.560	25.783	29.006	32.229	35.452
82	9.788	13.050	16.313	19.576	22.839	26.101	29.364	32.627	35.889
83	9.907	13.210	16.512	19.815	23.117	26.420	29.722	33.025	36.327

TABLE OF PITCH CIRCLE DIAMETERS.

Number of Teeth.	CIRCULAR PITCH.								
	1½"	1⅝"	1¾"	1⅞"	2"	2¼"	2½"	2¾"	3"
47	22.441	24.311	26.181	28.051	29.921	33.661	37 401	41.142	44.882
48	22.918	24.828	26.738	28.648	30.558	34.377	38.197	42.017	45.837
49	23.396	25.345	27.295	29.245	31.194	35.094	38.993	42.892	46.792
50	23.873	25.863	27.852	29.842	31.831	35.810	39.789	43.768	47 746
51	24.351	26.380	28.409	30.438	32.468	36.526	40.585	44.643	48.701
52	24.828	26.897	28.966	31.035	33.104	37.242	41.380	45.518	49.656
53	25.306	27.414	29.523	31.632	33.741	37.958	42.176	46.394	50.611
54	25.783	27.932	30.080	32.229	34.377	38.675	42.972	47.269	51.566
55	26.261	28.449	30.637	32.826	35.014	39.391	43 768	48.144	52.521
56	26.738	28.966	31.194	33.423	35.651	40.107	44.563	49 020	53.476
57	27.215	29.483	31.751	34.019	36.287	40.823	45 359	49.895	54.431
58	27.693	30.001	32.308	34.616	36.924	41.539	46.155	50.770	55.386
59	28.170	30.518	32.865	35.213	37.561	42.256	46.951	51.646	56.341
60	28.648	31.035	33.423	35.810	38.197	42.972	47 746	52.521	57.296
61	29.125	31.552	33.980	36.407	38.834	43.688	48.542	53 396	58.251
62	29.603	32.070	34.537	37.004	39.470	44.404	49.338	54.272	59.206
63	30.080	32.587	35.094	37.600	40.107	45.120	50.134	55.147	60.161
64	30.558	33.104	35.651	38.197	40.744	45.837	50.930	56.023	61.115
65	31.035	33.621	36.208	38.794	41.380	46.553	51 725	56.898	62.070
66	31.513	34.139	36.765	39.391	42.017	47.269	52.521	57.773	63.025
67	31.990	34.656	37 322	39.988	42.654	47.985	53.317	58.649	63.980
68	32.468	35.173	37.879	40.585	43.290	48.701	54.113	59.524	64.935
69	32.945	35.690	38.436	41.181	43.927	49.418	54.908	60.399	65.890
70	33.423	36.208	38.993	41.778	44.563	50.134	55.704	61.275	66.845
71	33.900	36.725	39.550	42.375	45.200	50.850	56.500	62.150	67.800
72	34.377	37.242	40.107	42.972	45.837	51.566	57.296	63.025	68.755
73	34.855	37.760	40.664	43.569	46.473	52.282	58.092	63.901	69.710
74	35.332	38.277	41.221	44.165	47 110	52.999	58.887	64.776	70.665
75	35.810	38.794	41.778	44.762	47.746	53.715	59.683	65.651	71.620
76	36.287	39.311	42.335	45.359	48.383	54.431	60.479	66 527	72.575
77	36.765	39.829	42.892	45.956	49.020	55.147	61.275	67.402	73.530
78	37.242	40.346	43.449	46.553	49.656	55.863	62.070	68.277	74.485
79	37.720	40.863	44.006	47.150	50.293	56.580	62.866	69.153	75.439
80	38.197	41.380	44.563	47.746	50.939	57.296	63.662	70.028	76.394
81	38.675	41.898	45.120	48.343	51.566	58.012	64.458	70.904	77.349
82	39.152	42.415	45.677	48.940	52.203	58.728	65.254	71.779	78.304
83	39.630	42.932	46.235	49.537	52.839	59.444	66.049	72.654	79.259

TABLE OF PITCH CIRCLE DIAMETERS.

Number of Teeth.	CIRCULAR PITCH.								
	3/8″	1/2″	5/8″	3/4″	7/8″	1″	1 1/8″	1 1/4″	1 3/8″
84	10.027	13.369	16.711	20.054	23.396	26.738	30.080	33.423	36.765
85	10.146	13.528	16.910	20.292	23.674	27.056	30.438	33.820	37.202
86	10.265	13.687	17.109	20.531	23.953	27.375	30.796	34.218	37.640
87	10.385	13.846	17.308	20.770	24.231	27.693	31.155	34.616	38.078
88	10.504	14.006	17.507	21.008	24.510	28.011	31.513	35.014	38.515
89	10.624	14.165	17.706	21.247	24.788	28.330	31.871	35.412	38.953
90	10.743	14.324	17.905	21.486	25.067	28.648	32.229	35.810	39.391
91	10.862	14.483	18.104	21.725	25.345	28.966	32.587	36.208	39.829
92	10.982	14.642	18.303	21.963	25.624	29.285	32.945	36.606	40.266
93	11.101	14.801	18.502	22.202	25.902	29.603	33.303	37.004	40.704
94	11.220	14.960	18.701	22.441	26.181	29.921	33.661	37.401	41.142
95	11.340	15.120	18.900	22.680	26.460	30.239	34.019	37.799	41.579
96	11.459	15.279	19.099	22.918	26.738	30.558	34.377	38.197	42.017
97	11.579	15.438	19.298	23.157	27.017	30.876	34.736	38.595	42.455
98	11.698	15.597	19.496	23.396	27.295	31.194	35.094	38.993	42.892
99	11.817	15.756	19.695	23.635	27.574	31.513	35.452	39.391	43.330
100	11.937	15.915	19.894	23.873	27.852	31.831	35.810	39.789	43.768
101	12.056	16.075	20.093	24.112	28.131	32.149	36.168	40.187	44.205
102	12.175	16.234	20.292	24.351	28.409	32.468	36.526	40.585	44.643
103	12.295	16.393	20.491	24.589	28.688	32.786	36.884	40.982	45.081
104	12.414	16.552	20.690	24.828	29.966	33.104	37.242	41.380	45.518
105	12.533	16.711	20.889	25.067	29.245	33.423	37.600	41.778	45.956
106	12.653	16.870	21.088	25.306	29.523	33.741	37.958	42.176	46.394
107	12.772	17.030	21.287	25.544	29.802	34.059	38.317	42.574	46.831
108	12.892	17.189	21.486	25.783	30.080	34.377	38.675	42.972	47.269
109	13.011	17.348	21.685	26.022	30.359	34.696	39.033	43.370	47.707
110	13.130	17.507	21.884	26.261	30.637	35.014	39.391	43.768	48.144
111	13.250	17.666	22.083	26.499	30.916	35.332	39.749	44.165	48.582
112	13.369	17.825	22.282	26.738	31.194	35.651	40.107	44.563	49.020
113	13.488	17.985	22.481	26.977	31.473	35.969	40.465	44.961	49.457
114	13.608	18.144	22.680	27.215	31.751	36.287	40.823	45.359	49.895
115	13.727	18.303	22.879	27.454	32.030	36.606	41.181	45.757	50.333
116	13.846	18.462	23.077	27.693	32.308	36.924	41.539	46.155	50.770
117	13.966	18.621	23.276	27.932	32.587	37.242	41.898	46.553	51.208
118	14.085	18.780	23.475	28.170	32.865	37.561	42.256	46.951	51.646
119	14.205	18.939	23.674	28.408	33.144	37.879	42.614	47.349	52.083
120	14.324	19.099	23.873	28.648	33.423	38.197	42.972	47.746	52.521

TABLE OF PITCH CIRCLE DIAMETERS.

Number of Teeth.	CIRCULAR PITCH.								
	1½″	1⅝″	1¾″	1⅞″	2″	2¼″	2½″	2¾″	3″
84	40.107	43.449	46.792	50.134	53.476	60.161	66.845	73.530	80.214
85	40.585	43.967	47.349	50.731	54.113	60.877	67.641	74.405	81.169
86	41.062	44.484	47.906	51.327	54.749	61.593	68.437	75.280	82.124
87	41.539	45.001	48.463	51.924	55.386	62.309	69.232	76.156	83.079
88	42.017	45.518	49.020	52.521	56.023	63.025	70.028	77.031	84.034
89	42.494	46.036	49.577	53.118	56.659	63.742	70.824	77.906	84.989
90	42.972	46.553	50.134	53.715	57.296	64.458	71.620	78.782	85.944
91	43.449	47.070	50.691	54.312	57.932	65.174	72.415	79.657	86.899
92	43.927	47.587	51.248	54.908	58.569	65.890	73.211	80.532	87.854
93	44.404	48.105	51.805	55.505	59.206	66.606	74.007	81.408	88.808
94	44.882	48.622	52.362	56.102	59.842	67.323	74.803	82.283	89.763
95	45.359	49.139	52.919	56.699	60.479	68.039	75.599	83.158	90.718
96	45.837	49.656	53.476	57.296	61.116	68.755	76.394	84.034	91.673
97	46.314	50.174	54.033	57.893	61.752	69.471	77.190	84.909	92.628
98	46.792	50.691	54.590	58.489	62.389	70.187	77.986	85.785	93.583
99	47.269	51.208	55.147	59.086	63.025	70.904	78.782	86.660	94.538
100	47.746	51.725	55.704	59.683	63.662	71.620	79.577	87.535	95.493
101	48.224	52.243	56.261	60.280	64.299	72.336	80.373	88.411	96.448
102	48.701	52.760	56.818	60.877	64.935	73.052	81.169	89.286	97.403
103	49.179	53.277	57.375	61.474	65.572	73.768	81.965	90.161	98.358
104	49.656	53.794	57.932	62.070	66.208	74.485	82.761	91.037	99.313
105	50.134	54.312	58.489	62.667	66.845	75.201	83.556	91.912	100.268
106	50.611	54.829	59.046	63.264	67.482	75.917	84.352	92.787	101.223
107	51.089	55.346	59.604	63.861	68.118	76.633	85.148	93.663	102.177
108	51.566	55.863	60.161	64.458	68.755	77.349	85.944	94.538	103.132
109	52.044	56.381	60.718	65.055	69.392	78.065	86.739	95.413	104.087
110	52.521	56.898	61.275	65.651	70.028	78.782	87.535	96.289	105.042
111	52.999	57.415	61.832	66.248	70.665	79.498	88.331	97.164	105.997
112	53.476	57.932	62.389	66.845	71.301	80.214	89.127	98.039	106.952
113	53.954	58.450	62.946	67.442	71.938	80.930	89.923	98.915	107.907
114	54.431	58.967	63.503	68.039	72.575	81.646	90.718	99.790	108.862
115	54.908	59.484	64.060	68.636	73.211	82.363	91.514	100.666	109.817
116	55.386	60.001	64.617	69.232	73.848	83.079	92.310	101.541	110.772
117	55.863	60.519	65.174	69.829	74.485	83.795	93.106	102.416	111.727
118	56.341	61.036	65.731	70.426	75.121	84.511	93.901	103.292	112.682
119	56.818	61.553	66.288	71.023	75.758	85.227	94.697	104.167	113.637
120	57.296	62.070	66.845	71.620	76.394	85.944	95.493	105.042	114.592

SECTION XXII

PART I

MACHINE TOOLS—PORTABLE TOOLS—POWER REQUIRED BY MODERN HIGH-SPEED MACHINE TOOLS—WOOD-WORKING AND PATTERN-MAKING MACHINERY—INTERNAL AND EXTERNAL BROACHING—TOOL STEEL—TUNGSTEN CARBIDE-TIPPED CUTTING TOOLS, THEIR PREPARATION, SHAPE AND USE—REPAIRING WORN PARTS BY DEPOSITION OF METAL—HARDNESS OF MATERIALS (pp. 1019-1096)

(Revised by N. K. Beard, M.I.Mech.E.)

PART II

GRINDING, ABRASIVES, AND POLISHING APPLIANCES (pp. 1097-1116)

(Revised by Alfred B. Searle)

PART III

STEAM AND POWER HAMMERS (pp. 1117-1122)

SECTION XXII

PART I

MACHINE TOOLS—PORTABLE TOOLS—POWER REQUIRED BY MODERN HIGH-SPEED MACHINE TOOLS—WOOD-WORKING AND PATTERN-MAKING MACHINERY—INTERNAL AND EXTERNAL BROACHING — TOOL STEEL — TUNGSTEN CARBIDE-TIPPED CUTTING TOOLS, THEIR PREPARATION, SHAPE AND USE — REPAIRING WORN PARTS BY DEPOSITION OF METAL—HARDNESS OF MATERIALS

(Revised by N. K. Beard, M.I.Mech.E.)

Machine Tools.

The trend of modern machine tool design has been governed by several progressive developments which have each played their part in the composition of the present-day machine tool. The demand by the user for guaranteed output times has caused the machine tool maker to build the machine tool round the job. This is evident in the many excellent multi-tool lathes, multi-spindle milling and drilling machines which have been in the first place designed specially for the motor-car industry and which have later been modified for other manufacturing purposes. Many of the above machines have been made either automatic or semi-automatic, the operator having only to load and unload the workpiece and to start and stop the machine. On other machines, for example those of the capstan and turret type, stops are provided so that after the tools have been set the operator can produce articles of the same size within very close limits for long periods dependent on the tool life only. Labour saving devices such as quick power motions have been applied to reduce the idle time and considerable use has been made of air, hydraulic and electric power for chucking the workpiece and for clamping down or locking moving parts of the machine. All these features ease the fatigue of the operator and tend to increase the productive efficiency of the machine tool. The machine tool manufacturer has also taken advantage of the higher cutting speeds which can be obtained by using the various types of hard metal or cimented carbide tips.

Economics have also had their influence on machine tool design, for the purchaser must consider the initial cost of the machine tool together with the probable return and the length of its useful life.

The most recent improvements in machine tools to meet modern conditions are given below :—

1. Increased depth of bed, increased thickness of bed walls and flanges, and the introduction of diagonal or elliptical cross ribs, thus increasing the rigidity of the bed to resist distortions and absorb vibration when working at the maximum speed and feed.
2. The material of the bed has been improved and is now often made of high duty cast iron, flame hardened, on the wearing surfaces of the bed shears.
3. Main spindle bearings are generally of the ball or roller type pre-loaded or may be made of special anti-friction metal with forced feed lubrication.
4. Automatic pressure lubrication to bearings in headstock, gear boxes, saddle aprons and saddles with visual indicators.
5. Shafts and main spindles made of alloy steels, heat treated and ground.
6. Splines used instead of keys and keyways to give increased strength and more accurate fits.
7. Improved design of spindle noses to facilitate changing faceplates and chucks.
8. Main gearing in headstocks and gear boxes made of heat-treated high tensile alloy steel with teeth generated and ground. The pressure angle used being mostly 20° with teeth corrections on pinions below 22 T. to prevent undercutting. The gear teeth are rounded to facilitate sliding engagement.
9. Power traverse motions to all moving saddles.
10. Narrow guides to all slides, inserted hardened steel wearing plates, taper gibs to facilitate alignment adjustment and covered slideways.
11. Live roller bearing centres.
12. Centralised controls.
13. Direct motor drive either through vee belts or couplings. Flange mounted and built in motors and electric control gear are also used to advantage.
14. Spotlighting by low voltage system to give the maximum illumination of the workpiece.
15. Compressed air chucking and clamping.
16. Micrometer stops and dials.
17. Tool setting devices.

There is still a useful place for belt-driven machines, particularly those of the single pulley type, although the majority of the machine tools now made are made with some form of the individual motor drive.

There have been considerable developments in the use of hydraulic feeds as applied to grinding machines, milling machines, broaching machines and special lathes. Hydraulic feeds have also been used on planing and shaping machines.

Portable machine tools have also been designed to meet modern requirements, for in shipyards, locomotive running sheds and on heavy steel constructional work it is often more economical to take the machine to the job. These machines may be summarised as follows: Deck planing and facing machines, portable cylinder boring machines, portable valve facing machines, portable outside crank pin truing machines, portable manhole facing machines, portable boiler stay hole drilling and tapping machines and portable drilling machines for structural work.

High Speed Lathes.

The increase in r.p.m. of the main spindle demands a greater number of speed changes, and a higher ratio in successive spindle speeds. This has been made necessary in order to reduce the r.p.m. to a level sufficiently low to keep the cutting speed on large diameters within reasonable limits.

Ratio of successive spindle speeds accepted as good practice in modern design is:—

For step cone headstock, 1 to 1·3 or 1·36.
For all gear headstock, 1 to 1·4 or 1·43.

Note.—The lowest and highest spindle speeds should be determined by the nature of the work the lathe is to be used for. Thus for turning relatively soft non-ferreous metals, both the lowest and the highest speeds will greatly exceed that allowed for machining the harder ferreous metals.

Table I gives the highest and lowest spindle speeds and ratios, also the total number of speed changes usually allowed for modern high-speed lathes using good quality high-speed steel or tungsten carbide tools.

Table I.

Swing over bed	9″	12″	16″	20″	24″	28″	36″
Highest r.p.m.	858	717	600	500	400	330	94·8
Lowest r.p.m.	18	17	12	10	8	6	2·13
Ratio of highest to lowest	47·7	42	50	50	50	55	44·5
No. of spindle speeds	12	12	12	12	12	12	16

The spindle speeds given in the above table are for single pulley drive direct from the main line shaft, and for constant speed motor drive direct to the lathe. These can be increased where desired by means of a two-speed countershaft in the case of the belt drive, and by the use of a variable speed motor for direct motor drive. But generally the speeds given are found sufficient to cover a wide range of work.

For practical purposes, when designing the headstock of a medium-sized general purpose lathe, it is convenient to first decide the highest r.p.m. at which the spindle is to run, that being the factor mainly governing the type of bearing to be fitted to carry the maximum speed and load. The lowest speed will then be determined by the ratio of successive intermediate spindle speeds, and the number of speed changes, taking care that the lowest speed obtainable by this rule will allow a proper cutting speed when the tool is operating at the periphery of the maximum swing of the lathe.

Metal Cutting Speeds.

Cutting speeds obtainable with modern high-speed lathes.

Table II.

Cutting Tool.	Average Speed in Ft. per Minute.
High-speed steel	100 to 120 on medium mild steel
Tungsten carbide or similar alloy	350 to 450 on medium mild steel

Power Required for Cutting.

The factors to be taken into account when calculating the power for cutting metals are, the angle of cutting edge of the tool, condition of the tool in use and the hardness of the material to be cut.

A useful formula for finding the H.P. required for cutting steel with properly shaped tools is given below :—

$$\text{H.P.} = \frac{A \times P \times S \times C}{33{,}000}$$

where A = area of cut in square inches (depth and feed),
P = pressure on tool,
S = cutting speed in feet per minute,
C = 2,240.

The value for P being taken as	45 tons per sq. in. of cut for	cast iron 200 Brinell.
,, ,, ,, ,, ,, ,,	75 ,, ,, ,, ,, ,, ,,	cast iron above 200 Brinell.
,, ,, ,, ,, ,, ,,	60 ,, ,, ,, ,, ,, ,,	copper.
,, ,, ,, ,, ,, ,,	60 ,, ,, ,, ,, ,, ,,	soft brass.
,, ,, ,, ,, ,, ,,	150 ,, ,, ,, ,, ,, ,,	hard brass.
,, ,, ,, ,, ,, ,,	90 ,, ,, ,, ,, ,, ,,	manganese bronze.
,, ,, ,, ,, ,, ,,	95 ,, ,, ,, ,, ,, ,,	soft M.S. (28/35 tons tensile).
,, ,, ,, ,, ,, ,,	120 ,, ,, ,, ,, ,, ,,	medium M.S. (40/45 tons tensile)
,, ,, ,, ,, ,, ,,	150 ,, ,, ,, ,, ,, ,,	hard M.S. (60 tons tensile).

The resulting H.P. is that required at the point of the tool, a margin of 9 per cent. to 12 per cent. should be allowed for frictional losses.

Guiding the Slides.

All modern high-speed lathes are now built with the narrow guide fitted to the saddle and slides, the guide for the saddle being arranged on the front shear of the lathe bed. The higher the ratio of length to width, the more durable and efficient is the guide.

$$\text{Ratio. } \frac{L}{W} \text{ where } L = \text{length and } W = \text{width of guide.}$$

In good practice the ratio of L to W = 7 to 1, but in deciding this ratio other considerations in the design must naturally be taken into account.

Back Centres for Loose Headstock.

Increased r.p.m. of high speed lathes throws a heavy duty on the loose headstock and back centre, and to ensure the necessary rigidity, the loose headstock is now designed on more liberal lines than formerly, to prevent the tendency to lift under heavy cutting. Separate guide ways on the lathe bed are usually provided to control alignment, and to allow the extended ends of the saddle to clear the sides. Crosswise adjustment is provided for taper turning, and locating faces for resetting in central position for parallel turning.

Back centres require special considerations. For the high revolutions now common in everyday practice, the ordinary hardened carbon steel back centre is unsatisfactory, the point being easily destroyed. Where the stationary or dead centre is used it should be made from high-speed steel correctly heat-treated and ground.

But the best practice now is to use the revolving back centre, for high spindle speeds. These are of special construction and have to be accurately made to ensure interchangeability of the running centre. The radial load is taken on a taper roller bearing, fitted at the outside end of the holder or housing, the inside end running on ball bearings, and the end thrust taken on ball thrust washers. The stationary housing is made of nickel chrome heat-treated steel, being ground to fine limit gauges after hardening.

TABLE III.

PRINCIPAL DIMENSIONS AND RANGE OF REVOLVING CENTRES BY LANG & SONS, LTD., JOHNSTONE

Shank. Morse Taper No.	Overall Length of Centre.	Shank Length.	Front Bearing Length.	Centre Point Length.	Front Bearing Outside Dia.	Shank Dia. at Front End.	Shank Dia. at Back End.	Angle of Point.
	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	°
3	$6\frac{5}{16}$	$3\frac{9}{16}$	$1\frac{25}{32}$	$\frac{13}{16}$	$2\frac{1}{16}$	·938	$\frac{11}{16}$	60
4	$7\frac{1}{4}$	$4\frac{7}{16}$	2	$\frac{7}{8}$	$2\frac{11}{16}$	1·231	$\frac{15}{16}$	60
5	$9\frac{5}{16}$	$4\frac{13}{16}$	$2\frac{3}{8}$	$1\frac{1}{8}$	$3\frac{3}{16}$	1·748	$1\frac{7}{16}$	60
6	$13\frac{1}{4}$	$7\frac{3}{8}$	$3\frac{13}{16}$	$1\frac{1}{4}$	$5\frac{1}{4}$	2·484	2	60
7	$17\frac{13}{16}$	$10\frac{5}{8}$	$4\frac{9}{16}$	$2\frac{13}{16}$	$6\frac{3}{8}$	3·270	$2\frac{5}{8}$	60

Hydraulic Feeds.

Hydraulic feeds are now extensively used on grinding machines, milling machines, broaching machines and cold sawing machines. They have been used, but not to a great extent, on shaping and planing machines. Many special purpose or manufacturing lathes have also been equipped with hydraulic feeds, but for obvious practical reasons hydraulic feeds are not used on standard lathes.

The advantages of hydraulic feeds are :—

(*a*) The rate of feed may be infinitely varied between given limits.

(*b*) The saddle can travel against a dead stop thus enabling accurate lengths to be turned.

(*c*) The hydraulic medium can be used for the quick return traverse.

(*d*) The feed is flexible and tends to adjust itself should the depth of cut increase or should there be hard spots in the metal being machined.

There are two systems used in hydraulic feeds: one uses a variable delivery pump and the other a constant delivery pump. In the former case the rate of feed is controlled by varying the pump delivery. In the second system, see fig. 1, a constant delivery pump is used and the feed varia

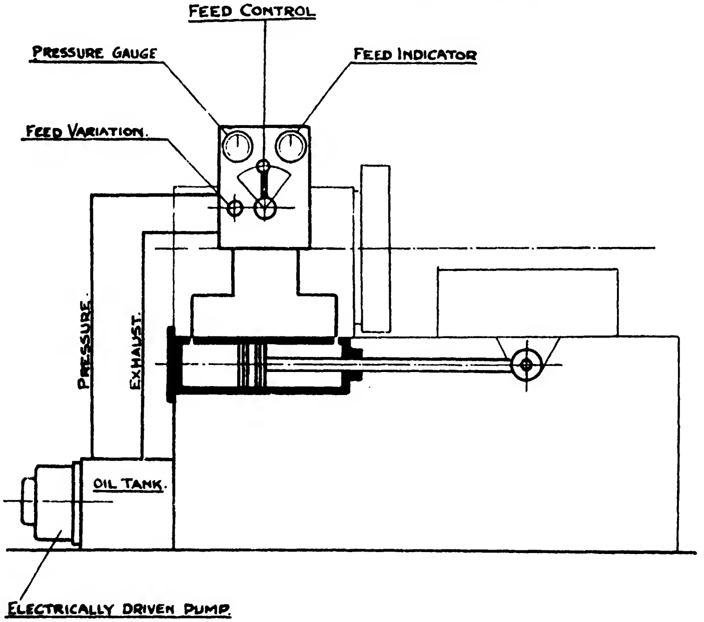
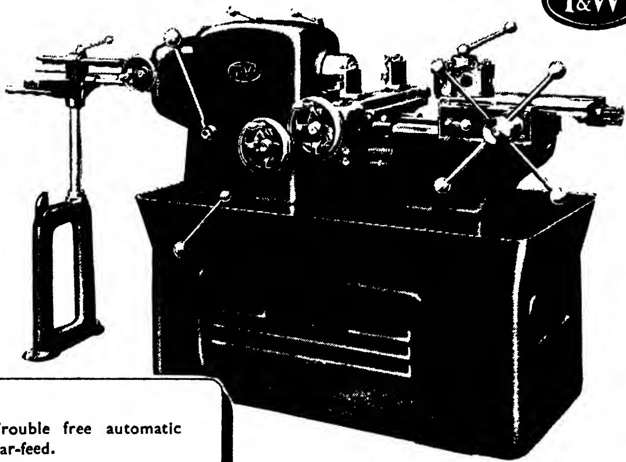


DIAGRAM OF HYDRAULIC FEED SYSTEM.

FIG. 1.

tions are obtained by controlling the rate of exhaust through an adjustable metering valve. The working pressure is controlled by a relief valve, at the pump discharge, which is set to give the desired pressure. Pressure is always connected to the right hand side of the cylinder, the left hand side being alternatively connected to exhaust for power feed and to pressure for quick power return of the saddle. The working pressure of the system is 1,200 lbs. per sq. in. supplied by a Towler high-speed pump driven by a $\frac{1}{2}$ h.p. electric motor. Hydraluic power may also be used to control the tailstock barrel and to operate the chuck.

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This system has been successfully applied to boring lathes and rough turning lathes where the cuts have been as large as 0·030 in. feed by ⅜ in. deep at 230 ft. per min. using two tungsten carbide tools simultaneously.

On a turning lathe having 6½ ins. swing, 18 ins. between centres, 12 spindle speeds giving 80 to 2,500 r.p.m. Feeds on all tool slides, longitudinal and crosswise, variable from 0 to 20 ins. per minute. Constant return speed to all slides 25 feet per minute. Hydraulic power from constant pressure type pump giving 100 lbs. per sq. in. for quick return, and variable from a metering pump for controlling the rate of feed for cutting.

Power for driving the main spindle, 5 h.p. at 1,430 r.p.m.

Power for driving the hydraulic pump, 1 h.p. at 1,420 r.p.m.

Several turret lathes have lately been introduced using pressure oil for feeding, indexing and automatic speed changing as each tool comes into action. Self-contained units with all controls and pumps in one box are becoming increasingly available for all types of machines, along with new methods of compensation for feed variation due to changes in temperature and oil viscosity.

Note.—**Super high-speed steel and tungsten carbide tools are used exclusively on this machine. A cutting speed of 3,825 ft. per min. can be obtained on a diameter of 6 ins.**

Planing Machines.

Planing machines usually perform the first machining operation on large castings, and therefore the planing machine must be accurate and reliable, for the horizontal and vertical surfaces it produces are often used as reference planes for subsequent operations. It is obvious that the more accurate the surface produced on the planing machine the easier the finishing operation, whether it be hand scraping or surface grinding. The combination of good design, suitability of materials and good workmanship makes for the reliability of the machine. The planing machine has undergone several stages of development in recent years, although the basic principles remain unaltered.

Belt Driven Planing Machines.

Light duty planing machines, particularly those of 4 ft. 0 ins. width of table and below, are generally driven through a two-speed countershaft and shifting belt built as an integral part of the machine. The countershaft is now generally driven by an electric motor, although it may be driven by belt from a convenient lineshaft. The countershaft drive is simple and cheap but has a limiting effect on the speed of the table, for it is difficult to design a belt shifting mechanism that will work satisfactorily at both high and low speeds.

Electrically Driven Planing Machines.

The improvements in high-speed steel and the introduction of cimented carbide-tipped tools have each made possible increased planing speeds.

Most planing machines, especially those with a 4 ft. 0 ins. width of table and upwards, are now almost entirely driven by reversing electric motors geared directly to the table and controlled by a 'Ward Leonard' type of control gear. With this type of drive it is possible to obtain a wide range of both cutting and return speeds controlled by fine graduations, the return speeds being independent of the cutting speeds. Motors of this type have constant torque characteristics from the lowest to approximately half-speed, and constant horse-power from half to maximum speed. In planing it is not usually a disadvantage that the power falls as the high speed drops, because on general work less power is often required at the lower speeds. Should, however, greater power be required at the lower speeds, it can often be obtained by introducing a reduction gear between the driving motor and the table. Fig. 2 shows the graphical representation of a typical example of the speed characteristics of a modern planing machine.

The advantage of using cimented carbide-tipped tools is less apparent on planing machines than on any other machine tool, for, apart from the interrupted cutting action, the time taken for the return strokes reduces any gain that may be made on the cutting stroke. If, for example, the stroke being performed is 20 ft. and the cutting speed is 60 ft. per minute using high-speed steel with a return speed of 200 ft. per minute, the cycle time, that is for one cutting and one return stroke, will be

$$\left(\frac{20}{60} + \frac{20}{200}\right) \times 60 = 26 \text{ seconds.}$$

If we assume that by using a cimented carbide tool we now increase the cutting speed to 200 ft. per minute, the cycle time will be

$$\left(\frac{20}{200} + \frac{20}{200}\right) \times 60 = 12 \text{ seconds.}$$

Thus by increasing the cutting speed 3⅓ times the cycle time has only been increased 2⅙ times. Had a similar increase in cutting speed been made on a lathe, the full advantage would have been obtained.

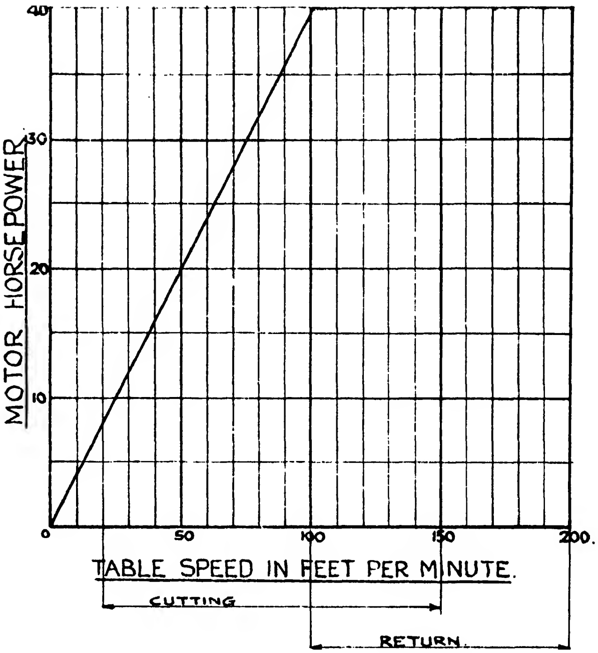


FIG. 2.

The peak load comes on the planer motor at the point of reversal on the idle stroke, because it has to bring both the table gearing and armature of the motor to rest before the cutting stroke can commence. Special attention is paid by the motor designers to keep down the size and speed of the motor armature to facilitate the easy reversal. In calculating the stored energy of the table it is usual to assume that the weight of the job is equal to the weight of the table.

The beds of planing machines are now usually made with the slideways at least twice the total length of the table stroke. This increased length of bed allows for the use of tandem tables which can be bolted together or used separately. When they are used separately one can be being loaded while the other is in operation. A special dumb bell connecting link is provided to bring the table rack into gear with the driving pinion. The drive to the table has received considerable attention and the most favoured type of drive for medium sized machines is known as the spiral drive. This consists of a multi-start worm or spiral pinion with a 45° lead angle which engages with the table rack. The spiral pinion shaft, which lies at an angle of 45° to the bed ways, is provided with ball thrust bearings to take the end thrust. At the end of spiral pinion shaft there is usually a spur gear which in turn engages with a spiral pinion also with a lead angle of 45° on the motor armature shaft. This arrangement caused the driving motor to lie parallel with the table ways and makes a compact drive. Another type of drive makes use of worm gearing.

Electric Feed Motions.

Electric feeds are now extensively used on planing machines, the electric feed motor being connected by gearing to the feed shafts and screws.

The usual principle is that on the feed switch being tripped by a dog, either on the side of the planing machine table or on the disc control, the feed motor makes a predetermined number of revolutions and so gives the selected feed. The same motor is also used for the power traverse motions to the tool-boxes and for raising and lowering the cross slide. Not the least of the advantages of the electric feed is its silence in operation.

Cutting Speeds.

Cutting and return speeds of planers, even the large heavy type, have reached high levels. Cutting speeds of 500 ft. per minute, and quick return on that speed have been reached, but this may be considered quite abnormal and outside practical limits.

Cutting speeds of 300 ft. per minute and quick return can be run fairly satisfactorily on a 4 ft. × 4 ft. × 10 ft. planing machine, while a cutting speed of 100 ft. per minute, and a quick return speed of 180 ft. per minute has been proved quite practicable on a large planing machine with a capacity of 12 ft. 6 ins. wide × 10 ft. high × 30 ft. long.

TABLE IV.

Size of Planer in Width, Height and Length of Bed.	Cutting Speed. F.P.M.	Return Speed. F.P.M.	No. of Feeds.	Range of Feeds.	Size of Table.
3′ 0″ × 3′ 0″ × 16′ 0″	20–200	110–220	14	$\frac{1}{64}$″–$\frac{3}{4}$″	2′ 6″ × 8′ 0″
3′ 6″ × 3′ 6″ × 20′ 0″	20–200	110–220	14	$\frac{1}{64}$″–$\frac{3}{4}$″	3′ 0″ × 10′ 0″
4′ 0″ × 4′ 0″ × 20′ 0″	20–200	110–220	14	$\frac{1}{64}$″–$\frac{3}{4}$″	3′ 6″ × 10′ 0″
4′ 0″ × 4′ 0″ × 24′ 0″	20–200	110–220	14	$\frac{1}{64}$″–$\frac{3}{4}$″	3′ 9″ × 12′ 0″
4′ 6″ × 4′ 6″ × 24′ 0″	20–200	110–220	14	$\frac{1}{64}$″–$\frac{3}{4}$″	4′ 0″ × 12′ 0″
5′ 0″ × 5′ 0″ × 24′ 0″	20–200	110–220	14	$\frac{1}{64}$″–$\frac{3}{4}$″	4′ 6″ × 12′ 0″
6′ 0″ × 6′ 0″ × 24′ 0″	20–200	110–220	14	$\frac{1}{64}$″–$\frac{3}{4}$″	5′ 6″ × 12′ 0″

Note.—The fine feeds are used for roughing cuts and the coarse feeds for the scraping or finishing cut.

Most planing machines of the high-speed type are equipped with a solenoid tool lifter, to hold the tool clear of the work on the return stroke. Controls are centralised by means of pendant switches which hang within easy reach of the operator.

MILLING MACHINES.

Modern milling machines are made in the following types :—

(1) Vertical single spindle milling machines where the workpiece is mounted on a table which can be traversed past the cutter.

(2) Horizontal spindle milling machines of the planer type construction of table uprights and crossrail using cylindrical cutter.

(3) Planer type milling machines having one or more vertical cutter spindle heads or one or more horizontal cutter spindle heads mounted on the faces of the uprights or columns.

(4) Vertical spindle milling machines operating on heavy stationary workpieces.

(5) Horizontal milling machines for face milling on heavy stationary workpieces.

(6) Horizontal spindle knee and column type milling machines which may be of the planer, universal or manufacturing type.

Usually modern milling machines are electrically driven but it is possible to obtain the smaller types of machine arranged for belt drive. It has been found that to take full advantage of the use of modern high-speed steel cutters and cimented carbide inserted tooth cutters wider ranges of speeds and feeds with increased power have been necessary. It has also been necessary to provide more rigid machine frames and tables. Flywheels have been fitted to the spindles to damp down vibration. In many machines the feed and speed changes are obtained by the use of gear boxes which are built into the machines. Hydraulic feeds which are infinitely variable are now fitted as standard by some makers.

Most high speed milling machine spindles are mounted on taper roller bearings, but in certain cases it has been found that a plain lubricated bearing of the conical type gives better results.

If a machine is intended to be used on climb or down cut milling, see fig. 3., it is essential that the table be provided with a backlash eliminator, otherwise the cutter will pull into the workpiece to the extent of any existing backlash in the nut with disastrous effects on the cutter.

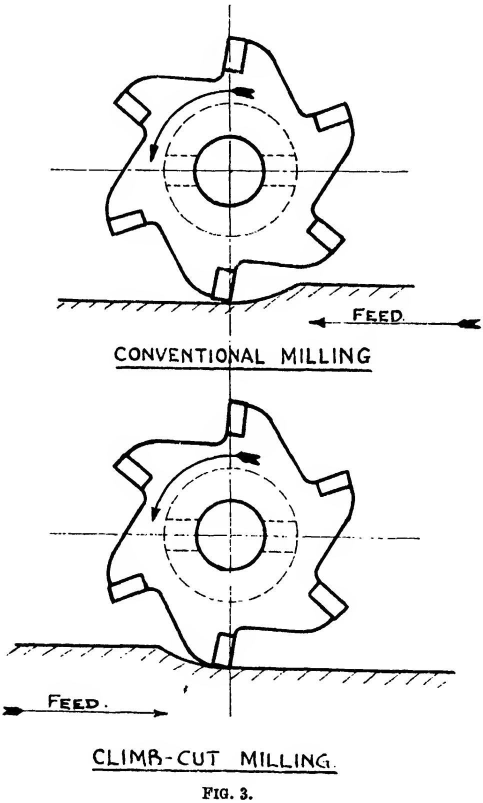


FIG. 3.

Milling practice is controlled by the conditions prevailing in the machining of each individual workpiece to a predetermined degree of accuracy.

One of the chief of these conditions is the amount of metal that can be removed per minute without distortion of the workpiece by cutting stresses or interference with the requisite accuracy and finish.

It is therefore necessary to arrange the cutter speed and the rate of feed to suit each individual job of work. It is essential to know the hardness factor of the material being machined because

upon this depends the optimum speed and rate of feed to give a reasonable cutter life. The expensive nature of the cutter and the time taken to re-grind has a greater effect on production costs than has the cost of the tool used on any other machining operation.

Results of milling tests taken recently on a milling machine fitted with anti-friction metal bearings are given in Table V. The tests were intended to prove the rigidity and capacity of the machine, and for this purpose a higher power motor was used than that usually supplied with the machine.

The machine used was a plain horizontal miller, power being supplied by belt from motor to single pulley drive on the machine.

Leading Dimensions of Machine.

Table 70 ins. by 16½ ins. Longitudinal feed 40 ins. Driving pulley dia. 16 ins. by 4 ins. width. Speed of pulley 525 r.p.m. Maximum cutting feed per minute 15 ins. Spindle nose $5\frac{1}{16}$ ins. dia.

TABLE V.

Material Cut.	Width of Cut.	Depth of Cut.	Feed per Minute.	Cu. Ins. of Metal Cut per Min.	H.P. of Motor.
	ins.	ins.	ins.	ins.	
0·4 Carbon steel 38–40 tons tensile.	4	0·25	7½	7½	8
" "	4	0·25	12½	12½	12½
" "	4	0·420	12½	20½	18–20

Note.—As demonstrating the margin of strength and potential power in well-constructed machines, it should be stated that the usual power of motor supplied with the machine is 7½ to 10 H.P.

Leading dimensions of typical horizontal plain milling machines, production type, are given in Table VI.

TABLE VI.

Work table	51″ × 12¾″	64″ × 14″	70″ × 16½″
Lengthwise feed	30″	40″	40″
Cross feed	10″	12″	12″
Max. height, spindle to table	18½″	20″	18″
Dia. driving pulley	14″	16″	16″
Speed r.p.m. of pulley	525	340	525
No. of spindle speeds	12	12	16
Range of spindle speeds	17 to 450	14 to 340	17 to 450
Cutting feeds and range per min.	12 rates ½″ to 10½″	16 rates ½″ to 148″	16 rates ½″ to 15″
Quick traverse. Length	75″	60″	75″
Cross per min.	56″	60″	56″
Max. dia. of cutter used	12½″	15″	15″
H.P. fixed by nature of work	5 to 7	8 to 10	7 to 9
Type of spindle bearings	Taper roller	Taper roller	Taper roller

See page 1047 for observations on negative rake milling.

SHAPING MACHINES: CRANK TYPE.

The shaping machine still retains its place in the machine shop as a useful productive tool. It is economical in operation, and like the lathe and planer, uses an inexpensive tool. Recent modifications in design have brought it into line with other highly developed machine tools for rapid and accurate production, while for rate of cutting it is capable of using to advantage the best cutting tools. The drive may be by belt or direct by motor. It occupies relatively little floor space for the wide range of work it is capable of dealing with.

Cutting speed changes from single pulley drive are through a gear box, with heat-treated gears, shafts running on ball or roller bearings. Automatic feeds and quick traverse are fitted to the table, which usually can be changed while the machine is running.

Automatic lubrication by means of a pump supplies oil to slides and trunnion bearings.

Control levers are centralised at the workman's side of the table.

Table VII gives relative sizes of machines, power, and floor space occupied, speed changes and cycles per minute obtainable.

TABLE VII.

Work table	17″ × 14″	21″ × 17″	25″ × 18″	30″ × 20″	35″ × 20″
Travel of work table	20″	25″	30″	30″	30″
Speed cycles per min.	13–150	11–120	9–100	7–87	7–87
Speed changes	8	8	8	8	8
Max. stroke	18″	22″	26″	32″	36″
Usual H.P. supplied	5	7½	10	12	12
Floor space	6′ 6″ × 4′ 0″	9′ 0″ × 4′ 6″	10′ 0″ × 5′ 0″	11′ 6″ × 5′ 6″	12′ 0″ × 5′ 6″

The general usefulness of the shaping machine can be increased by fitting a swivelling and tilting table, this being of special advantage in the tool room. Other attachments are, extended tool bar for cutting internal keyways, and mandrel with support, and dividing head for circular work.

HIGH-SPEED DRILLING MACHINES.

In no class of machine tool is there so wide a variety of types and sizes, ranging as they do from super-sensitive air driven machines to the heavy pillar, multiple spindle, and radial drill.

Air driven machines are made for drills from 0·010 in. to $\frac{1}{32}$ in. dia., with speeds of 60,000 to 80,000 r.p.m.; compressed air pressure being 80 lbs. per sq. in.; special hand lever arrangement being fitted to control and ensure the very fine feed necessary.

Small electrically driven machines with direct coupled variable speed motors for drills up to ¼ in. for steel and ⅜ in. for brass run up to 12,000 r.p.m., feed being by ordinary hand lever.

Multi-spindle drilling machines are made with motor drive and speed change gear box. Machines with one up to six or more spindles are made, and can be arranged for both drilling and tapping holes, also reamering, by means of special fixtures adapted for quickly setting in place.

Pillar drilling machines for holes up to 2 ins. or more diameter are made with single pulley and direct motor drive, a good range of speed changes through the gear box being provided for high-speed drills.

Radial drilling machines cover a general variety of work and are easily handled. Principal dimensions and speeds of medium sized machines are given in Table VIII.

TABLE VIII.

Max. swing of spindle	3′ 6″	4′ 0″	4′ 6″	5′ 6″	6′ 0″
Overall length of arm	5′ 3″	5′ 9″	6′ 3″	6′ 9″	7′ 9″
Spindle dia.	1¾″	1¾″	1¾″	1¾″	1¾″
No. of spindle speeds	12	12	12	12	12
Speeds in r.p.m., all sizes	31 57 42 72	100 134	190 262	353 443	615 830
Base plate working surface	3′ 1½″ × 3′ 0″	3′ 7½″ × 3′ 0″	4′ 1½″ × 3′ 0″	4′ 4″ × 3′ 5″	5′ 4″ × 3′ 5″
Weight of machine, not including motors	cwts. 90	cwts. 96	cwts. 130½	cwts. 187½	cwts. 196

Spindle speeds given in Table VIII are common to all the range of radial arm sizes given. The ratio of speeds are arranged to give a nearly uniform cutting speed of 70 ft. per minute for the drill diameters shown on chart below.

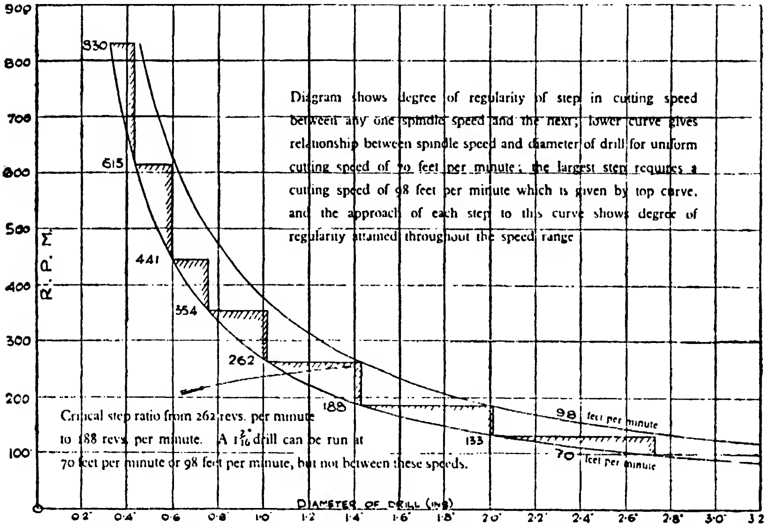


FIG. 4.

Asquith's Radial Drilling Machines.

***Note.*—The power rating for drilling machines is based on the penetrating rate of the drill, also size and speed of the spindle carrying the drill, and not on the size of the machine. Thus the power of motor required to drive a radial drilling machine having 4 ft. radius. may be the same as a machine having 8 ft. radius, if the size of drill spindle is the same in both machines. The extra swing of the arm or radius of spindle making little or no difference to the power absorbed in driving the drill. The length of arm required is purely governed by the dimensions of the work piece to be covered.**

GRINDING MACHINES.

Production grinding machines and precision or finish grinding machines are now practically universally fitted with hydraulic power for table traverse and for in-feed to wheel and quick returns. In some cases both work headstock and tailstock are driven and moved by hydraulic power.

When electrically operated the drive is generally by Vee pulley and belts from the motor to main spindle of machine.

Hydraulic power to the table movement allows infinitely feeds within a wide range, sometimes varying from 6 ins. to 240 ins. travel per minute. The sliding movement of the table is smooth and without shock at the point of reverse.

Production grinding from rough casting or forging to finished surface is economically sound, as shown by the examples below :—

Removing 0·125 in. stock from rough casting and finishing surface of flange 8½ ins. by 7 ins. Actual grinding time 1 min. 28 secs.

Removing 0·0625 in. each side of nitralloy steel plate 6½ ins. by 5 ins. by ⅜ in. thick, grinding 6 plates per loading. Floor to floor time average 6 mins. each.

Grinding C.I. railway axle liners 8 ins. long by 1½ ins. wide, stock removed 0·038 in. each side, floor to floor time 3 pieces per loading = 36 per hour ;

the work being carried out on a vertical spindle grinding machine having hydraulically operated table traverse with a speed range of 1 to 35 ft. per minute. Grinding wheel 14 ins. dia. with 7 segments, driven by 20 H.P. squirrel cage motor running at 1,450 r.p.m. and 4 H.P. motor for the hydraulic system. Coolant pump 0·75 H.P. delivering 26 gallons per minute.

Portable Tools.

Power tools of the portable type most frequently used in engineering workshops are pneumatic, electrical and flexible shaft drive. These are popular because of the ease with which they can be moved and used in places which otherwise are inaccessible to power operated tools.

Pneumatic Tools.

Used for drilling, grinding, chipping and riveting.

For Light Drilling and Grinding.

Air Pressure. Lbs. per Sq. In.	Air Consumption. Cu. Ft. Free Air per Min.	Speed. R.P.M.	Capacity.	Weight.	Used for	Type.
100	4	Up to 2,400	Drill $\frac{5}{16}$" holes in M.S. $\frac{1}{4}$" in brass	1 lb. 15 oz.	Drilling.	Spiral piston.
100	4	700 + 400 with reduction gears	Drill $\frac{5}{16}$" holes in M.S. $\frac{3}{8}$" in brass	2 lb. 6 oz.	,,	,,
30	$3\frac{1}{4}$	30,000	$\frac{3}{8}$" to 1" wheel	13 oz.	Grinding	Turbine
40	$3\frac{3}{4}$	40,000	$\frac{3}{8}$" to 1" ,,	13 oz.	,,	,,
50	$4\frac{1}{4}$	45,000	$\frac{3}{8}$" to 1" ,,	13 oz.	,,	,,
60	$4\frac{3}{4}$	50,000	$\frac{3}{8}$" to 1" ,,	13 oz.	,,	,,

For Light Riveting.

Air Pressure. Lbs. per Sq. In.	Air Consumption. Cu. Ft. Free Air. per Min.	Size of Rivets.		Weight.
		Alumin.	Steel.	lbs.
80	8	$\frac{1}{8}$"	—	$4\frac{3}{4}$
80	9	$\frac{3}{16}$"	$\frac{1}{8}$"	$5\frac{1}{4}$
80	10	$\frac{1}{4}$"	$\frac{3}{16}$"	$5\frac{1}{2}$
80	14	$\frac{5}{16}$"	$\frac{1}{4}$"	6

For Heavy Riveting.

Air Pressure. Lbs. per Sq. In.	Air Consumption. Cu. Ft. Free Air per Min.	Size of Rivets.	Weight.
			lbs.
80 to 100	22	$\frac{7}{8}$"	19
80 to 100	23	1"	20
80 to 100	24	$1\frac{1}{8}$"	22
80 to 100	26	$1\frac{1}{4}$'	$22\frac{1}{2}$

Heavy Portable Universal Radial Drilling and Tapping Machines.

Max. Radius of Drill Spindle.	Vertical Height from Floor to Spindle. Max.	Vertical Height from Floor to Spindle. Min.	Horizontal Height from Floor to Spindle. Max.	Horizontal Height from Floor to Spindle. Min.	Spindle Dia.	Morse Taper in Spindle.	Motor H.P.	Weight of Machine with Bogie.
						No.		Cwts.
3' 9"	5' 6"	2' 0"	7' 0"	1' 6"	2"	4	3	49
4 9	5 0	2 0	7 4	1 0	2	4	3	61
5 9	6 0	2 0	8 6	1 0	2	4	3	88

Spindle speeds for each machine range from 41 to 300 r.p.m.

For Medium Drilling.

Air Pressure. Lbs per Sq. In.	Air Consumption. Cu. Ft. Free Air per Min.	Size of Drill.	Weight.
			lbs.
80 to 100	25	$\frac{5}{8}$"	18
80 to 100	30	1"	24
80 to 100	50	$1\frac{1}{4}$"	35
80 to 100	60	$1\frac{1}{2}$"	37

Portable Electric Drills.

Voltage	No Load Speed. R.P.M.	Capacity in Steel.	Capacity in Hard Wood.	How Used.	Machine Net Weight.
		ins.	Ins.		lb.
220–250	2,000	$\frac{1}{4}$	$\frac{1}{2}$	On stand or hand	$5\frac{1}{2}$
220–250	1,400	$\frac{5}{16}$	$\frac{5}{8}$	” ”	$7\frac{3}{4}$
220–250	900	$\frac{3}{8}$	$\frac{3}{4}$	” ”	$12\frac{3}{4}$
220–250	400	$\frac{1}{2}$	1	Stand or post	$13\frac{1}{4}$
		In Steel. Ins.			
220–250	500	$\frac{5}{32}$ to $\frac{5}{8}$		” ”	$17\frac{1}{2}$
220–250	375	$\frac{7}{8}$		” ”	$26\frac{1}{2}$
220–250	250	$1\frac{1}{4}$		” Stand ”	66

Portable Electric Driving through Flexible Shaft.

For Drilling, Grinding and Boiler-tube Cleaning.

Power of Motor.	Speed. R.P.M.	Flexible Shaft Length and Core Dia.	Capacity.
D.C. ⅛ H.P. A.C. ¼ „	2,000 2,800	6′ long, ½″ core dia.	With pistol grip drills up to ½″ dia. on stand with screw feed up to ¾″
D.C. ⅛ „ A.C. ¼ „	2,000 2,800	„ „	Carries 6-in. diam. polishing mop.
D.C. A.C. } 1½ „	2,800	—	Driving double-ended polishing mop.

Flexible shafts are also used for boiler-tube cleaning, ranging from ½-in. dia. to 4-in. dia. tubes.

Units are supplied with motor or engine power mounted on pedestal or runabout trolleys.

Horse-Power Required for Modern Machine Tools.

The power units given in these tables are those generally supplied with the machines as standard. For special purposes the power may be greater or less, according to the use to which the machine is put. Manufacturers should be consulted on questions of power.

Combined Slotting and Cross Planing Machine.

Length of Slotting Stroke.	Distance between Up-rights.	Max. Height admitted under Slide.	Stroke for Cross Planing.	Cutting Speeds. F.P.M.	Return Speeds. F.P.M.	Extreme Height of Machine.	Driving Motor Slotting.	Driving Motor Planing.	Approx. Weight without Motors.
4′ 0″	8′ 3″	4′ 1″	7′ 0″	15 to 50	up to 75	19′ 6½″	25 H.P.	25 H.P.	58 tons
6 0	8 3	4 10	7 0	12 to 50	up to 60	24 6	25 H.P.	25 H.P.	65 tons

High Speed Lathes. S.S. and S.

Swing over Bed.	Standard H.P.	Swing over Bed.	Standard H.P.
Ins.		Ins.	
9	3½	24	20
12	5	28	28
16	7½	36	50
20	2	Heavy type 64	60/150

High Speed Planing Machines.

Width × Height × Travel.	Standard H.P.	Width × Height × Travel.	Standard H.P.
3′ × 3′ × 8′	30	4′ 6″ × 4′ 6″ × 12′	40
3′ 6″ × 3′ 6″ × 10′	30	5′ × 5′ × 12′	45
4′ × 4′ × 10′	35	5′ ″6 × 5′ 6″ 12′	50
4′ × 4′ × 12′	40	6′ 0″ × 6′ 12′	50

12′ 6′ × 10′ × 30′	Main motor	80
	Aux. for traverse	10

Single Head High Speed Milling Machines.

Table Length and Width.	Length of Feeds. Length × Cross.	Standard H.P.
Ins.	Ins.	
51 × 12½	30 × 10	5
64 × 14	40 × 12	8
70 × 16	40 × 12	8–10

Double Head High Speed Milling Machines.

Table Length × Width.	H.P.
Ins.	
40 × 34	5
45 × 54	10
67 × 54	20

Single Head Shaping Machines.

Maximum Stroke.	Standard H.P.
Ins.	
18	5
22	7½
26	10
36	12

Single Head Traverse Shapers.

Maximum Stroke.	Standard H.P.
Ins.	
14	7½
20	12
26	15
32	20

High Speed Slotting Machines.

Stroke.	Standard H.P.
Ins.	
8	4
10	5
14	12
20	15
28	20

Capstan Lathes Rated by Size of Hole in Spindle.

Light Duty.	H.P.	Heavy Duty.	H.P.
Ins.		Ins.	
1⅝	3	1⅝	4
2⅛	5	2⅛	7½
2⅝	6	2⅝	9

Auto-Lathes.			Heavy Combination Turret Lathe.		
Maximum Swing.	Maximum Stroke.	H.P.	Maximum Swing.	Maximum Travel of Turret.	H.P.
Ins.	Ins.		Ins.	Ins.	
19	10	7	33	115	40
23	12	10			

Locomotive and Tramway Wheel Lathes.		
Locomotive Wheels.	Number of Tool Rests.	H.P.
Turned when mounted on their axle. 7′ 6″ dia.	Front rests for turning wheels 2	Main driving motor 35
Carriage and Waggon Wheel Lathe.		
Turned when mounted on their axle from 2′ 9″ dia. to 3′ 9″ dia.	Number of tool rests for turning wheels 2	H.P. of two-speed A.C. motor 20/40
Tramway Wheel Lathe. Heavy Duty.		
Turned when mounted on their axle from 1′ 9½″ to 2′ 9½″ overall dia.	Number of tool rests in operation at same time 2	H.P. of main driving motor 25/50

High Speed Surfacing and Boring Lathes.			Lathes with Sliding Bed.		
Swing over Bed.	Admits in Front of Chuck.	Standard H.P.	Normal Height of Centres.	Max. dia. Swung in Gap.	Standard H.P.
Ins.	Ins.		Ins.		
20	20	7·5	14½	5′ 0″	20
26	26	12·5	18½	6′ 0″	25
33	33	20			
42	42	28			

Production Grinding Machines. Hydraulic Cylindrical Grinding.

Max. Dia. and Length of Work.	Max. Dia. and Width of Wheel.	Main Drive H.P.	Hydraulic Feed H.P.
Ins.	Ins.		
6 dia. 24 long	20 × 2	5 to 7½	1½
10 dia. 24 long	24 × 8	20	2
14 dia. 40 long	24 × 8	25	3

Production Grinding Machines. Rotary Surface Grinding Machines with Hydraulic Feed to Head.

Grinding Lengths.	Grinding Widths.	Grinding Heights.	Wheel. Dia.	H.P. to Grinding Wheel.	H.P. Hydraulic Feed.
Ins.	Ins.	Ins.	Ins.		
48	12	12	14	20	4
60	18	18	24	40	4
72	24	18	27	50	5

Vertical Drilling Machines.

Type.	Spindle Dia. at Drive Point.	Capacity in Dia.	H.P.
	Ins.	Ins.	
Sensitive	11/16	½	1
”	13/16	11/16	1¼
”	1	⅞	1½

Multiple Spindle Vertical Drilling.

Size of Drill.	Penetration per Min.	No. of Drills Carried.	H.P.
Ins.	Ins.		
9/16	3¼	8	15
⅞	2½	8	15
1¼	1¼	8	25

Note.—Many types of multiple spindle drilling and boring machines are made having a wide range of adjustable pitch circles. These are regarded as special purpose machines.

Vertical and Radial Single Spindle Drills.

Spindle Dia. at Drive Point.	Size of Drill Dia. up to:	Morse Taper No.	H.P.
Ins.			
1¼	1" M.S. 1¼" C.I.	3	2
1½	1¼" M.S. 2" C.I.	4	3
1¾	1½" M.S. 2¼" C.I.	5	4
2¼	2¼" M.S. 2¾" C.I.	5	7½
3½	3¼" M.S. 3½" C.I.	6	12½

Single Spindle Honing Machine

Max. Dia. of Hone.	Min. Dia. of Hone.	Length of Stroke.	H.P.
Ins.	Ins.	Ins.	
8½	2	36	7½

Production Grinding Machines with Hydraulic Feed Motion.
Vertical Spindle Surface Grinders.

Grinding Length.	Grinding Width.	Grinding Height.	Grinding Wheel Dia.	H.P. Grinding Wheel Motor.	H.P. Hydraulic System.
Ins.	Ins.	Ins.	Ins.		
48	12	12	14	20	4
60	18	18	24	40	4
72	24	18	27	50	5

METAL COLD SAWING. A.W. HIGH-SPEED STEEL.

Inserted-Tooth Saw.	Material.	Cutting Speed.	Time to Cut Through.
Diam. in Ins.		Ft. per Min.	Mins.
36	Steel (joist, 20 ins. deep by 7½ ins. wide; ¾ in. web)	122	17
36	Steel (rail, 100 lbs. per yard)	90	2
36	Forged steel (bar 3 ins. diam.)	90	2½
36	Forged steel (bar 10 ins. diam.)	90	11
36	Forged steel (bar 7½ ins. by 9 ins.)	84	9

MISCELLANEOUS.	H.P.
Double punch, running light	2·0
Punching both ends 1 in. hole, ¾ in. plate, 28 punches per minute at each end	5·0
Punch and shear, running light	2·0
Punching hole as above, and shearing ¾ in. plate, 5½ in. cut, 28 strokes per minute	6·5
Large plate-bending rolls, running light	5·5
Rolling plate $\frac{9}{16}$ in. thick, 4 ft. 4 ins. wide, and 16 ft. 6 ins. long, endways on (4 ft. 4 ins.)	6·9
Sideways on (16 ft. 6 ins.)	12·2
Plate $\frac{11}{16}$ in. by 4 ft. 8½ ins. by 21 ft. long, sideways on	19·3
Same rolls lifting and lowering top roll: Lifting	8·5
Lowering	7·0
Forcing $\frac{11}{16}$ in. plate down	10·0
Forge fan, for 24 fires	10·5
Angle squeezer, running light	0·8
Squeezing	2·5
Riveting machine, running light	1·6
Riveting	3·0
Plate planing machine, running light	1
Reversing light	2
Cutting	5
Stern frame, boring, cutting	1·75
Angle-cutting machine, running light	1·6
Cutting angle 6 ins. by 3½ ins. by ½ in.	3·5
Winch, lifting 28 cwt. single purchase, 50 ft. per minute	6·5
Ending machine, running light	1·9
Cutting girder 18 ins. by 7 ins.	4 to 4·8
Cold saw, running light, 24 in. saw	2·0
Cutting 4 in. by 4 in. angle	4 to 4·5

MISCELLANEOUS *Continued*)	H.P.
Air compressor, 10 ins. by 14 ins. cylinder, 50 lbs. pressure	15 to 23·5
Hydraulic pumps, three pumps, 3¾ ins. diameter; 4¾ ins. stroke, 60 strokes per minute; pressure 870 lbs.	38·0
Electric winch, running light	1·2
Lifting 24 cwt. 16 ft. per minute	5·6
Vertical cylinder boring machine (marine), 14 ins. diam. bar	15·0
" " " 18 "	20·0
Bevel gear planer to take wheels up to 12 ins. diam.	2·0
" " " 24 "	4·0
" " " 36 "	5·0
Sunderland gear planers to cut up to 3 D.P. (No. 6)	7·0
" " " 1½ D.P. (No. 9)	10·0
" " 1¼ D.P. (No. 11)	15·0

Wood-Working Machinery.

Improvements in the design of wood-working machinery as used in engineers' pattern shops and general wood-working factories, has kept pace with modern requirements. Full advantage has been taken of electric driving, both direct coupled and also operating through pulleys and belts for power transmission.

The high speeds at which this class of machinery has to run to obtain the best results makes the electric drive desirable from several points of view. In the pattern shop in particular, where the machine should always be placed in the position most convenient to the pattern maker, the electric drive becomes practically essential, moreover in some pattern shops it often happens that one machine is used in turn by several men, and in that case the machine should be placed in that position where waste of time in reaching the machine will be avoided. The electric drive makes it possible to always choose the right position.

Practically all the best machines are now fitted with dust-proof heavy type ball bearings, rigidly supported in well-designed housings to prevent vibration, a feature so necessary to any form of quick and accurate machining.

The introduction of the 'Frequency Changer' in electrical plant allows for higher r.p.m. than can usually be obtained from standard squirrel-cage motors. Experience shows that the necessary apparatus for high-frequency operation is reliable and occupies but little floor space.

But probably for the heavy types of wood-working machinery, where the r.p.m. required exceeds that of the motor, it is mechanically sound to use vee pulley and belts, properly proportioned, and placed at conveniently close centres, to reach the speed desired.

Machinery for Engineers' Pattern Shop.

The semi-auto pattern miller is fitted with self-contained electric drive, with contactor control. All feed motions are positive and there are index scales and positive stops. The table has universal movements, and the overhanging arm carrying the spindles and cutter heads can be raised or lowered by hand or power.

Wadkin Semi-Auto Pattern Millers.

Cutter Centre to Body Frame.	Max. Height Table to Cutter.	Spindle Speeds R.P.M.	H.P. for Cutter Spindle.	Foor Space.	Net Weight.
′ ″	″			′ ″	Cwts.
4 0	Vert. 19½ Hor. 24	1,390 to 4,140	5	10 9× 10 9	70
2 4	Vert. 14½ Hor. 18	2,700 to 4,000	4	8 0× 6 4	35

Motor driven cross-cutting and trenching machine, suitable for cross-cutting timber, straight or angular, or for grooving and trenching. The saw carriage is pulled forward by hand for cutting, and returned by spring action, being air cushioned at the end of return stroke.

Electric Cross-Cutting and Trenching Machine.

Saw Dia.	Size of Timber Out.	Groves Out.	Motor H.P.	Net Weight.
″	″ ″	″ ″		Cwts.
18	28 × 4½ 33 × 1	2½ × 1⅝ deep × 25½ wide	5	9½
18	20 × 4½ 25 × 1	2½ × 1⅝ deep × 17¾ wide	5	8¾

Circular saw benches may be driven by direct coupled motor to saw spindle, or from motor to spindle by vee belts. Also made for belt drive with fast and loose pulley, the loose pulley being mounted on sleeve.

Tables may be fixed, or fitted with hand-operated screw for raising and lowering.

Circular Saw Benches.

R.P.M.	Dia. of Saw.	Max. Depth of Cut.	Size of Table.	Rise and Fall of Table.	Max. Dist. Between Saw and Fence.	Floor Space with Vee-Belt Drive.	Motor H.P.	Net Weight with Vee-Belt Drive.
	″	″		″	″			Cwts.
1,600	26	9	4′ 8″ × 2′ 6″	6	15½	6′ 9″ × 3′ 4″	9	15½
1,300	32	12	5′ 2″ × 2′ 6″	6	15½	7′ 0″ × 3′ 4″	12	17¾
1,100	36	14	5′ 6″ × 2′ 6″	6	15½	7′ 3″ × 3′ 4″	15	20

Surface planing and thicknessing machines are made for driving by motor with vee pulley and belts, or ordinary belt drive from countershaft having fast and loose pulleys. The thicknessing table is raised or lowered by hand-operated gearing, and is supported and guided with vertical slides.

Surfacing and Thicknessing Machines.								Electric Drive.	
Machine Size.	Thick-nessing Capacity.	Surfacing Capacity.	Surface Table Length.	Cutter Block. R.P.M.	Feed. Ft. per Minute.	Main Drive. H.P.	Feed Drive. H.P.	Floor Space.	Net Weight.
″	″	″							Cwts.
20	20 × 9	22	6′ 1″	4,200	20, 30 and 46	5	1	6′ 1″ × 5 0	27½
24	24 × 9	26	6 1	4,200	20, 30 and 46	5	1	6 1 × 5 4	28¾

Vertical spindle moulders and shapers are available with either one or two spindles to each machine; they may be driven by belt from countershaft, by motor through vee pulley and belt, or by high-frequency electric motors. In the latter case the motor is usually built directly on the cutter spindle, having B.T.H. rotor and stator, especially designed for high rotary speeds up to 10,000 R.P.M. and are suitable for use on high grade work.

Vertical Spindle Moulders and Shapers (Belt and Countershaft Drive).

	Size of Table.	Capacity in Depth.	Fence Plates.	Between Spindles.	C/S Speed R.P.M.	F. & L. Pulley.	H.P.	Net Weignt.
	′ ″ ′ ″	″	″ ″	′ ″		′ ″	″	Cwts.
Single spindle	2 8 × 2 8	6	6 × 18	—	850	8 × 3½	3	11
Double spindle	5 0 × 3 0	6	6 × 18	2 6	850	10 × 4½	6	18
Double spindle	6 0 × 3 0	6	6 × 18	3 6	850	10 × 4½	6	23

Vertical Spindle Moulders and Shapers (Motor and Vee Pulley Drive).

Single spindle	3 0 × 2 8	6	6 × 18	—	Spindle R.P.M. up to 6,000	—	4	12
Double spindle	5 0 × 3 0	6	6 × 18	2 6	6,000	—	each spindle 4	21
Double spindle	6 6 × 3 0	6	6 × 18	3 6	6,000	—	4	23½

Vertical Spindle Moulders and Shapers (High Frequency Electric Spindle).

Single spindle	3 0 × 2 8	6	6 × 18	—	Spindle R.P.M. up to 10,000	—	4	11
Double spindle	5 6 × 3 0	6	6 × 18	2 6	10,000	—	each spindle 4	22

Band sawing machines, with stiffened frames, improved guiding, admitting of higher cutting speeds, compete favourably with other forms of hard- and soft-wood cutting. When adapted for motor drive, the motor is fitted close up to the bottom wheel, and is enclosed in the body of the machine.

Band Sawing Machines.

Diam. of Saw Wheel.	Width of Saw.	Length of Saw.	Depth between Guide and Table.	Speeds up to	H.P.	Floor Space.	Net Weight. Cwts.
″	″	′ ″	″	R.P.M.		′ ″	
30	1¼	17 0	14	1,000	3	4 8 × 2 10	12½
36	1½	20 0	18½	1,000	4	5 6 × 3 2	17½

Vertical and horizontal boring machines, driven by belt or direct motor drive are useful tools, in the pattern shop. Arranged for either hand or foot feed.

Boring Machines. Vertical and Horizontal Types.

Capacity.	Chuck for Tools.	Spindle Speeds. R.P.M.	Size of Table.	H.P.	Floor Space.	Net Weight. Cwts.
Will bore 1″ diam., 6″ deep	Up to ½″ shanks	2,800	20″ × 15″	1½	20″ × 31″	8½

Wood turning lathes have been improved in design by the introduction of the electric motor drive, the entire absence of outside or overhead belts makes for a clean-cut compact tool that can be placed in any part of the pattern shop or factory. The simplest, and probably the most reliable, form of motor drive as applied to wood turning lathes, has the motor built into the head-stock frame at the floor level, the extended end of the motor shaft carries a 3- or 4-step cone pulley and drives by an ordinary flat belt to a reversed step cone pulley mounted on the head-stock main spindle, the belt drive is visible and operatable through openings cast in frame opposite the top and bottom step pulleys. The headstock spindle runs in dust-proof ball bearings, and is screwed at both ends to take face plates.

Another form of motor drive is through vee pulleys and belts direct to all gear headstocks, this form being generally used for the heavier type of lathe with sliding bed to allow for maximum swing in the gap.

Some small wood turning lathes have the motor built into headstock, the shaft of the motor forming the headstock spindle.

The ordinary lathe with open step cone pulley headstock driven from the familiar counter-shaft is still made and serves a useful purpose in many of the older pattern shops.

Wood Turning Lathes.

Type.	Speeds. R.P.M.	Height of Centres.	Length between Centres.	Diam. turned with Gap.	Width in Gap.	Diam. turned over Rests.	H.P.	Floor Space.	Weight. Cwts.
		″	′ ″	′ ″	′ ″	″		′ ″	
Motor step cone drive	1,500 + 3,000 c/s 500	6¼	3 3½	2 0½	8½	9	1½	6 9 × 2 3	11½
Motor step cone drive . .	c/s 500	8¼	5 3½	2 4½	8½	13	1½	8 9 × 2 3	12½
Motor step cone drive . .	c/s 500	10¼	7 3½	2 8½	8½	17	1½	10 9 × 2 3	13½
Motor, vee belts and gear head . .	93 to 1,904	12¼	8 3	5 6	2 6	19	3	14 6 × 3 3	22
Sliding bed . .	98 to 1,904	15¼	8 3	6 0	2 6	25	3	14 6 × 3 3	23
Step cone headstock	c/s 500	7	1 10	1 9	5	10	2	4 0 × 2 0	6
,, ,,	c/s. 500	9	3 0	2 1	5	11½	3	6 0 × 2 4	8¾
,, ,,	c/s 500	11	3 0	2 3	5	13½	4	6 0 × 2 4	9½

Sander Machines.—Vertical bobbing type and the disc type should always be fitted with dust-collecting hoods and ducts. Canting tables are fitted for taper work, or draft on patterns.

Vertical Bobbing Sander. Direct Motor or O/S Drive.

Max. Depth Sanded.	Bobbin Sizes Supplied.	Bobbin Max. Size Allowed.	Bobbin Min. Size Allowed.	Size of Table.	Motor Drive. H.P.	Counter-shaft Drive. R.P.M.	Floor Space.
8″	3½″ Diam. 9½″ Long	5″	2″	26″ × 25½″	1	500	26″ × 27″

Double-Disc Type Sander Machine. Direct Motor or O/S Drive.						
Disc Diam.	Size of Tables.	H.P.	O/S R.P.M.	Floor Space.	Weight Motor Drive.	Weight O/S Drive.
30	2′ 10″ × 1′ 4″	4	500	2′ 10″ × 5′ 4″	16 cwts.	11 cwts.
36″	3′ 4″ × 1′ 7″	5	450	3′ 4″ × 5′8″	17½ „	14½ „

Grinding and honing machines for pattern makers' and woodworkers' tools have received practical consideration, and special types of machines for the purpose have been evolved.

Powerful planing and moulding machines are made with four or five cutter heads for floor boards, matchings, mouldings, etc. A wide range of machines are available with motor drive direct or from countershaft.

4 and 5 Head Planing and Moulding Machines (Motor and Belt Direct Drive).									
	Max. Size Boards.	Cutter Head. R.P.M.	Top and Bott. Heads. H.P.	Side Heads. H.P.	Feed H.P.	Top Rollers Diam.	Feed in Feet. Per Min.	Machine Overall Length.	Net Weight. Tons.
4 Head	12″ × 4″	6,000	15	7½	7½	8″	30 to 150	7′ 6″	4½
5 Head	12 × 4	6,000	15	7½	7½	8	30 to 150	11 0	5
4 Head	7 × 4	6,000	10	7½	5	8	30 to 145	9 3	3½
5 Head	7 × 4	6,000	10	7½	5	8	30 to 145	10 8	4½
4 Head	4 × 4	7,500	6	6	6	5	30 to 90	6 9½	T. C. 2 — 1½

4 and 5 Head Planing and Moulding Machines (Motor to Countershaft Drive).							
	Max. Size Boards.	Countershaft Pulleys.			Main Drive. H.P.	Weight with Countershaft. Cwts.	Floor Space.
		Diam.	Width.	R.P.M.			
4 Head	16″ × 6″	16″	16″	1,300	40	172	16′ 8″ × 5 8
5 Head	16 × 6	16	16	1,300	45	178	16 8 × 5 8
4 Head	10½ × 3	14	11	1,050	20	82	12 6 × 6 6
5 Head	10½ × 3½	14	11	1,050	25	92	12 6 × 6 6
4 Head	3 × ½	12	9½	760	6	32	7 4 × 4 8

All Electric Driven Finger Feed Planer.

Width Planed.	Overall Length of Table.	Feed. Feet per Min.	Cutter Block Speed. R.P.M.	Motor to Cutter Block. H.P.	Motor to Feed. H.P.	Floor Space.	Weight. Cwts.
2′ 0″	11′ 0″	25, 39, 58	4,000	7½	2	11′ 0″ × 5 2	54

9″, 12″, 16″ Surface Planers and Jointers. For Surfacing Timber, making Glue Joints, Rebating, Bevelling, Tonguing and Grooving. Motor Drive with Vee Pulley and Belts, or from Countershaft with Flat Belts.

Max. Depth of Cut.	Max. Depth of Rebate.	Length of Table for 12″ 12″ Machine.	Length of Table for 9′ Machine.	Cutter Block. R.P.M.	H.P.			Floor Space.		
					9″	12″	16″	9″	12″	16″
1″	½″	6′ 0″	5′ 0″	4,000	2	3	4	4′ 6″ × 2 9	6′ 0″ × 2 9	6′ 0″ × 3 0

Electric High Speed Router.

Distance Spindle to Frame.	Size of Table.	Rise and Fall of Table.	Spindle Speeds.	Motor H.P.	Floor Space.	Net Weight. Cwts.
24½″	30″ × 30″	9″	18,000 to 24,000	1	4′ 0″ × 2 6	12½

Chain and Chisel Mortiser, with Adjustable Stops for Depth.

Takes Timber	Size of Mortise using Chain.	Size of Chisel for Hard Wood.	Size of Chisel for Soft Wood.	Motor H.P.	Floor Space.	Net Weight. Cwts.
11 × 8 or 2 × 8	1″ × 2½″ × 6″ 1½ × 3 × 6	¾″ ¾	1 1	4 4	4′ 5″ × 3′ 6″ 4 8 × 4 3	10½ 11

Pendulum Cross-Cut Saws. Motor and Belt Drive.					
Size.	Diam. of Saw.	Cross-Cut Timber.	Centres from Beam to Saw.	Motor H.P.	Net Weight. Cwts.
24″	24″	14″ × 7″	6′ 10″	4	9
32	32	24 × 9	6 10	5	10½

Fret-Sawing Machines, with Under-Drive Countershafts.							
Stroke of Saw.	Size of Table.	Fast and Loose Pulleys on Machine.		Speed. R.P.M.	Power. H.P.	Floor Space.	Net Weight. Cwts.
		Diam.	Width.				
1½″	28″ × 24″	5″	3″	900	1	30″ × 36″	5½
3	40 × 28	6	4	450	2	44 × 60	9

Fret-Sawing Machine. Self-Contained Motor Drive.					
Distance from Saw to Column.	Height from Table to Head.	Max. Depth of Cut.	Size of Table.	Floor Space.	Net Weight. Cwts.
31″	7″	6″	30″ × 30″	4′ 3″ × 2′ 6″	9½

Rotary Veneer Peeling Machines.							
For Logs up to	Veneers up to Thick.	Fast and Loose Pulleys on Countershaft.		Speed. R.P.M.	Motor. H.P.	Floor Space.	Weight. Cwts.
		Diam.	Width.				
24″ diam. 24″ long.	½″	36″	8″	100	6	8′ 4″ × 4′ 8″	33½
28″ diam. 52″ long	½	18	9½	220	15	16 0 × 4 4	94
36″ diam. 78″ long	½	26	16	250	21	16 8 × 4 4	203

Force Required to Punch Holes and Cut Plates.

If W = maximum orce in lbs. required to punch hole,

t = thickness of plate n ins.,

then

$$W = 190{,}000dt$$

for structural steel, soft iron, boiler and stee plate.

As W varies as dt, the same machine and power would punch a $\frac{3}{4}$-in. hole in a $\frac{3}{4}$-in. plate, as would punch a $2\frac{1}{4}$-in. hole in a $\frac{1}{4}$-in. plate, since $\frac{3}{4} \times \frac{3}{4} = 2\frac{1}{4} \times \frac{1}{4}$.

If F = force required to shear plates with straight knives inclined at an angle $\alpha°$ with each other, then

$$F = 30{,}000\, t^2 \cot \alpha°.$$

An approximate rule for finding the force required to cut metal by punching or shearing is

Force = 80 per cent. of the tensile strength of a bar having a cross-sectional area equal to the area of the metal cut.

The side thrust on a shearing blade when it is dull and the plate is tipped may be taken as equal to $1 \cdot 5$ W, when W is the force acting normal to the plate.

Flat punches should always be used when the thickness of the plate is greater than two-thirds the diameter of the hole. If the thickness is less than half the hole diameter, much less force is required when a shearing punch is used. Shearing punches are also recommended for cases where a large number of holes are to be punched close together, because they do not distort the plate as much as flat-ended punches.

The diameter of a punched hole is equal to the diameter of the punch, but if there is an appreciable clearance between the punch and the die, the hole will be conical.

Small Tools for Use on Machine Tools.

Milling Cutters.

Standard milling cutters made from the best high-speed steels are supplied by several well-known makers ; there are also many types of built-up milling heads carrying various brands of alloy cutting tools for maximum cutting speeds.

The size and shape of the milling cutter is entirely dependent on the class of work for which it is to be used. As a general rule, in determining the correct angle for the face of the teeth, the diameter and shape of the cutter should be taken into account. For small-sized cutters it will, in order to obtain a strong shape, generally be found best to make the faces truly radial, but for large cutters the teeth may with advantage be undercut 8° to 12° (figs. 5 and 6). This also is

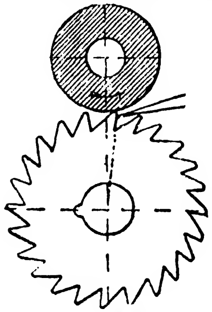


FIG. 5

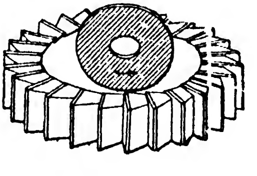


FIG. 6.

more convenient when the cutter is to have teeth both on the sides and periphery. Cutters with the teeth so formed have worked upon both wrought and cast iron perfectly well, and at the same time they are not so liable to seize hold and bite into the work as cutters which have the faces of their teeth too much undercut. The pitch of the teeth is, again, partly determined by the diameter and class of cutter. In those cutters which are cut at the sides, the pitch at the circumference has often to be made wider than is advisable, in order to give clearance room for the emery wheel when grinding the converging side teeth (see figs. 5 and 6). A pitch from $\frac{5}{16}$ inch to $\frac{3}{8}$ inch will usually be found to give good results. In all except very narrow cutters it is advisable to cut the teeth on the periphery spirally. This ensures a smoother action of the machine and produces better work. In deciding whether to make the cutters right- or left-hand spirals, attention should be paid to the direction in which they are to revolve. For example, in a vertical machine whose spindle revolves to the right hand, as in an ordinary drilling machine, the cutters should be cut with a left-hand spiral. This has the effect of forcing the work down on to the table and the spindle up against its end bearings, as the cutter and the work act as a screw and a nut.

No hard and fast rule can be laid down for cutting speeds and feeds, this being mainly dependent on the nature of the metal to be cut, depth of cut, the amount of metal to be removed, and the power available on the machine. Cutting tests have been taken showing that 20½ cu. ins. of 0·4 carbon steel has been removed per minute with a 4-in. length of milling cutter, absorbing 20 H.P.

But for ordinary work, with cutters made from good class high-speed steels, the cutting speeds for heavy cutting can be taken at 50 to 60 ft. per minute, and for light cutting at 65 to 70 ft. per minute. For cutting mild steel and cast iron these speeds can be taken as representing good practice.

A point to remember is that for heavy milling work a coarse pitched cutter gives more satisfactory results, also it is better to grind frequently to keep a keen edge than to chance ruining a cutter by allowing it to run after the edges have become dulled.

Reference table of cutting speeds obtained from cutters of stated diameters and revolutions per minute is given in Table IX.

TABLE IX.

Feet per Minute	15′	20′	25′	30′	35′	40′	45′	50′	60′	70′	80′
Diam.	REVOLUTIONS PER MINUTE.										
1/16	917·0	1223·0	1528·0	1834·0	2140·0	2445·0	2751·0	3057·0	3668·0	4280·0	4891·0
1/8	459·0	611·0	764·0	917·0	1070·0	1222·0	1375·0	1528·0	1834·0	2139·0	2445·0
3/16	306·0	408·0	509·0	611·0	713·0	815·0	917·0	1019·0	1222·0	1426·0	1630·0
1/4	229·0	306·0	382·0	458·0	535·0	611·0	688·0	764·0	917·0	1070·0	1222·0
5/16	183·0	245·0	306·0	367·0	428·0	489·0	550·0	611·0	733·0	856·0	978·0
3/8	153·0	204·0	255·0	306·0	357·0	408·0	458·0	509·0	611·0	713·0	815·0
7/16	131·0	175·0	218·0	262·0	306·0	349·0	393·0	437·0	524·0	611·0	699·0
1/2	115·0	153·0	191·0	229·0	268·0	306·0	344·0	382·0	459·0	535·0	611·0
5/8	91·8	123·0	153·0	184·0	214·0	245·0	276·0	306·0	367·0	428·0	489·0
3/4	76·3	102·0	127·0	153·0	178·0	203·0	229·0	254·0	306·0	357·0	408·0
7/8	65·5	87·3	109·0	131·0	153·0	175·0	196·0	219·0	262·0	306·0	349·0
1	57·3	76·4	95·5	115·0	134·0	153·0	172·0	191·0	229·0	267·0	306·0
1 1/8	51·0	68·0	85·0	102·0	119·0	136·0	153·0	170·0	204·0	238·0	272·0
1 1/4	45·8	61·2	76·3	91·8	107·0	123·0	137·0	153·0	183·0	214·0	245·0
1 3/8	41·7	55·6	69·5	83·3	97·2	111·0	125·0	139·0	167·0	195·0	222·0
1 1/2	38·2	50·8	63·7	76·3	89·2	102·0	115·0	127·0	153·0	178·0	204·0
1 5/8	35·0	47·0	58·8	70·5	82·2	93·9	106·0	117·0	141·0	165·0	188·0
1 3/4	32·7	43·6	54·5	65·5	76·4	87·3	98·2	109·0	131·0	153·0	175·0
1 7/8	30·6	40·7	50·9	61·1	71·3	81·5	91·9	102·0	122·0	143·0	163·0
2	28·7	38·2	47·8	57·3	66·9	76·4	86·0	95·5	115·0	134·0	153·0

TABLE IX (*continued*).

Feet per Minute	15′	20′	25′	30′	35′	40′	45′	50′	60′	70′	80′
Diam.	REVOLUTIONS PER MINUTE.										
2¼	25·4	34·0	42·4	51·0	59·4	68·0	76·2	85·0	102·0	119·0	136·0
2½	22·9	30·6	38·2	45·8	53·5	61·2	68·8	76·3	91·7	107·0	122·0
2¾	20·8	27·8	34·7	41·7	48·6	55·6	62·5	69·5	83·4	97·2	111·0
3	19·1	25·5	31·8	38·2	44·6	51·0	57·3	63·7	76·4	89·1	102·0
3½	16·4	21·8	27·3	32·7	38·2	43·6	49·1	54·5	65·5	76·4	87·4
4	14·3	19·1	23·9	28·7	33·4	38·2	43·0	47·8	57·3	66·9	76·4
4½	12·7	16·9	21·2	25·4	29·6	34·0	38·2	42·4	51·0	59·4	67·9
5	11·5	15·3	19·1	22·9	26·7	30·6	34·4	36·2	45·9	53·5	61·1
5½	10·4	13·9	17·4	20·8	24·3	27·8	31·3	34·8	41·7	48·6	55·6
6	9·6	12·7	15·9	19·1	22·3	25·5	28·7	31·8	38·2	44·6	51·0
7	8·1	10·9	13·6	16·4	10·1	21·8	24·6	27·3	32·7	38·2	43·7
8	7·2	9·6	11·9	14·3	16·7	19·1	21·1	23·9	28·7	33·4	38·2
9	6·4	8·5	10·6	12·7	14·9	17·0	19·1	21·2	25·5	29·7	34·0
10	5·7	7·6	9·6	11·5	13·4	15·3	17·2	19·1	22·9	26·7	30·6
11	5·2	6·9	8·7	10·4	12·2	13·9	15·6	17·4	20·8	24·3	27·8
12	4·8	6·4	8·0	9·6	11·1	12·7	14·3	15·9	19·1	22·3	25·5
13	4·4	5·9	7·3	8·8	10·3	11·8	13·2	14·7	17·6	20·6	23·5
14	4·1	5·5	6·8	8·1	9·8	10·9	12·3	13·6	16·4	19·1	21·8
15	3·8	5·1	6·4	7·6	8·9	10·2	11·5	12·7	15·3	17·8	20·4
16	3·6	4·8	6·0	7·2	8·4	9·6	10·7	11·9	14·3	16·7	19·1

(*S. Osborn's Table.*)

The cutting speeds given are normal for good class cutters made from recognised high-speed steels.

The particulars and dimensions given in fig. 7 cover a useful range of milling cutters for ordinary work.

CUTTER DIMENSIONS.

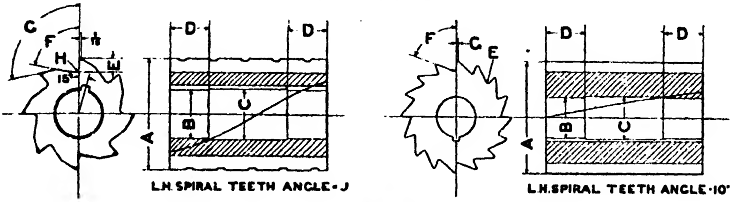


FIG. 7.

Coarse Pitch.
Dimensions in Inches.

Diam. of Cutter. A	B	C	D	E	F	G	H	J	Key-way.	No. of Teeth.
2½	1	1 1/16	1¼	3/8	80°	103°	1/8 R	26¼°	5/32 × 3/32	8
3	1¼	1 5/16	1¼	3/8	80°	103°	1/8 R	26¾°	3/16 × 1/8	8
3½	1½	1 9/16	1¼	7/16	80°	103°	1/8 R	26°	¼ × 5/32	8
4	1¾	1 13/16	1¼	7/16	71°	94°	3/16 R	25½°	¼ × 5/32	10
4½	2	2 1/16	1¼	½	71°	94°	3/16 R	26¾°	5/16 × 3/16	10

Fine Pitch.
Dimensions in Inches.

Diam. of Cutter. A	B	C	D	E	F	G	Key-way.	No. of Teeth.
2½	1	1 1/16	1¼	3/64 R	70°	3/64	5/32 × 3/32	14
3	1¼	1 5/16	1¼	3/64 R	70°	3/64	3/16 × 1/8	14
3½	1¼	1 5/16	1¼	3/64 R	70°	3/64	3/16 × 1/8	16
4	1½	1 9/16	1¼	1/16 R	65°	1/16	¼ × 5/32	18
4½	1½	1 9/16	1¼	1/16 R	65°	1/16	¼ × 5/32	20

The above table gives particulars of a range of cutters suitable for heavy roughing work, and also corresponding fine pitch cutters used for light milling.

(P. V. Vernon)

(Paper read before the Manchester Association of Engineers, 1912.)

Negative Rake Milling.

The application of cimented carbide tips to milling cutters has been the subject of considerable experiments both in the U.S.A. and in Great Britain.

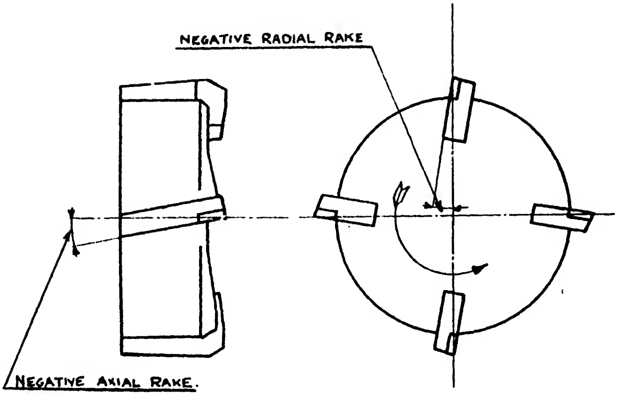
Milling is an intermittent cutting operation and therefore each tooth or cutter is subject to a series of shocks as it enters the workpiece. Cimented carbide is very brittle and its cutting edge soon collapses under shock. This has been the chief cause of the failures that have occurred when cimented carbide tips have been used on conventional milling centres. When positive rakes are used the full force of the cut is taken near the point of the cutter which is unsupported and is in tension, but if negative rakes are used the direction of the cutting thrust is such that the tip is in compression and it therefore can take the shock of intermittent cutting better with a corresponding increase in cutter life (fig. 8). It is, therefore, possible with negative rakes to take full advantage of the improved cutting qualities of cimented carbide and to overcome the inherent weakness of this hard but brittle cutting medium. With negative rakes higher cutting speeds are possible and an improvement in the finish of the work is obtained. Harder grades of tips can be used and by this means the effect of cratering due to abrasion near the cutting edge is considerably reduced. This is an important advantage, for usually cracks develop from a crater and so cause the failure of the cutter. In general fewer teeth are used in the cutter than is the case with conventional milling and loose flywheels are fitted on to the machine spindle to make up to some extent for the lack of rigidity in the available machines.

Axial Rake.

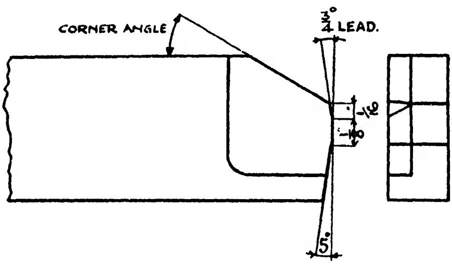
For the best cutter performance the negative axial rake should be selected to give the best cutting conditions. It is important that it should be of sufficient magnitude to protect the chamfer or radius on the corner of the teeth so that the impact load is taken at a point remote from the corner when the tooth contacts the entering side of the work. The axial rake angles in use at present vary between 3° negative and 15° negative.

Radial Rake.

The radial rakes being used vary from 5° negative to 10° negative for milling steels. One maker recommends 5° negative radial rake for steels in the 325–500 Brinell hardness range, gradually



NEGATIVE RAKE MILLING CUTTER.



ENLARGED VIEW OF INSERT.

FIG. 8.

increasing the rake to 8° positive for steel of Brinell hardness 200. Standard cutters may be obtained in the following combinations :—

	Axial Rake.	Radial Rake.
(a)	10° Negative.	10° Negative
(b)	5° ,,	5° ,,
(c)	10° ,,	5° ,,
(d)	5° ,,	10° ,,

Bevel or Corner Angle.

This angle on face or end milling cutters serves exactly the same purpose as the side cutting edge angle on a single point tool. It protects the chamfer or radius when the cutter first touches the work and so tends to increase the cutter life; therefore its width should always be greater than the depth of cut envisaged. When the cutter mills past the work a corner angle of 15° is generally satisfactory. If a 15° corner angle is not sufficient to allow the chamfer or radius to enter the metal last the angle should be increased and may be made as large as 35°. When milling up to a shoulder and a 0° bevel angle is used the radial rake should be from 3°–7° negative.

Face.

The face of the cutter should be ground 2°–4° concave from toe towards the heel and a clearance below the heel ground at 5°–7° making the face ⅛ in.–$\frac{3}{16}$ in. long. This eliminates the picking up of work material when the cutter is operating.

Tip Thickness.

The thickness of the tips used should be as thick as possible and those used for steel should be 30 per cent. to 50 per cent. thicker than those used for cast iron.

Cutting Speeds.

It is possible to use increased cutting speeds with negative rake milling cutters and the average cutting speeds for steel are given below.

Brinell Hardness.	Cutting Speeds in Ft. per Minute.
300 and below	500–700
300–400	250–500
400 and above	150–350

The formula for obtaining the cutting speed in feet per minute for a given diameter of cutter when the revolutions per minute are known is:

Cutting speed in ft. per minute = 0·262 × cutter diameter in ins. × R.P.M.

Feed.

The table feed rate in inches per minute for a given milling job should always be determined from the feed per tooth. It has to be established that for a reasonable cutter life that a feed per tooth of 0·005 in.–0·008 in. will give the maximum number of accurately finished workpieces per grind, when using face mills which cut past the work. In certain cases where the milling machine and fixture are sufficiently rigid and there is ample power available a feed of 0·015 in. may be used to advantage. For slotting cutters or other cutters which cut up to a shoulder the feed per tooth should only be about 50 per cent. of that used for ordinary face mills.

The rate of table feed which can be obtained is dependent on:

1. The number of teeth in the cutter.
2. The cutter speed in revolutions per minute.
3. The feed per tooth.
4. The horse-power available.

The following formulæ give the relationship between the various functions:

D = Average depth of cut in ins.
R = Revolutions per minute of cutter.
W = Maximum width of cut in ins.
F = Feed per tooth in ins.
H.P. = Horse-power of driving motor.
N = Number of teeth in cutter.

$$N = \frac{0{\cdot}56 \text{ H.P.}}{D \times F \times R \times W}$$

$$\text{H.P.} = 1{\cdot}8\, D \times F \times N \times R \times W.$$

The above formulæ are based on the generally accepted assumption that 1 h.p is required to remove ¾ cub. in. of steel per minute and that 35 per cent.–40 per cent. more power is required when the cutter is dull.

DRY CUTTING.

The best cutter life is obtained when milling dry because the chips are thin and most of the heat generated flows into them until they seem to approach a plastic state. In this condition they wear the cutting edge much less than when they are chilled by a coolant.

Cutter Bodies.

Cutter bodies should be designed to give the chips an unrestricted flow from the cut and consideration must always be given to chip clearance when designing a cutter. In certain instances it is possible to direct a strong air blast against the cut in order to remove any chips that are inclined to stick to the cutting edge. Where chips stick there is always a danger that they will ruin the cutting edge when it hits the work. Cutter life is lengthened if the work is not dragged across the cutter on the return stroke because a pressure is built up between the cutter and the work to the extent that the cutter actually takes a cut which is shallower than the machine setting. Therefore on the return stroke when the work and cutter resume their normal positions the cutting edge drags across the work. This not only marks the work but ruins the cutting edges and causes chipping when the cutter is not rotating. If the work cannot be removed before the table is returned to the starting position the cutter should be kept rotating. This will permit it to cut its way across the work and although it is liable to ruin the finish on the work and to increase the wear on the cutting edge, it is better practice than to drag an unrotating cutter across the work. The ideal set up is one where either the cutter or the workpiece is automatically relieved on the return stroke.

In all negative milling operations it is desirable to use as massive a cutter body as possible to give a flywheel effect. This reduces the shock when the cutting edge impinges against the work, provides a smooth cutting action and results in an increased number of pieces per grind. It enables the machine to take momentary peak loads considerably greater than that of the motor capacity. With most milling machines it is recommended that auxiliary flywheels be added to the spindles to make the cutting operation as smooth as possible.

Machine Requirements.

It is important that the machine should be robust, rigid and sufficiently powered for the desired milling operations. All backlash should be eliminated from the feed screws and the table strips or gibs should be adjusted so that a good sliding fit is obtained.

Spindle end play should be reduced by taking up any wear on the adjusting collars. The sliding ways of the machine should be checked periodically because wear occurs when the table is only used on one section and should a longer cutting stroke be required the table may be a good fit in places and sloppy in others, which will cause chatter.

Work Fixture.

The correct type of work holding fixture or clamping device has a direct bearing on the success of any milling operation. Owing to the increased feed rates which are now possible with cimented carbide cutters the cutting time is reduced to such an extent that the time required to load and unload the workpiece often becomes a major problem. Therefore the material handling time must receive special consideration. The only way to reduce this is to provide special fixtures and actuate them either by air or hydraulic pressure. One solution is to use an automatic cycle type of milling machine which has two fixtures, one of which can be unloaded and loaded by the operator while a part is being milled in the other. The following observations should be of help when designing a fixture:

1. The fixture should be arranged so that when the workpiece is clamped rigidly it remains in a state of rest without bending or torsional strains.
2. The workpiece should be clamped so that the surface being milled is as close as possible to the machine table.
3. The cutting area and the machine table should be kept as close as possible to the spindle nose.
4. The milling fixture should be robust enough to absorb all the forces created by clamping and cutting without springing or deflecting the machine table and should be able to absorb the vibration developed by the cut.
5. The fixture should be designed to allow for the free flow of chips from the workpiece.
6. All cutting pressures should be taken against rigid stops and supports.
7. Align all milling fixtures with the table slots by inserted keys on the bottom of the fixture.

Direction of Rotation of Cutter.

Longer cutter life is obtained with climb milling than with conventional milling. Fig. 8 illustrates the two methods. The main reason for the improved results from climb milling is that every tooth takes a chip of definite thickness at the time it enters the cut. With conventional milling every tooth starts with a chip which has no thickness and reaches its maximum thickness near the end of its cut. The maximum chip thickness obtained with conventional milling is practically the same as the chip thickness at the beginning of a climb cut. At the beginning of a climb cut each tooth actually cuts a chip, while at the beginning of a conventional cut there is an abrasive action until sufficient pressure has been built up to where the cutting edge penetrates the work. All milling machines are not designed for climb cutting and caution should be used before attempting climb cutting on any milling machine. Those machines which are provided with a backlash eliminator in the feed cut can be used for climb milling.

Broaching Machines and Broaching.

In common with other machine tools and small tools, the broaching machine and the broach has been developed to meet modern requirements, although the process of broaching for internal surfaces has been in use for many years, it is only in recent years that the process has been used for machining external surfaces of both plain and irregular contours. The advantages of broaching operations in the machine shop are now fully recognised, but it is still true to say that it is not an economical proposition for every machine shop, as it is only in factories engaged on repetition work in large quantities that the relatively high cost of the broach can be justified.

At the same time, in cases when the number of similar parts to be machined are big enough, the broaching process is extremely accurate and economical for internal and for external work.

Machine Design.

The modern broaching machine is now built on much heavier lines than formerly, it is recognised that the accuracy and quality of finish with this process, depends largely on the machine being sufficiently rigid to prevent vibration and distortion. It also affects the life of the broach, a springy or vibrating machine would quickly wreck the tool.

Both horizontal and vertical machines are used for broaching, the most suitable being entirely dependent on the nature and class of work the machine is used for. In some cases twin broaches can be operated simultaneously to accelerate the rate of production, whether used for machining internal or outside surfaces.

Types of Drive.

There are several recognised types of drive all operating satisfactorily on horizontal and vertical broaching machines on many varieties of work.

(i) Full mechanical drive with fast and loose pulleys, by belt from line shaft or countershaft.

(ii) Electric drive from motor to machine by Vee pulleys and belts, and mechanically operated clutch.

(iii) Full hydraulic drive for cutting and quick return stroke.

(iv) Combination of mechanical and hydraulic drive, the cutting stroke being mechanical and the quick return stroke hydraulic.

The factors that usually govern the choice of drive for the broaching machine are much the same as with other machine tools. These are—First cost. Product of the factory in which the machine is to be used. Position the machine is to occupy in the factory. Probable percentage of full time the machine will be worked.

In a factory where the machine can only be in operation a small percentage of full time, the cheapest form of drive will probably be found the most economical. But in a factory where the broaching process is continually in operation and maximum output is of first importance, the question of first cost is of relatively small importance, and the most reliable type of drive will prove most economical in the long run.

Cutting and Return Speeds.

The efficient cutting speed of the broach is generally governed by the hardness of the material to be cut. In that respect it differs but little from other forms of metal cutting, but does not always follow the same rule, as in practice it is found necessary to reduce the cutting speed for broaching very soft steel parts because of the risk of the teeth biting into and tearing the soft metal, but when cutting medium and hard steels the rules of cutting run true, the medium hard steel being cut at higher speeds than the very hard metal.

Broadly speaking, cutting speeds for broaching may be stated thus:—

Medium soft	15 to 25 ft. per min.
Medium hard	12 to 14 ,, ,,
Hard	5 to 10 ,, ,,

In all cases, however, the finish required on the work, and the life of the broach must be taken into account when deciding the permanent cutting speed for any particular class of work.

All broaching machines have quick return speeds, the usual ratio between the return and the cutting speeds being:—

Speed Ratio.

Medium and large machines—Return 2. Cutting 1.

Small machines—Return 2·5. Cutting 1.

Hydraulic variable speed internal broaching machines—Return 3. Cutting 1.

Coolant.

All broaching work requires a copious supply of coolant delivered under pressure, and must be of sufficient quantity to keep the work and the tool quite cold and prevent inaccuracies occurring through heating.

For internal work the coolant is delivered to the point where the broach enters the work piece and cutting commences.

For external work the coolant should be delivered to two points. One at the cutting position to cool and lubricate the tool, and the other at the point where the broach leaves the work, this being to wash the chips off the tool and keep it free from cutting before the return stroke is made.

There are a number of lubricants used for coolant purposes varying from a straight undiluted oil to a compounded lubricant containing several ingredients.

Broaches.

Broach tool making, both for internal and external work, is highly specialised work, and can only be successfully carried out by experienced men.

Selection of the steel from which the broach is to be made is of first importance, and should receive careful consideration.

Actually, the cost of the tool steel used in making the broach is only a small percentage of the total cost of the finished tool, and as the working life of the tool depends in large measure on the class of steel used and its treatment, it follows that it pays handsomely to use the best steel obtainable.

Special brands of steel are made by the various tool steel makers, and the correct heat treatment for annealing, hardening and tempering are always supplied by the steel maker.

To secure the best results the treatment specified by the steel maker should always be closely adhered to, and as that varies with the different classes of steel, according to its chemical analysis, no general rule can be given that would suit every case.

But as an example, showing the care that should be taken in the heat treatment of steel for *Broach* making in particular as well as for other tools in general, the following method of heat treatment is given in proper sequence for each process.

Heat Treatment for Broaches.

Process.

I. Machine broach roughly to within $\frac{3}{16}$ in. of finished sizes, all over.

II. Heat slowly and uniformly in furnace to 800° C. (1,472° F.) and cool right out in oil, preferably whale oil.

III. Anneal broach by heating slowly in furnace to a uniform temperature of 900° C. (1,652° F.), maintaining temperature for half an hour to two hours according to size of broach, then allow to cool in furnace with doors and dampers closed.

IV. After annealing, machine the broach to within $\frac{1}{64}$ in. of finished size all over.

V. Normalise by heating slowly to a uniform temperature of 650° C. (1,200° F.), allowing the broach to cool slowly with the furnace.

VI. Finish machine and grind to exact dimensions.

VII. For final hardening, pack broach in muffle or box with charcoal to a depth of 2 ins. round every surface, then heat slowly in furnace to 850° C. (1,562° F.) temperature inside the box, taking care that heating is uniform throughout (the temperature of the furnace will be rather higher than the interior of the box, probably about 900° C. (1,652° F.). After heating immerse in whale oil until quite cold.

VIII. To temper finally the broach immerse in oil which is maintained at a temperature of 200° C. (392° F.). The length of time the broach is immersed for tempering depends on the type and size, and judgment must be exercised in this respect.

Broach Design for Internal Work.

Pull type broaches for small and medium size work are made in solid tool steel, with cutting teeth formed on the outside to the shape and finished dimensions of the work to be broached.

On the larger sizes, these broaches are sometimes bored through the centre to facilitate hardening.

In the case of broaches of approximately 3½ ins. diameter and upwards, it is found more satisfactory to make the cutting teeth in the form of rings and mount them in series on a high carbon steel centre to make up the required length. The cutting parts are thus easily replaced when worn below size without involving the cost of a new broach.

Even with solid broaches, particularly of the expensive type, it is advisable to make the last 4 or 6 cutting teeth in the form of rings, and mount them on the solid broach, as this prolongs the working life of the broach.

Push type broaches, used on hand or power presses, are made shorter than the pull type; they are commonly used for sizing holes in bushes and other parts.

Burnishing broaches are usually made to be forced through the work and are comparatively short in length. They are made much the same as a sizing broach, but have blunt or rounded teeth which have the effect of burnishing the hole and removing the tool marks, leaving a smooth surface of exact size.

Practically any shape or form of internal surfaces can be cut and finished to exact size by broaching, providing a bore is first made to allow the broach to enter.

The most common form of work for the internal broach is cutting and finishing a round, square, or any shape hole. Cutting and finishing single or double keyways. Cutting and finishing splined holes to any number and shape of spline.

Examples of the form of broach used for internal work are shown in figs. 9, 10 and 11. Methods of attaching the broach to the machine are shown in figs. 9 and 10. These are the ordinary methods adopted, but some machines have semi-automatic gripping and releasing devices fitted, these being designed for quick operations. Fig. 10 also gives an example of the form of cutting teeth for broaching keyways and splined holes, and fig. 11 gives details of broaches for round holes. In cases where the amount of metal to be removed is greater than can be done with one broach it is usual to use two or three on the same piece.

Broach Design for External Work.

Broaches for external work are made for pulling or pushing over the surface of the work. In cases where broaching is done by pushing, an ordinary power press, with suitable attachments, may be used. Some machines are designed to do both internal or external broaching, these usually being of the pull type.

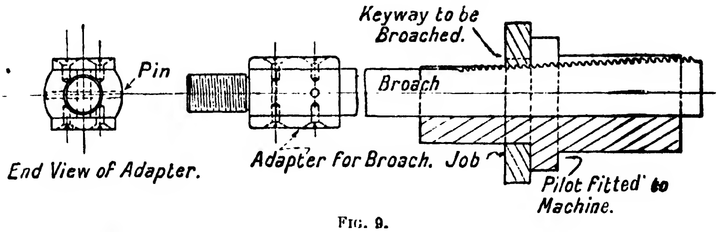


FIG. 9.

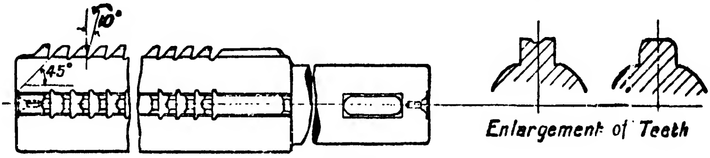


FIG. 10.

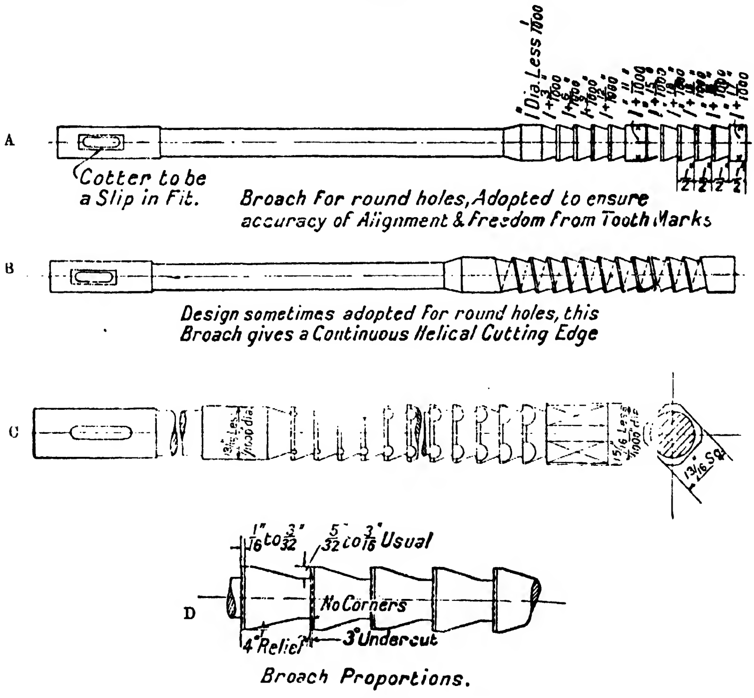


FIG. 11.—Types of Broaches.

The design of the cutting teeth for external work is the same in principle as that used for internal work, see examples in fig. 9, 10 and 11, but at most only three sides of the broach may be used on external surfaces, as one side at least must be used for support, and in order to prevent chatter and vibration, with consequent risk of breakage, it is essential to have the broach rigidly supported and held up to the face of the work.

While many solid broaches are used for external work, it is often more economical to use those of the built-up type, having the cutters inset in steel of good quality, but generally this type can only be used where the faces to be machined are sufficiently deep to allow the cutters to be spaced and leave adequate metal between them to securely hold them in place. In all forms of broaching, internal and external, it is necessary to have at least two, but preferably three or four, teeth in contact with the work, otherwise good finish will not result.

The actual cutting time in all forms of broaching is relatively short, and often changing the work takes longer than the cutting operation. For this reason it is necessary to have specially designed locating and clamping attachments made for external broaching, as otherwise much of the advantage of the process may be lost.

Where well-designed fixtures are provided, the number of pieces that can be accurately machined by the external broaching process runs into hundreds per hour, and may equal in quantity and quality that of internal broaching.

Power Required by Broaching Machines.

Horizontal Hydraulic, Variable Speed Broaching Machine.

Normal pulling capacity, in lbs. .	6,100	21,000	41,000	105,000
Cutting speed (variable) ft. per min.	30 max.	26 max.	20 max.	15 max.
Return speed " " "	120 "	180 "	100 "	60 "
Hydraulic pressure, lbs. per sq. in. .	1,000	1,000	1,000	1,000
Maximum length of stroke, ins. .	38	32	64	76
Direct connected motor, H.P. .	3	7½	10	25

Vertical Hydraulic, Variable Speed, Surface Broaching Machine.

Capacity, tons	2½	5	15	20
Maximum travel, ins. . . .	30	36	42	54
Maximum length of broach, ins. .	24	30	36	48
Variable cutting speed, ft. per min.	32	25	25	20
Variable return speed " "	68	60	60	40
Motor recommended, H.P. . .	5	7½	15	20

Mechanically Driven Vertical Broaching Machine.

	Surface Broaching.	Internal Broaching.
Capacity, tons	5	5
Stroke of Ram, ins.	26	26
Maximum length of Broach, ins. .	24	31
Broaching speed, ft. per min. .	3·8 to 10	6·3 to 15·5
Motor drive, H.P.	5	5

Changes of speed effected by altering ratio of belt pulleys.

Twist Drills.

Special grinding machines for sharpening twist drills are now universally used in production factories and works, and if properly set these machines ensure that the angle of the cutting edges are correct

A twist drill possesses three cutting edges, A, B, and C, as shown in fig. 12.

The point A, being the least effective, since it cannot be made anything like as keen as the other two, B and C, is a hindrance to clean and rapid cutting and is one of the most important factors to be considered when arriving at the amount of feed progression per inch of travel.

In order to obtain the greatest possible duty from a twist drill, it is imperative that it should be

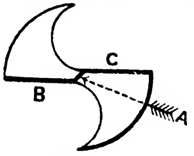


FIG. 12.

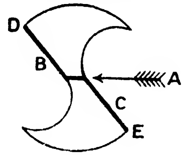


FIG. 13.

ground perfectly true, so that the point of the drill shall be central and in line with the axis on which it revolves. The cutting edges must be of equal length and at the same angle to the axis of the drill. If the point of the drill is not central to the axis, the two cutting edges are not at the same angle, the result being that all the duty falls on one cutting edge, and the hole made will still be larger in diameter than the drill, there being a tendency for the cutting edge B to push, or crowd, the drill over to the opposite side of the hole.

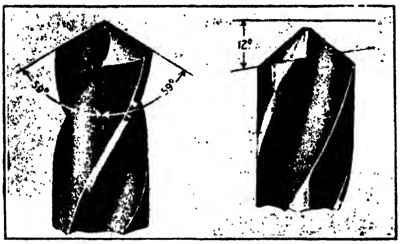
Another important point when the maximum duty is expected from a twist drill is the relationship between the speed and feed, which varies more or less with different materials and with different grades of hardness of like materials.

The grooves of a twist drill make either a constant angle with the axis or an angle that increases gradually from point to shank. The latter form is supposed to facilitate the escape of the cuttings, though opinions differ, but it has the serious effect of rendering the cutting angle more obtuse as the drill becomes ground back. When the angle is constant it ranges from 24° to 32° with the axis, measured at the periphery, 26° being an average. In the increasing-twist drills it ranges roughly between 30° and 25°, the angle lessening by that amount from point to shank. The thickness of the web is increased from point to shank to enable it to resist torsional stresses. The diminished area for the escape of chips consequent on this thickening is counteracted by increasing the twist.

Grinding Drills.

When grinding the drill points, the following rules should be observed :—

I. Both cutting lips, fig. 14, should be inclined at the same angle with the axis of the drill, and must be of equal length. The point angle of 59° has been universally adopted as best suited for average conditions



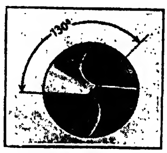


FIG. 16.

FIG. 14. FIG. 15.

II. The drill point must have the proper clearance or contour of surface behind the cutting edges, and this clearance must be identical on both sides. Approximately a 12° clearance angle, fig. 15, combined with the centre angle of 130°, which will give a constantly increasing clearance toward the centre, fig. 16, has proved best for average conditions.

Most twist drills are made with a gradual increase in the thickness of the web or centre of the drill toward the shank. As the drill becomes shorter and the web thicker, greater force is required to drive it. To overcome this, it is good practice to thin the web by grinding away the excess thickness, reducing it to its original dimensions. This grinding must not extend too far up the flute of the drill, and care must be exercised that the cutting lips are not injured and the same amount is ground out of each groove.

Twist drills are made with a slight taper from point to shank, so that the largest diameter is always across the corners of the cutting lips. This prevents the drills from binding in the work when they are sharp. If the outer corners are allowed to become badly worn, the drills will bind and cannot perform satisfactorily. Whenever the outer corners of the cutting lips show wear, the drills should be reground and every particle of worn surface removed, or the drill will continue to bind and very quickly be damaged beyond repair.

In grinding high-speed drills care should be taken not to overheat them, and when heated they should never be plunged into cold water. Doing so is likely to cause small surface cracks, which reduce the efficiency of the drill and may result in serious damage to it. Forcing the grinding on a wet grinder may also bring about the same condition.

It is hardly possible to do grinding as accurately by hand as by using a good twist-drill grinding machine.

Broken or damaged tangs of drills are generally the result of an imperfect fit of the drill shank in its socket, which may be caused by a worn-out socket, dirt or chips accumulating in the socket, or bruises on the shank of the drill.

In any case, the driving power of the taper is reduced or destroyed, resulting in an abnormal strain being put upon the tang.

Morse Taper Measures for Twist Drills.

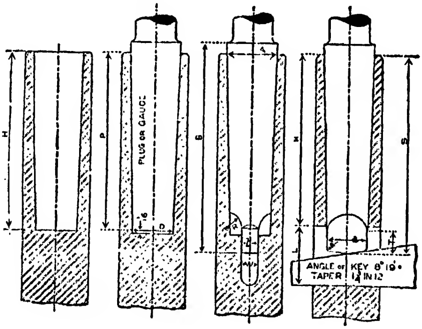


FIG. 17.

Number of Taper.	Diam. of Plug at small End.	Diam. at End of Socket.	Shank. Whole Length of Shank.	Shank. Shank Depth.	Depth of Hole.	Standard Plug Depth.	Tongue. Thickness of Tongue.	Tongue. Length of Tongue.	Tongue. Rad. of Mill for Tongue.	Tongue. Diam. of Tongue.	Tongue. Radius of Tongue.	Keyway. Width of Keyway.	Keyway. Length of Keyway.	End of Socket to Keyway.	Taper per Foot.	Taper per Inch.	Number of Key.
	D	A	B	S	H	P	t	T	B	d	a	W	L	K			
0	·252	·3561	2 11/32	2 7/32	2 1/32	2	5/32	1/4	5/32	·235	·04	·160	9/16	1 15/16	·625	·05205	0
1	·369	·475	2 9/16	2 7/16	2 3/16	2 1/8	13/64	3/8	3/16	·343	·05	·213	3/4	2 1/16	·600	·04988	1
2	·572	·700	3 1/8	2 15/16	2 5/8	2 9/16	1/4	7/16	1/4	17/32	·06	·260	7/8	2 1/2	·602	·04995	2
3	·778	·938	3 7/8	3 11/16	3 1/4	3 3/16	5/16	9/16	9/32	23/32	·08	·322	1 3/16	3 1/16	·602	·05019	3
4		1·231	4 7/8	4 5/8	4 1/8	4 1/16	15/32	5/8	5/16	31/32	·10	·478	1 1/4	3 7/8	·623	·05193	4
5		1·748	6 1/8	5 7/8	5 1/4	5 3/16	5/8	3/4	3/8	1 13/32	·12	·635	1 1/2	4 15/16	·631	·05262	5
6	2·116	2·494	8 9/16	8 1/4	7 3/8	7 1/4	3/4	1 1/8	1/2	2	·15	·760	1 3/4	7	·626	·05213	6
7	2·750	3·270	11 5/8	11 1/4	10 1/8	10	1 1/8	1 3/8	3/4	2 5/8	·18	1·135	2 5/8	9 1/2	·624	·05200	7

British Standard Specification for Twist and Straight Flute Drills.

(No. 328—1928.) (*Extract.*) *

Tables are given in this specification of the standard diameters, overall lengths and flute lengths for the various types of drills, and the dimensions of the Morse taper shanks. Standard limits are laid down for the permissible variation in the diameter of the drills, and performance tests are specified for both carbon and high-speed drills from No. 60 to 2 ins. diameter, inclusive. The performance tests consist in drilling a series of holes, as continuously as possible, in steel billets of a given analysis. The penetration per minute and the number and depth of the holes to be drilled are specified, but the revolutions per minute and the feed per revolution are left to the discretion of the maker. The drills must withstand these tests without seizing, choking, or fusing, and the points of the drills and the lips must be in good condition and fit for further service without regrinding on completion of the tests. Standard nomenclature and definitions for the different types of drills in common use are also included.

Morse Wire Drill Gauge.

No. Wire.	Decimal Equivalent Inches.	No. Wire.	Decimal Equivalent Inches.	No. Wire.	Decimal Equivalent Inches.	No. Wire.	Decimal Equivalent Inches.	Letter.	Decimal Equivalent Inches.
80	0·0135	58	0·0420	36	0·1065	15	0·1800	F	0·2570
79	0·0145	57	0·0430	35	0·1100	14	0·1820	G	0·2610
78	0·0160	56	0·0465	34	0·1110	13	0·1850	H	0·2660
77	0·0180	55	0·0520	33	0·1130	12	0·1890	I	0·2720
76	0·0200	54	0·0550	32	0·1160	11	0·1910	J	0·2770
75	0·0210	53	0·0595	31	0·1200	10	0·1935	K	0·2810
74	0·0225	52	0·0635	30	0·1285	9	0·1960	L	0·2900
73	0·0240	51	0·0670	29	0·1360	8	0·1990	M	0·2950
72	0·0250	50	0·0700	28	0·1405	7	0·2010	N	0·3020
71	0·0260	49	0·0730	27	0·1440	6	0·2040	O	0·3160
70	0·0280	48	0·0760	26	0·1470	5	0·2055	P	0·3230
69	0·0292	47	0·0785	25	0·1495	4	0·2090	Q	0·3320
68	0·0310	46	0·0810	24	0·1520	3	0·2130	R	0·3390
67	0·0320	45	0·0820	23	0·1540	2	0·2210	S	0·3480
66	0·0330	44	0·0860	22	0·1570	1	0·2280	T	0·3580
65	0·0350	43	0·0890	21	0·1590	Letter.		U	0·3680
64	0·0360	42	0·0935	20	0·1610	A	0·2340	V	0·3770
63	0·0370	41	0·0960	19	0·1660	B	0·2380	W	0·3860
62	0·0380	40	0·0980	18	0·1695	C	0·2420	X	0·3970
61	0·0390	39	0·0995	17	0·1730	D	0·2460	Y	0·4040
60	0·0400	38	0·1015	16	0·1770	E	0·2500	Z	0·4130
59	0·0410	37	0·1040						

Note.—The Drill sizes No. 80 to No. 60, inclusive, are manufactured in Carbon Steel only.

Notes on Drilling.

The power for a given diameter of drill and feed is proportional to the cutting speed.

The horse-power is proportional to the torque, and for a given drill and speed does not increase as fast as the feed.

Since the torque is practically proportional to the diameter of the drill squared, the horse-power, for a given feed and cutting speed, is proportional to the diameter of the drill.

The horse-power per cubic inch of metal removed is inversely proportional to the feed and independent of the drill and cutting speed.

The work required to drill a given hole, when one drill is used, is greater than that required to drill the same hole in two operations with drills of different diameters. The greater the difference in the drill diameters, the greater the saving in work; speed and feed remaining the same throughout. This is due to the fact that the mean cutting angle of the single drill is greater than the average angle in use for the two drills and that the stress is proportional to the angle.

With twist drills of the usual proportions, the cutting angle is not sufficiently keen to drag the drill into the work when enlarging a hole in cast iron or steel.

The horse-power when operating on soft cast iron or medium steel varies as $t^{0\cdot7}$ (t = feed per rev.) for a given drill and speed.

The horse-power for a given feed and speed does not increase as fast as the diameter but varies as $d^{1\cdot8}$ (where d is the diameter of the drill in inches).

* By permission of the British Standards Institution.

The torque and horse-power when drilling medium steel is about 2·1 times that required to drill soft cast iron with the same drill speed and feed.

The end thrust when operating on cast iron or steel does not increase in proportion to the feed for a given diameter of drill or in proportion to the diameter for a given feed.

While the chisel point scarcely affects the torque it is accountable for about 20 per cent. of the end thrust.

Lubricated trials on steel when compared with the dry tests show a diminution in the torque and horse-power, varying from 28 per cent. with the $\frac{1}{400}$ feed to 8 per cent. with the $\frac{1}{54}$ feed. This may be due to the lubricant washing away the small metal chips which tend to jam between the walls of the hole and the drill, and to the preserved cutting edge. The diminished frictional resistance of the shaving across the lip together with above reduces the end thrust by about 25 per cent. for all feeds.

As the result of a series of experiments on twist drills, recently undertaken at the engineering experiment station of the Illinois University, the following lubricants are recommended:— For tool steel—lard oil, machine oil, turpentine, soda-water, kerosene; for soft steel—lard oil, machine oil, soda-water; for cast iron—dry or compressed air; for brass—dry or paraffin oil; for aluminium—kerosene, soda-water, and aqualine and soda-water.

The drill most commonly adopted in practice has an included angle, at the point, of 120°. If this angle is increased, the torque diminishes but the end thrust increases, while if this angle is decreased. the reverse is the result. So far as economy in power is concerned the torque is the factor to consider, as the feeding horse-power is only about 1 per cent. of the whole, in small drills, and very much less for the larger sizes. From this point of view the drill with the larger point angle is to be preferred. The accompanying increased end thrust, however, strains the machine parts in proportion. When the point of the drill breaks through the metal at the bottom of the hole, a considerable portion of the end load is removed. The strain due to that load is released, thereby causing the drill to advance more than its rated feed, and possibly to break. The drill with the greater included angle will be most likely to give trouble in this direction, both on account of the increased strain and torque, and conversely.

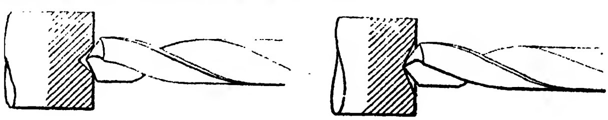
By decreasing the spiral of the drill a keener cutting angle with a decreased end thrust and torque can be obtained without altering the point angle above the accepted standard, 120°. This, however, would in turn affect the durability of the drill.

With a small included point angle there is little metal to support the cutting edge at the chisel point, and trouble due to blunting of this part is to be expected.

In estimating the time required to drill a hole of given depth, the length of the drill point must be taken into account. The length of the point for different included point angles is:

$$90° = 0·5d; \quad 120° = 0·29d; \quad 150° = 0·134d.$$

True drilling is usually dependent upon good centering, and it may be advisable in turret lathe work to draw attention to the well-known but often ignored rule, namely, that the centering tool must have a lesser cutting angle than the drill point, otherwise the drill will strike on the point, wobble, start out of centre, and sometimes break.



A Good centering. FIG. 18. *B* Bad centering.

Fig. 18 shows good centering at *A* and bad at *B*. As a general rule the centering drill should have an included angle of 100° against the standard angle for the drill point of 119°.

Speed and feeds at which twist drills can be run are dependent on the material to be drilled and the depth of hole. A general guide for drilling various metals is given in Table X. The

speeds and feeds given are those usually obtainable with good class high-speed drills used on machines where the necessary power is available.

TABLE X.

Diam. of Drill in Inches.	Soft Brass, Aluminium and Gun Metal.		Soft Cast Iron.		Mild Steel.		Malleable Iron, Phosphor Bronze, Medium Carbon Steel, Mild Steel Drop Forgings.	
	200 F.P.M.		100 F.P.M.		90 F.P.M.		70 F.P.M.	
	R.P.M.	Revs. per inch of Feed.	R.P.M.	Revs. per inch of Feed.	R.P.M.	Revs. per inch of Feed.	R.P.M.	Revs. per inch of Feed.
1/16	12230	350	6115	350	5500	350	4278	350
1/8	6112	250	3056	250	2750	250	2139	250
3/16	4074	175	2037	175	1833	175	1426	175
1/4	3056	120	1528	120	1375	120	1070	120
5/16	2444	120	1222	120	1100	120	856	120
3/8	2038	90	1019	90	917	90	713	90
7/16	1746	90	873	90	786	90	611	90
1/2	1528	75	764	75	688	75	535	75
5/8	1222	60	611	60	550	60	428	60
3/4	1018	60	509	60	458	60	357	60
7/8	872	60	436	60	393	60	306	60
1	764	60	382	60	344	60	267	60
1-1/4	612	60	306	60	275	60	214	60
1-1/2	510	60	255	60	229	60	178	60
1-3/4	436	60	218	60	196	60	153	60
2	382	60	191	60	172	60	134	60

Diam. of Drill in Inches.	Hard Cast Iron, 40 Ton Axle Steel, Steel Castings.		Chrome Tool Steel, High Speed Steel, Nickel Chrome Steel		5% Nickel Steel.		Hard Steel Castings.	
	65 F.P.M.		60 F.P.M.		45 F.P.M		40 F.P.M.	
	R.P.M.	Revs per inch of Feed.	R.P.M.	Revs. per inch of Feed.	R.P.M.	Revs. per inch of Feed.	R.P.M.	Revs. per inch of Feed.
1/16	3974	350	3557	350	2750	350	2445	350
1/8	1987	250	1833	250	1375	250	1222	250
3/16	1324	175	1222	175	917	175	815	175
3/4	993	120	917	120	688	120	611	120
5/16	795	120	733	120	550	120	489	120
3/8	662	90	611	90	459	90	407	90
7/16	568	90	524	90	393	90	349	90
1/2	497	75	458	75	344	75	306	75
5/8	397	60	367	60	275	60	244	60
3/4	331	60	306	60	229	60	203	60
7/8	284	60	262	60	197	60	175	60
1	249	60	229	60	172	60	153	60
1-1/4	198	60	183	60	138	60	122	60
1-1/2	166	60	153	60	115	60	102	60
1-3/4	142	60	131	60	98	60	87	60
2	124	60	115	60	86	60	76	60

(S. Osborn's Table.)

Cemented Carbide Tipped Tools.

Cemented carbide tipped tools were first introduced on a commercial basis in 1926 and since that time they have been used increasingly in all branches of machining. See fig. 19.

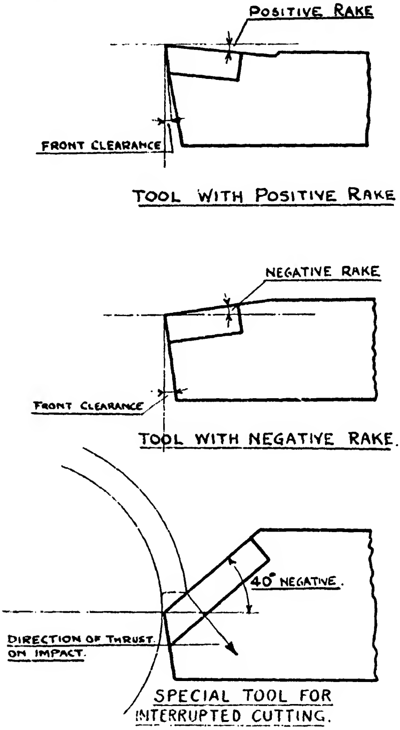


FIG. 19.

The tip is moulded and cintered and can afterwards only be shaped by grinding with special wheels or by cutting with diamond impregnated wheels.

The tip is brazed on to its shank by the use of a torch, furnace or electric resistance heater.

Cemented carbide is very hard, but its resistance to shock is much lower than that of high-speed steel. It is also strong in compression but weak in tension, therefore it must be well supported by the shank.

Tool Shanks.

The cross section of the shank should be as large as possible, and the material recommended is 0·5 carbon steel with a tensile strength of 45/50 tons per sq. in.

Care must be taken when brazing the tip to the shank that during the process the tool must be heated up and cooled down slowly.

Conditions required for Successful Application.

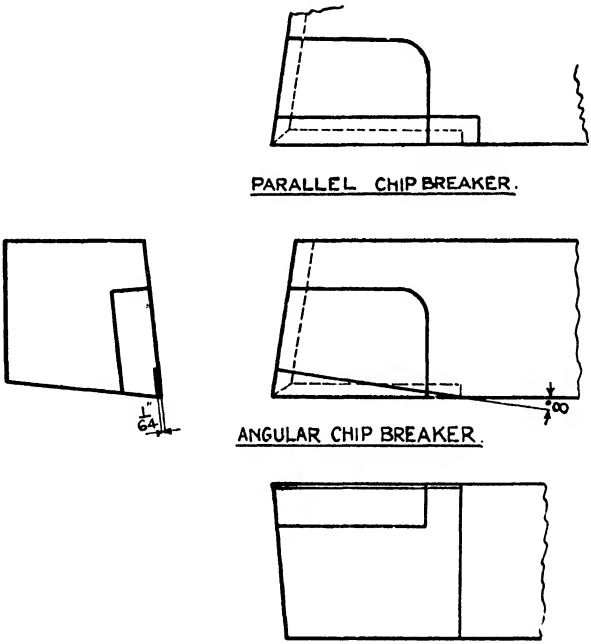
The lathe on which cemented carbide tipped tools are used must be sufficiently powerful for the work to be carried out at the necessary speed. It should be rigid; there must be no end play or radial play in the spindle bearings and there must be no slip in the drive. The chuck and running centre must hold the workpiece positively. Vibration or chatter is absolutely fatal to a cemented carbide tip. It is important that the cutting speed should not be too low.

Grinding.

It is essential to regrind the tool before the cutting edge becomes dull, and specially designed grinding machines have been made for this purpose. Special abrasive and diamond impregnated wheels must be used to obtain the best possible finish on the cutting edge. The nose radius should, unless there is some definite reason to the contrary, be kept as small as possible, for a large radius produces a chip section which runs out to a very thin edge and tends to cause excessive building up of metal on the cutting edge, which produces a rough finish on the work and shortens the tool life through abrasion. It also increases the tendency to chatter.

Chip Breakers.

The normal angles at which tools are usually ground produce long turnings when machining steel. At the high speeds used when cutting with cemented carbide they are difficult to control and are a danger to the operator. A chip breaker ground on the top of the tool will correct this. The depth and width of the chip breakers vary with the depth of cut, the feed, and the class of material being machined. Fig. 20 shows two types of chip breakers: (*a*) where the groove is



TYPICAL CHIP BREAKERS.

FIG. 20.

ground parallel to the cutting edge, and (*b*) where the groove is ground at 8° to the cutting edge. The depth of the groove should not be more than $\frac{1}{64}$ in.

Negative Rake.

The strength of a cemented carbide tip is considerably increased by the use of a negative top rake. The effect of the negative rake is to increase the 'lip' angle by about 20° and to increase that section of the tip which bears the shearing stress of the cutting force (fig. 19). By this means the possibility of failure due to shear is practically eliminated. The impact load on the tip is moved away from the point of the tool so enabling the tool to meet conditions of greater severity such as heavy roughing cuts on rough castings or forgings, interrupted cuts on lathes and on planing machines. The abrasive action between the chip and the tip also takes place at a point remote from the point of the tool, therefore failure due to cratering is long delayed.

Tool Angles.

It has been found in practice that good results are obtained with a negative top rake of between 2° and 8°, 3° and 5° being the average. The side rake should be positive and of about 2°–5° greater than the negative rake angle. With this combination it has been found that a better finish and longer tool life have been possible; there is also a reduction in the power consumed.

General Conditions.

The tool overhang should always be reduced as much as possible by supporting the tool shank as near to the cutting edge as practicable.

The projection of the tool should only be sufficient to allow clearance for the work and a free passage for the chips.

The cutting edge should never be set up to the work and then the tool clamped down.

The tool should never be set by tapping the tip with a spanner.

Never stop the work with the tool engaged, withdraw the tool first.

The correct speed should be chosen.

For the material being cut and the type of tip being used the table below gives average figures.

MATERIAL.	Roughing.			Finishing.		
	Cutting Speed ft./min.	Feed ins./rev.	Depth of Cut. in ins.	Cutting Speed ft./min.	Feed ins./rev.	Depth of Cut in ins.
Steel below 40/45 tons tensile	200–250	0·025–0·040	$\frac{3}{8}$–$\frac{1}{2}$	600–800	0·005–0·008	0·015–0·030
Steel 50–75 tons tensile	180–250	0·025–0·040	$\frac{3}{16}$–$\frac{1}{2}$	600–800	0·005–0·008	0·015–0·030
Nickel chrome steel above 75 tons tensile	120–200	0·020–0·040	$\frac{1}{4}$–$\frac{3}{8}$	250–400	0·003–0·005	0·015–0·030
Steel castings	150–200	0·020–0·040	$\frac{1}{4}$–$\frac{3}{8}$	300–500	0·005–0·008	0·010–0·020
Cast iron below 200 Brinell	180–220	0·010–0·020	$\frac{1}{4}$–$\frac{5}{8}$	350–450	0·010	0·015–0·030
Cast iron above 200 Brinell	160–200	0·012	$\frac{1}{4}$–$\frac{3}{8}$	200–300	0·005–0·008	0·015–0·030
Planing cast iron	70–90	0·060	$\frac{3}{8}$	140–180	0·015	0·062
Copper	500–800	0·030	$\frac{1}{4}$	750–1000	0·012	0·010–0·020
Hard brass	400–600	0·015–0·020	$\frac{3}{16}$	500–800	0·010	0·010–0·020
Aluminium	600–1000	0·012	$\frac{1}{2}$	1000–3000	0·004	0·010–0·020
Zinc base alloys	300–400	0·015–0·020	$\frac{1}{4}$	500–900	0·004	0·010–0·020
Plastics Erinoid hard rubber	400–600	0·006	$\frac{1}{8}$–$\frac{3}{16}$	600–800	0·003	0·010–0·020

The following cutting tests were made on a Lodge and Shipley 27 in. lathe driven by a 50 h.p. motor using tungsten carbide tipped tools:

Material, 65 tons tensile nickel steel bar.

Sliding cut $\frac{3}{8}$ in. deep × 0·030 in. feed.

Cutting speed, 300 ft. per minute.

H.P. required to run the lathe when idling, 3.

	Test 1.	Test 2.
Tool Angles	10° Negative Back Rake.	4° Negative Back Rake.
	5° Negative Side Rake.	8° Positive Side Rake.
Gross H.P. used . .	14	12
Net H.P. used . .	11	9

Where there are severe interruptions in the cut specially designed tools are required; a typical case is illustrated in fig. 19. This tool has a negative back rake of 40° and a positive side rake of 15° which gives a pronounced shear action on the chip. It will be noticed that the chip impinges on the tip at a point well behind the nose.

Magnetic Chucks.

The Magnetic Chuck is now largely used for holding machine parts of iron and steel for grinding, turning, and planing operations. Many pieces, such as milling cutters, thin saws, air-compressor valve discs, chain links, etc., can be more effectively held by this method than by any other. Not only is there a great saving in the time ordinarily required for setting up, but the likelihood of distorting the job due to unequal straining of the holding down bolts is eliminated.

It must be understood that the holding power of a magnetic chuck depends as much on the shape and section of the piece to be held as on the magnetic chuck itself. If the piece is small in section the hold will be very light, no matter what the section or intensity of the chuck magnets may be. The hold is entirely dependent upon the number of magnetic lines that can be passed through the section of the piece, and not on the total magnetism that the chuck can produce. Some makers claim a holding power of 100 lbs. per sq. in., this having been attained on a perfectly surfaced cube of Swedish iron applied between the chuck poles. This affords no useful measure of the holding value of a magnetic chuck, as theoretically a small magnet either of steel, wrought iron or even cast iron, if perfect at the joints, would give (on such a cube) 100 lbs. per in. On the other hand, if two such cubes were placed together the pull would not be proportionately increased, as the magnet being small the section of the first cube would absorb all the magnetism it could supply. Thus a chuck, say 30 ins. by 8 ins., which would pull 100 lbs. on any sq. in. of its surface could not, under any circumstances, pull 30 ins. by 8 ins. by 100 lbs., or nearly 11 tons, on its entire surface. The pitch of the poles should be arranged to suit the class of work to be held; if the pieces are of heavy section, the magnet must be of heavy section to get the best results.

Fine Grinding.—For fine grinding it is well to reduce the magnetic pull of a chuck face, either by inserting resistances in the coil circuit, or by the interposition of non-magnetic material between the work and the chuck face. The lighter the pull the more accurate the resultant work. Specially constructed chucks of a very light but even pull are necessary for thin saws, light piston rings, etc. Grinding with water, soda and other liquids makes it absolutely essential that the magnetic chucking apparatus must be impervious to water.

Switches.—One of the most important pieces of apparatus in connection with the magnetic chucks are the switches used for operating. When the current from a magnetic chuck is cut off, a back current is induced in the chuck coils. This induced current is of a very high voltage, and if the switch gear is not perfectly designed to deal with it, the insulation of the chuck, however well designed and carried out, will inevitably break down. The switch must be double pole, that is, it must operate on both the positive and negative supply wires at the same time, also it must be capable of admitting current to the chuck in either direction, and in order to deal with the induced back current when switching the chuck off, it must be provided with a non-inductive resistance which must automatically be put in circuit with the chuck prior to breaking the circuit.

TOOL STEEL AND TOOLS.*

Carbon Steel and High-Speed Steel Cutting Tools.

Their Preparation and Use.

CARBON STEEL TOOLS.

Carbon tool steel is suitable for making such tools as are required to keep a keen edge that is not easily dulled. It can be hardened and tempered to the degree necessary for the purpose for which it is to be used, and can therefore be used with advantage for a large variety of purposes

* For chemical composition see METALLURGY, Section XXIII, Part 1.

such as finishing tools for metal cutting, woodworking tools and razor blades, also for drawing dies, stamping and pressing tools.

But care must be taken to select the proper grade of steel for the purpose for which it is to be used, remembering that the 'temper,' or degree of hardness, is governed by the carbon content of the steel, so that when ordering carbon steel, the percentage of carbon required should always be specified.

Some well-known makers of carbon tool steel recommend the per cent. of carbon for various uses as given in Table XI.

TABLE XI.

Purpose for which Carbon Steel is Used.	Carbon Content.
Roll turning tools, razor blades, vulcanite turners' tools . .	$1\frac{1}{2}$ per cent.
Cutting tools for lathes, slotters, planers, borers	$1\frac{1}{4}$,,
Drills, reamers, milling cutters, screwing dies, joiners' tools .	$1\frac{1}{8}$,,
Taps, pneumatic chisels, lathe centres, mint and other dies .	1 ,,
Hand chisels, caulkers' tools, shear blades, punches, cold sets .	$\frac{7}{8}$,,
Hammers, masons' tools, stamping and pressing tools, miners' drills	$\frac{3}{4}$,,

Like all other tool steels, carbon tool steel requires careful treatment when forging, hardening and tempering, as the grain of the steel may easily be affected. When the steel is heated in an ordinary smith's hearth the blast should never be allowed to impinge directly on it, as this may result in hard and soft places in the tool, and may even cause cracks during the hardening process. Whenever possible the steel should be heated in a muffle where the temperature is under better control. It is always advisable to allow sufficient time for the heat to penetrate through the steel before beginning to forge to shape.

The cutting tools used on such machines as lathes, planers and slotters are forged to near the shape and angle required, they are then finished on dry grinding machines before hardening and tempering. When rough grinding the tools to shape and angle, it is advisable to use free-cutting carborundum grinding wheels, and to grind away the rough metal at a rate that will not overheat the tool. It is always necessary to grind well below the forging surface so as to remove the film of decarburised steel that is always formed by the forging or even rolling process. This decarburised metal will not harden, so that for small light tools at least $\frac{1}{32}$-in. of metal should be ground away, and on the heavier tools not less than $\frac{1}{16}$-in. is recommended. If this is done it will prevent hard and soft spots, and help to ensure a uniform hardness throughout the cutting edge of the tool.

It is usual to make drills, taps and reamers by machining direct from the rolled bar, but it is false economy to use bar steel that is too near the finished size, adequate machining of the surface must be allowed for, to remove the decarburised metal.

Milling cutters and forming tools may also be made from rolled bar steel, but generally it is more economical to obtain forged blanks from the steel makers than to cut from the bar, particularly if only a few tools of the same size are required. In all cases where the steel is cut from the bar, it is always safer to cut it while hot than to nick it and break off cold—there is always danger of causing internal cracks if heavy carbon steel is broken off cold even after it has been deeply nicked. There is generally less waste of steel when the blanks are cut from the bar either with a power hack saw or in a cutting-off machine. But always allow sufficient machining on the blanks for the reason given above.

All tools and cutters that are machined to shape should first be rough machined all over, and all surplus metal cut away until the shape approximates to the finished form. It is then necessary to anneal the tool to relieve all the strains in the metal, and, in the case of forged blanks, to restore the structure and density of the metal.

Annealing of any metal, and particularly carbon tool steel, is always more effective when the process is carried out in a muffle or air-tight box. The muffle or box containing the steel is then brought to the proper temperature and allowed to remain at that temperature until the heat has penetrated through the metal, after which it is allowed to cool very slowly.

It is sometimes necessary to anneal the steel before it can be machined, but even when it is not absolutely necessary, it is generally found more economical first to resort to the annealing process because of the time that is saved by being able to cut the steel at higher speeds, particularly is that the case where large numbers of tools are being made.

Where a muffle is not available, steel of moderate size can be annealed by bringing it slowly to the given temperature, but not soaking it too long, and then covering it up with a thick layer of hot ashes or sawdust and allowing it to cool before being removed.

In every process of making the tool of whatever size or shape, there is a correct temperature which must be closely adhered to, if satisfactory results are to be obtained.

The required temperature for each process of forging, annealing and hardening, are always more definitely obtained by the use of a pyrometer and a heat-controlled furnace, and these should always be used where large numbers of tools are being made.

• When carbon tool steel has been heated to a degree reaching the carbon change point, that is a temperature of 685° C. to 735° C. and then quickly cooled, it reaches its maximum hardness and is said to be 'dead hard.' If, however, scale has formed on the tool during previous heating, it is advisable to raise the temperature 50° C. over the temperature given above to ensure the heat penetrating right through the steel before quenching. In this condition the metal is highly stressed, and if used for cutting in that state it would be liable to chip away at the cutting edge, or may even fracture at the point of extreme hardness. So that before using it is necessary to reduce the degree of hardness by the process known as tempering.

To temper hardened carbon steel tools, it is necessary to reheat the tool to a temperature varying from 220° C. to 330° C. or 428° F. to 626° F. and then quickly cool in water or other cooling liquid. The temper depends entirely on the class of tool and the purpose for which it is to be used.

When tempering tools, the temperature is judged by the colour which forms on the surface, due to oxidation, while being heated. But before starting to temper it is necessary to thoroughly clean the surface of the tool with emery cloth or emery block, in order to remove any oxidation already there from previous heating, as otherwise the temper colour cannot form on the surface. The temperature corresponding to the temper colour is given in Table XV. It should be noted that colour changes only operate between 220° C. and 330° C. and that for other temperatures a pyrometer should be used.

After tempering it is only necessary to grind the tool lightly to bring it to a keen edge, but care should be exercised to avoid heating the tool during this grinding, or any subsequent grinding, as this may destroy the tempered hardness required for the work.

The sequence of operation in tool-making from carbon tool steel can be taken in the following order. See Table XII.

Forged Tools.—From carbon tool steel having 1¼ per cent. carbon content. These generally relate to: turning, planing, boring, slotting and shaping tools.

TABLE XII.

Operation No.	Operation.	Temperature.		Heat Colour.
		Cent.	Fahr.	
1	Cut steel of suitable size and length from bar. First heat the steel if cut by set, or cut cold with hack saw	825°	1,517°	Cherry Red
2	Forge near to shape required	825°	1,517°	,,
3	Anneal (only in cases where more than ordinary forging has been necessary)	720°	1,328°	Blood Red
4	Grind to shape and angle. Removing $\frac{1}{16}$-in. metal from surface	—	—	—
5	Harden tool. Heating evenly all through the metal	730°	1,346°	Low Cherry Red
6	Clean surface of tool with emery cloth or emery block	—	—	—
7	Temper tool. Allow the heat to travel towards the tip of the tool	225°	437°	Straw
8	Finish. Grind the tool to be ready for use. Take care not to draw the temper	—	—	—

Milling cutters, twist drills, reamers, taps and broaches are made from tool steel bar, or in the case of milling cutters of large diameters from tool steel blanks. When made from carbon tool steel, it is usual to select a steel having 1⅛ per cent. carbon.

The outside machining allowance for rolled bar or forged blanks should not be less than ⅛-in. diameter, on sizes from 1 in. to 3 in., $\frac{3}{16}$-in. diameter on sizes from 3⅛ to 5 ins., this will ensure the decarburised metal being removed and prevent soft places in the cutter.

A reliable course of treatment for expensive tools, such as milling cutters, form tools, broaches, dies, and long tops used for tapping boiler stay holes, etc., is given in Table XIII. This treatment should obviate risk of distortion when finally hardened.

TABLE XIII.

Operation. No.	Treatment.	Temperature. Cent.	Temperature. Fahr.	Heat Colour.
1	Rough machine all over, leaving $\frac{1}{32}$-in. all over on small tools and $\frac{1}{16}$-in. all over on large tools . .	Steel already annealed		
2	First hardening. Heat slowly all over and immerse in oil	800°	1,472°	Dull Red
3	Annealing. Heat slowly in closed furnace for 1 hour and cool down in furnace	720°	1,328°	Dark Red
4	Machine again to within $\frac{1}{64}$-in. for small tools and $\frac{1}{32}$-in. for large tools	Steel annealed		
5	Normalise. Heat slowly in furnace all over, and cool down in furnace	565°	1,049°	Brown Red
6	Finish tool to exact size.	Steel normalised		
7	Finally harden in muffle packed with charcoal . .	730°	1,346°	Cherry Red
8	Tempering requires great care. Tools of long length should be heated in oil or lead bath to required temperature	230°	446°	Straw

The temperatures recommended for forging, hardening and tempering carbon tool steel vary somewhat with the different tool steel makers, but the temperatures given in Tables XIV and XV for forging and hardening respectively, with the specified carbon content of the steel, may be taken as safe practice.

TABLE XIV.

Carbon Content.	Forging Carbon Steel Tools. Maximum Temp. Fahr.	Maximum Temp. Cent.	Corresponding Surface Colour.	Hardening Carbon Steel Tools. Maximum Temp. Fahr.	Maximum Temp. Cent.	Corresponding Surface Colour.
1⅜ %	1,517°	825°	Bright Cherry Red	1,346°	730°	Low Cherry Red
1¼ %	1,517°	825°	Bright Cherry Red	1,346°	730°	,, ,,
1⅛ %	1,562°	850°	Full Red	1,382°	750°	Medium Cherry Red
1 %	1,562°	850°	,, ,,	1,400°	760°	Medium Cherry Red
⅞ %	1,562°	850°	,, ,,	1,436°	780°	Cherry Red
¾ %	1,562°	850°	,, ,,	1,436°	780°	,, ,,

When heating the steel for forging or hardening, do not bring it up to full heat too quickly, but allow sufficient time for the heat to penetrate right through the steel, without overheating the outside surface, at the same time do not soak the tool by allowing it to remain in the furnace too long. The best practice is to heat up gradually and uniformly until the proper temperature is reached and then proceed with the forging or hardening at once. Also it is better to reheat the tool several times for forging than to continue hammering after the tool has cooled to a brown-red colour, which indicates a temperature of 565° C. and 1,049° F.

The correct temper for carbon tool steel varies with the class of tool and the purpose for which it is used. For instance, the colour temper of a turner's tool for cutting brass is pale straw, corre-

sponding to a temperature of 220° C. and 428° F., while the colour temper for shear blades is full blue, indicating a temperature of 293° C. and 560° F. But where a given class of tool is used constantly for the same purpose, experience very quickly shows which is the most suitable temper for the work. General practice however is given in Table XV, and this will be found reliable for the range of tools covered.

TABLE XV.

Tempering Carbon Steel Tools.

Purpose for which the tool is used.	Temp., Fahr.	Temp., Cent.	Temper Colour.
Turning tools for non-ferrous metals	428°	220°	Pale straw
Turning tools for iron and steel .	437°	225°	Straw
Planing and slotting tools . .	446°	230°	Dark straw
Circular paper cutters . . .	455°	235°	Very dark straw
Milling cutters and drills . .	467°	242°	Brownish yellow
Taps, dies, hard wood planers . .	495°	257°	Brown
Press tool, surgical instruments .	518°	270°	Red-brown
Woodworker's hand tools . .	527°	275°	Purple
Hand chisels, centre punches . .	540°	282°	Light blue
Shear blades, fine saws . . .	560°	293°	Full blue
Saws for wood, circular steel saws .	565°	296°	Dark blue
Steel springs	572°	300°	Greyish-blue

Various cooling mediums are used for hardening and tempering carbon steel tools. For ordinary forged tools used on lathes, planers, slotters and the like, water is found quite satisfactory, particularly water that is soft and free from lime, but it is necessary to have an ample supply to prevent the water becoming heated with continual quenching of tools. Where it can be arranged running water is recommended, but in any case sufficient cold water should be constantly added to the hardening bath to keep it cool. To harden these tools it is only necessary to heat the cutting tip to the specified temperature, and then to dip the heated tip of the tool in the quenching water. If the water is still, that is not running or being stirred, the tool should be moved about so as to prevent the water in contact with the hot tool becoming heated.

For hardening and tempering milling cutters, taps, dies and other carbon steel tools of that nature, it is advisable to quench in oil. Whale oil and cotton-seed oil are frequently used with satisfactory results, but it is necessary that whatever oil is used it should have a high flash point, and means provided to prevent heating. For tempering the more intricate tools special tempering baths are obtainable, using oil, lead and lead alloys which are heated to the proper temperature for tempering the tool. These are recommended for use when tempering expensive tools such as long taps, internal and external broaches.

HIGH-SPEED STEEL TOOLS.*

Suitable high-speed steels are available for making practically every class of tool used. The first known self-hardening tool steel was discovered by Mushet in 1871, and for some thirty years was the only tool steel that remained unaffected by the heat generated by cutting metals at high speeds, the high speeds of those days being about 30 ft. per min. Alloy cutting tools will now cut metals up to 300 ft. per min., and cutting speeds up to 100 ft. per min. with several brands of high-speed steels are common every day practice.

High-speed steels are readily forged to shape to be used as cutting tools on the lathe, planer, slotter, borer and other machine tools. It is primarily used for rough machining metal parts to within narrow limits of finished size, leaving only a small margin to be removed by grinding.

Finishing tools are also made from high-speed steel of suitable chemical composition. These are capable of being hardened and tempered to retain a keen cutting edge that is not affected by ordinary cutting temperatures.

Measured in terms of price per pound of steel, high-speed steels appear expensive, but measured in terms of performance and durability they are found to be very economical.

In the annealed state high-speed steel is readily machined, and there is no difficulty in making milling cutters, form cutters, twist drills, taps, reamers and many other engineers' small tools, hardened and tempered to suit a great variety of work.

* Refer to METALLURGY, Section XXIII, Part I, p. 1160, for chemical composition.

Owing to the high cutting speeds and feeds made possible by the use of these steels, it is necessary to use them on machine tools that have the requisite power for driving, and the rigidity to prevent vibration when cutting to the full capacity of the high-speed steel tool, as otherwise much of the advantage of using superior tools will be lost.

It should be noted that the cutting efficiency of high speed steel tools is dependent on the correct heat treatment being followed during the forging and hardening process. Also that the heat treatment varies according to the chemical composition of the steel. Hence it is only possible to give general rules and examples for guidance. The correct temperatures for forging, hardening, annealing and tempering should always be obtained from the steel makers, who are best qualified to give directions for the proper heat treatment of their particular brand. The directions given must be closely followed.

Steel for turning and planing tools can be heated in a blacksmith's fire, but for milling cutters and drills it is advisable, wherever possible, high-speed steel should be heated in a closed muffle furnace, with facilities for temperature control and registration. The heating medium may be solid fuel (coal or coke), liquid (oil), gas or electric current. Gas as a means for heating has many advantages, being quick in heating and easily controlled.

The size of the furnace naturally depends on the weight of steel to be treated, but while it is wasteful to use a large furnace for small work, it is a mistake to use one that is too small, as in that case the furnace will be appreciably cooled every time cold steel is put in. It is more economical in the long run to have a furnace that is automatically heat controlled and that is not unduly affected by the weight of steel put in.

Temperature registration is most important, and wherever possible should be taken with pyrometers or thermometers, where these are not available the temperature may be judged fairly near by the colour of the heated steel. Table XVI gives the colour for varying heats in Centigrade and Fahrenheit. To work successfully by colour alone requires judgment and experience, but where doubt exists it is better as a rule to treat the steel at a temperature above rather than below that specified, particularly when heating for forging purposes.

TABLE XVI.

Heating Steel—Colours and Temperatures.

Colour.	Temperature Centigrade.	Temperature Fahrenheit.
Brown Red	565°	1,049°
Dark Red	600°	1,112°
Blood Red	700°	1,292°
Blood or Low Cherry Red	730°	1,346°
Low to Medium Cherry Red	740°	1,364°
Medium Cherry Red	750°	1,382°
,, ,, ,,	760°	1,400°
Cherry Red	770°	1,418°
,, ,,	780°	1,436°
,, ,,	800°	1,472°
Bright Cherry Red	825°	1,517°
Full Red	850°	1,562°
Bright Red	900°	1,652°
,, ,,	920°	1,688°
Full Bright Red	955°	1,751°
Yellow Red	980°	1,796°
White	1,250°	2,282°
Full White	1,300°	2,372°
,, ,,	1,330°	2,426°
Incandescent White	1,350°	2,462°
,, ,,	1,360°	2,480°

Forging and heat-treatment temperatures usually recommended by the makers of 14 per cent. tungsten and 18 per cent. tungsten vanadium high-speed steel tools, are generally as follows:—

HIGH-SPEED LATHE AND PLANING TOOLS.

First heat gently in a slow part of the fire until the steel is red (850° C.) then bring the heat quickly to near white heat (1,200° C.) and forge to shape. Re-heat again when the heat falls to a bright cherry red. Do not attempt to forge the steel below bright cherry red or splitting may result.

In the case of heavy tools or tools of uneven section, it is possible the steel may be strained during forging, and it is necessary to allow them to cool slowly in the smith's hearth during the night, or to anneal them properly.

Hardening.

Before commencing the hardening process it is convenient to rough grind the tool to shape and angle, using a dry, free cutting carborundum or similar grinding wheel. If the steel heats during grinding do not cool it in water or cracking may result, the steel must be allowed to cool in the atmosphere.

To harden, heat the cutting end to a white heat (1,250° C. to 1,300° C.) and then cool it down quickly in an air blast. The air blast supplied by fan to a smith's hearth is suitable, and it is sometimes convenient to take a connection from this supply for the purpose of cooling high-speed steel tools. The connection for this air should be approximately 6 ins. diam. and reduced to say 4 ins. diam. to form a nozzle. Place the heated point of the tool near the air blast, allowing the air to flow over the tool lengthwise. Air blasts from air compressors are not desirable: it is not only wasteful but may produce small cracks in the tool because of the particles of moisture usually found in compressed air.

When heating the tool for hardening take care to heat only the actual cutting part of the tool—that is, the tip—do not heat the stock.

If an air blast is not available, the tool may be cooled in oil; either whale or cotton-seed oil is suitable.

Tempering.

Ordinary lathe and planer tools do not as a rule require tempering, but it is sometimes advisable where intermittent cutting is done, or where the tool is small or fragile.

Where tempering is necessary, heat the tool to a straw colour, say to 230° C., and cool.

Secondary treatment is advised by some steel makers, as a means to obtaining maximum cutting efficiency from the tool. The treatment consists in re-heating a hardened tool to 580° C. and then blowing cold. This should increase the toughness, but is only successful after correct first hardening has been carried out.

Final Grinding.

High-speed steel tools should always be well ground after hardening to remove the top surface metal. A free-cutting abrasive wheel of vitrified or silicate bond is suitable, ample water supply is necessary to keep the tool cool while being ground.

High-Speed Milling Cutters and Twist Drills.

Forged blanks are supplied by leading tool steel makers, suitable for making milling cutters and form tools. It is usual to make twist drills, reamers and taps from the rolled steel bar.

The process in sequence of operation is given in Table XIII, and may be taken as a reliable guide in making these expensive tools.

For dealing with high-speed steels for these high-class tools, the precise heat treatment can be had from the maker, the steel can usually be supplied in the annealed condition if so specified, but where it is necessary to carry out the annealing process in the tool-room, the following is the routine advised by the makers of 14 per cent. tungsten and 18 per cent. tungsten vanadium steel.

Annealing.

Heat slowly in a closed furnace or muffle to a bright red (860° C.) taking care to prevent the entrance of cold air. Allow the steel to remain in the furnace until quite cold, when it can be withdrawn. Proceed to make the tool as given in Table XIII, but operate at the temperature as under.

Hardening.

First heat gradually in the furnace to 750° C. and then transfer the tool to a muffle furnace or a heated bath of lead and bring the temperature of the tool to 1,250° C. (white heat). At this heat withdraw the tool and quench in a bath of whale oil; when it is cooled to a black heat it may be finished cooled outright in the air blast.

Tempering.

Heat the tool in an oil bath to 250° C., or temper in the usual way for tempering cast steel to a light straw colour (225° C.). This considerably relieves the strains in the metal. For special tools, the secondary treatment may be applied, as given under Lathe and Planing Tools Section.

In all cases it is necessary to carry out each process at the temperatures recommended to ensure good results.

CUTTING TOOL ANGLES.

Cutting angles of high-speed steel tools vary according to their uses. With ordinary turning and planing tools the first essential is to adopt a shape as well as a cutting angle that will remove the maximum of metal in the shortest possible time, taken over the period of a working day. For this reason it is necessary that the shape and the cutting angle shall allow free cutting of metal without absorbing an undue amount of power, and at the same time have the quality of durability to avoid too much time being taken up by frequent re-grinding.

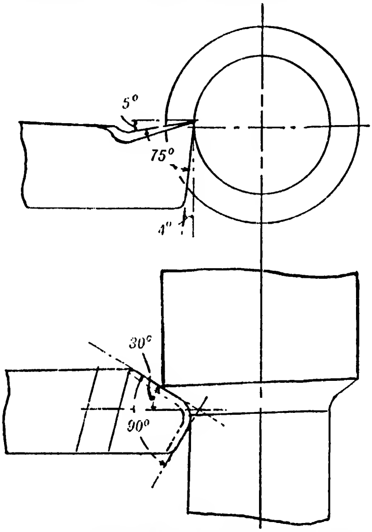


FIG. 21.—High-Speed Steel Rough Turning Tool.

The shape and cutting angle of the tool shown in the accompanying line drawing (fig. 21), is often used in machine shops engaged in turning medium and heavy parts in high tensile steel.

This tool has a combined cutting angle of 75°, the end clearance is 4°, side rake 10°, top rake (front to back) 5°, shear rake (left to right) 7°. The radius of the cutting point is ¼ in. for medium heavy work, this being scaled down to a radius of $\frac{1}{16}$ in. for light work, the leading cutting edge is ground to an angle of 30° and the trailing edge to 60°, giving a combined nose angle of 90°. Allowing sufficient metal to absorb the heat generated by cutting. The tool is generally freer cutting and less likely to cause vibration than the fuller bull-nosed type of tool, and the top shear rake allows the cuttings to come away in a natural flow.

A similar shape of tool operates equally well on the planing machine, though as a rule the radii would be reduced to $\frac{3}{16}$ in. when used on medium planing work.

Leading angles for tools cutting various metals may be stated thus:—

TABLE XVII.

Material.	Cutting Angle.	Clearance Angle.	Side Rake.	Top Rake.	Leading Cutting Angle.
Soft mild steel . . .	63°	6°	18°	7°	25°
Medium hard steel . .	69°	5°	14°	6°	30°
Hard steel . . .	75°	4°	10°	5°	30°
Cast iron, medium grey .	78°	8°	12°	4°	30°
Brass	82°	4°	14°	2°	4°
Good gun metal . .	84°	3°	14°	3°	4°

TABLE XVIII.—TABLE OF CUTTING SPEEDS.*

Feet per Minute	15 ft.	20 ft.	25 ft.	30 ft.	35 ft.	40 ft.	45 ft.	50 ft.	60 ft.	70 ft.	80 ft.
Diam.	REVOLUTIONS PER MINUTE.										
1/16	917·	1223·	1528·	1834·	2140·	2445·	2751·	3057·	3668·	4280·	4891·
1/8	459·	611·	764·	917·	1070·	1222·	1375·	1528·	1834·	2139·	2445·
3/16	306·	408·	509·	611·	713·	815·	917·	1019·	1222·	1426·	1630·
1/4	229·	306·	382·	458·	535·	611·	688·	764·	917·	1070·	1222·
5/16	183·	245·	306·	367·	428·	489·	550·	611·	733·	856·	978·
3/8	153·	204·	255·	306·	357·	408·	458·	509·	611·	713·	815·
7/16	131·	175·	218·	262·	306·	349·	393·	437·	524·	611·	699·
1/2	115·	153·	191·	229·	268·	306·	344·	382·	459·	535·	611·
5/8	91·8	123·	153·	184·	214·	245·	276·	306·	367·	428·	489·
3/4	76·3	102·	127·	153·	178·	203·	229·	254·	306·	357·	408·
7/8	65·5	87·3	109·	131·	153·	175·	196·	219·	262·	306·	349·
1	57·3	76·4	95·5	115·	134·	153·	172·	191·	229·	267·	306·
1 1/8	51·0	68·0	85·0	102·	119·	136·	153·	170·	204·	238·	272·
1 1/4	45·8	61·2	76·3	91·8	107·	123·	137·	153·	183·	214·	245·
1 3/8	41·7	55·6	69·5	83·3	97·2	111·	125·	139·	167·	195·	222·
1 1/2	38·2	50·8	63·7	76·3	89·2	102·	115·	127·	153·	178·	204·
1 5/8	35·0	47·0	58·8	70·5	82·2	93·9	106·	117·	141·	165·	188·
1 3/4	32·7	43·6	54·5	65·5	76·4	87·3	98·2	109·	131·	153·	175·
1 7/8	30·6	40·7	50·9	61·1	71·3	81·5	91·9	102·	122·	143·	163·
2	28·7	38·2	47·8	57·3	66·9	76·4	86·0	95·5	115·	134·	153·
2 1/4	25·4	34·0	42·4	51·0	59·4	68·0	76·2	85·0	102·	119·	136·
2 1/2	22·9	30·6	38·2	45·8	53·5	61·2	68·8	76·3	91·7	107·	122·
2 3/4	20·8	27·8	34·7	41·7	48·6	55·6	62·5	69·5	83·4	97·2	111·
3	19·1	25·5	31·8	38·2	44·6	51·0	57·3	63·7	76·4	89·1	102·
3 1/2	16·4	21·8	27·3	32·7	38·2	43·6	49·1	54·5	65·5	76·4	87·4
4	14·3	19·1	23·9	28·7	33·4	38·2	43·0	47·8	57·3	66·9	76·4
4 1/2	12·7	16·9	21·2	25·4	29·6	34·0	38·2	42·4	51·0	59·4	67·9
5	11·5	15·3	19·1	22·9	26·7	30·6	34·4	38·2	45·9	53·5	61·1
5 1/2	10·4	13·9	17·4	20·8	24·3	27·8	31·3	34·7	41·7	48·6	55·6
6	9·6	12·7	15·9	19·1	22·3	25·5	28·7	31·8	38·2	44·6	51·0
7	8·1	10·9	13·6	16·4	19·1	21·8	24·6	27·3	32·7	38·2	43·7
8	7·2	9·6	11·9	14·3	16·7	19·1	21·1	23·9	28·7	33·4	38·2
9	6·4	8·5	10·6	12·7	14·9	17·0	19·1	21·2	25·5	29·7	34·0
10	5·7	7·6	9·6	11·5	13·4	15·3	17·2	19·1	22·9	26·7	30·6
11	5·2	6·9	8·7	10·4	12·2	13·9	15·6	17·4	20·8	24·3	27·8
12	4·8	6·4	8·0	9·6	11·1	12·7	14·3	15·9	19·1	22·3	25·5
13	4·4	5·9	7·3	8·8	10·3	11·8	13·2	14·7	17·6	20·6	23·5
14	4·1	5·5	6·8	8·1	9·6	10·9	12·3	13·6	16·4	19·1	21·8
15	3·8	5·1	6·4	7·6	8·9	10·2	11·5	12·7	15·3	17·8	20·4
16	3·6	4·8	6·0	7·2	8·4	9·6	10·7	11·9	14·3	16·7	19·1

NOTE:—To find the revs. per min. of Speeds not mentioned in the above table, add or subtract any two numbers in the same line.

EXAMPLE:—To find the revs. per min. at 120 feet per min. for 6" diameter.

Speed at 80 ft. per min. = 51·00 revs. per min.

" " 40 " " = 25·50 " "

" " 120 " " = 76·50 " "

* *Balfour's Table.*

Shapes, sizes, and cutting angles of tools vary in different localities and works. As a rule each finds from experience (the best teacher) the tool to suit the work in hand. The tool illustrated and the table of angles given in Table XVII are those the author has found to give good results in works producing medium-heavy engineering products, and are intended as a guide to tool-making of good cutting quality.

Table XVIII gives cutting speed in feet per minute, on certain diameters, running at stated revolutions per minute.

Colours of Tempered Steel.

The *tempering* in this case relates to lowering the degree of hardness obtained by plunging the steel when heated to a cherry red (1,650° F., 899° C.) in cold water:—

Colour.	F.°	C.°	Colour.	F.°	C.°
Very pale yellow .	430	221·1	Dark purple . . .	550	287·8
Straw yellow . .	460	237·8	Clear blue . . .	570	298·9
Brown yellow . .	500	260·0	Pale blue . . .	610	321·1
Bright purple . .	530	276·7	Blue tinged with green .	630	332·2

Composition of Quenching Baths for Tempering Cutting Tools.

Fused nitrates of potassium and of sodium are too high in temperature for certain cutting tools, as they do not permit of cooling below 328° F. (220° C.). Mixtures of nitrate of potassium and of nitrate of sodium can, however, be employed, and a series of mixtures, fusing at different temperatures, be obtained.

Temperature.		Nitrate of Potassium.	Nitrate of Sodium.	Temperature.		Nitrate of Potassium.	Nitrate of Sodium.
F.°	C.°			F.°	C.°		
536	280	0 parts	100 parts	279	137	55 parts	45 parts
446	230	20 ,,	80 ,,	293	145	60 ,,	40 ,,
342	172	40 ,,	60 ,,	437	225	80 ,,	20 ,,
293	145	50 ,,	50 ,,	635	335	100 ,,	0 ,,

Higher temperatures than 752° F. (400° C.) cannot be obtained with these mixtures. In steels where without extreme hardness absolute absence of brittleness is necessary, 932° F. (500° C.) to 1,112° F. (600° C.) are temperatures more suitable. The following bath gives, on fusion, a temperature of 932° F.(500° C.):—

Sodium chloride, 1 part; potassium chloride, 1 part; fused calcium chloride, 2 parts; hydrated barium chloride, 1 part; hydrated strontium chloride, 3 parts.

For a bath fusing at 1,292° F. (700° C.) the following mixture may be used:—

Hydrated boric acid crystals, 1 part; silver sand, 1½ part; anhydrous potassium carbonate, 1 part; anhydrous sodium carbonate, 1 part.

When prolonged treatment is required, a little cyanide or charcoal may be added to prevent superficial decarburisation; but, in view of the strongly cementating action of cyanide, this salt must be used with caution.

Quenching Liquids.

When hot steel is continually dumped into a tank of oil, the lighter and volatile oils are evaporated and finally only the heavier oils will be left. When the oil becomes thick, it will not take the heat away from the steel rapidly; consequently, the steel will be soft and will not show the maximum physical properties. To obtain good results, the quenching liquid should be of a constant specific gravity and maintained at a temperature of 70° to 90°, but the exact temperature at which the bath is maintained does not matter so much if it is always the same. To ensure the best results, the bath should always be maintained at a certain specific gravity and at some certain temperature.

In hardening and tempering a chisel, the best way, says the *English Mechanic*, is to heat it to a very dull red for a good inch up from the edge, holding the tongs in the left hand, and in the right a rub stone, a piece of broken grindstone or emery wheel, or, failing these, a piece of emery cloth wrapped round a small file. Dip the chisel for about half-an-inch until it just turns black, then withdraw it from the water and let it rest against the side of the pail, so as to steady it; rub it sharply with the stone, so as to brighten it, and watch the colour as it runs down from the part which is still hot, and when the edge is of a deep plum colour, verging on blue, dip it right out.

Grinding.

1. The tool should be forged roughly to shape and ground to the requisite contour by means of a composition wheel (preferably dry) without any lubricant being applied. The heating of the tool during grinding at this stage is not of any importance.

2. After shaping, the tool should be subjected to the heat treatments indicated (page 1069).

3. It is advisable to finish-grind with a copious stream of water playing on the nose of the tool. If reasonable care to avoid heating be exercised, the tool may be ground on a composition wheel running in water.

Alternative Shape of Tool for Roughing Purposes.

The durability or length of time a tool will continue to cut under constant conditions as to feed, speed, etc., increases with the nose radius (fig. 22). The tendency to chatter, however, limits the extent to which the nose radius may be increased, and a round-nosed tool having a radius of $\frac{1}{4}$ in. to $\frac{3}{8}$ in. has been found to give the most satisfactory results for the general run of work. With work of a springy character a smaller nose radius may be required. The durability is also affected by the cutting angle, *i.e.*, the angle between the cutting face of the tool and the tangent to the work or 90° minus the top rake angle. The cutting angle is increased when the tool edge stands above the centre of the work, by an amount equal to the angle contained between the cutting edge of the tool and the axis of the bar.

The cutting angles which give the greatest durability to tools for roughing purposes are as follows:—

Material.	Cutting Angle.	Material.	Cutting Angle.
Soft steel . . .	65°	Soft cast iron . .	70°
Medium steel . .	70°	Medium cast iron . .	75°
Hard steel . . .	80°	Hard cast iron . .	85°
Nickel steel . . .	80°	Brass	85°
Manganese steel . .	85°	Hard gunmetal . .	85°

The above angles are measured in a plane at 45° to the length of the tool which is approximately the plane of the shaving. The side and front clearance angles for tools with cutting edges on the line of centres need not exceed 6°. The shape of the tool recommended is shown opposite.

Front top rake ($t°$); height above centre (Z); front top rake (reduced to centre) ($\beta + t°$); side top rake (S°); front clearance (C°); radius of work (R); angle (β); front clearance (reduced to centre) ($C° - \beta$); side clearance ($k°$).

Cutting Speeds for Carbon Steel Lathe Tools.

When finishing soft, medium, and hard steel the cutting speeds which may be used are 20 ft., 15·5 ft., and 11 ft. per minute, whilst for finishing soft, medium, and hard cast iron the cutting speeds are 24 ft., 19 ft., and 14 ft. per minute in the order given. When finishing brass and copper, speeds of 80 ft. and 150 ft. per minute respectively should be adopted. The finishing cut should not exceed $\frac{1}{32}$ in. in any case. (*Dempster Smith.*)

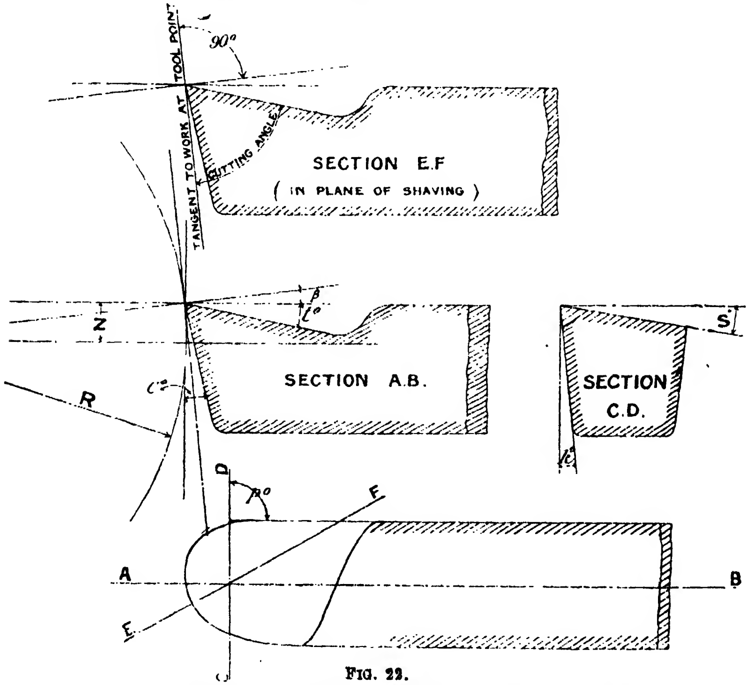


FIG. 22.

Cutting Speed for High-Speed Steel Tools when Cutting Steel.

An extensive series of experiments have been carried out by the Lathe Tool Research Committee of the Manchester Association of Engineers, when operating on steel under different conditions.* The Committee in many of its investigations selected cutting speeds which would give a twenty minutes life to the tool, but state that for practical purposes and economic considerations, a speed which will give a two hours run is desirable. A relation is, therefore, given between the cutting speed and durability of the tool, and this conforms to the expression :

$$VT^{\frac{1}{8}} = \text{constant},$$

where,

$$V = \text{cutting speed in ft. per min., and } T = \text{duration in mins.}$$

The value of the constant varies with the material operated upon and also with the dimensions of the cut. This relationship is of the utmost value, since for a given material, tool, and cut if

* See Report of Lathe Tool Research Committee, by Dempster Smith, published by H.M. Stationery Office, 1922. Price 5*s*.

the cutting speed V and the life of the tool T be observed, the cutting speed V_0 for the tool to last a time T_0 can be obtained thus:

$$V_0 = V\left(\frac{T}{T_0}\right)^{\frac{1}{8}}.$$

The change in the cutting speed with time as given by the above law may be better appreciated fiom its graphical representation in fig. 23, where the appropriate cutting speed is plotted on a base of tool life.

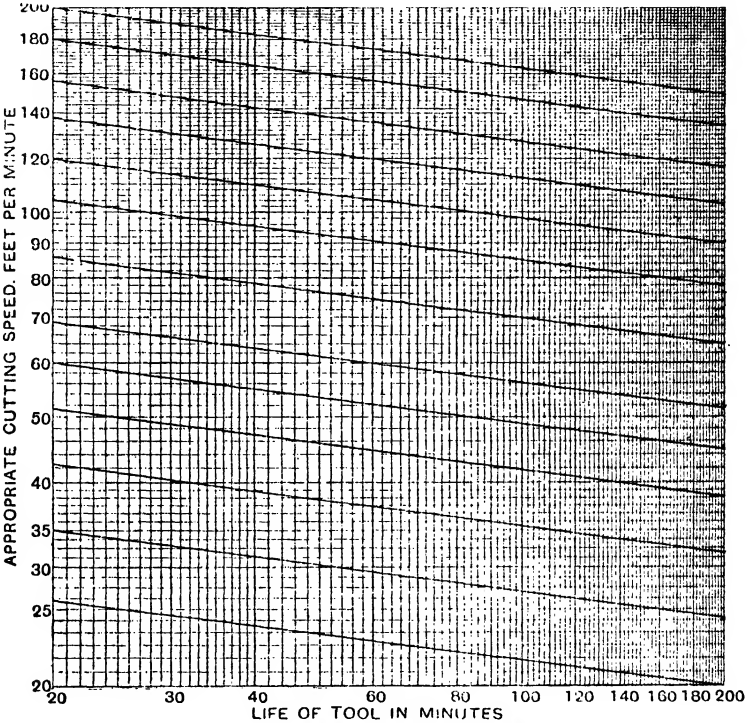


FIG. 23.—Relation between appropriate cutting speed and life of tool.

As an example, assuming a tool under testl asts 40 minutes when cutting at 110 ft. per minute, and it is required to find the speed at which the tool will last 120 minutes, otherconditions remaining unaltered. Trace vertically above 40 minutes until the diagonal line, passing through the horizontal line of 110 ft. per minute, is reached. Follow this diagonal line until vertical over 120 minutes, when it intersects the horizontal line at 96 ft. per minute, which is the speed required.

CUTTING SPEED AND DIMENSIONS OF CUT (STEEL).

The value of the cutting speed varies with the dimensions of the depth of cut and traverse, shape and section of tool, as well as the material cut.

The cutting speeds for the various cuts and traverses given in fig. 24 correspond to a 2-hours' life of tool (which is considered to be the most economical life) when cutting a steel having an ultimate tensile stress of 38 tons per sq. in. aud without any cooling medium.

The values were obtained when using a $1\frac{1}{2}$ in. square tool having a 60° plan angle, 70° cutting angle, and $\frac{1}{8}$ in. nose radius.

Feeds for turning, drilling, and reaming operations on ebonite should always be high, and about three times those used on brass. This remark does not apply so much to forming operations, when the feeds should be a little less than for brass on account of the fragile nature of the material.

Fibre.

Red and black fibre can be worked in much the same manner as ebonite as regards feeds, speeds, and the type of tools employed, but diamond tools are not so satisfactory as on ebonite on account of the irregular quality of these materials, for when the stone strikes a bad patch in the rod or tube there is a tendency to dislodge the diamond or sapphire. Drills, which should be of high-speed steel, should be fed at a higher rate than for ebonite, as should also reamers. The 1 to 4 solution of paraffin and lard oil will be found a good lubricant.

. Erinoid, galalith, and other lactic base compounds are best machined when run at a high speed similar to that for ebonite, but, owing to the fact that these materials do not generate heat to the same degree as ebonite or fibre, they do not require any lubricant, and should be run dry. High-speed steel tools should be used, and wherever it is possible to employ a diamond tool for either boring, turning, or forming, it will be found an advantage to do so. The same points in regard to the type of tools apply to erinoid and galalith as in the case of ebonite, whilst the feeds are approximately the same.

PORTABLE PNEUMATIC TOOLS.

Riveting Hammers.

Pneumatic riveting hammers are now recognised as thoroughly practical, and efficient tools for dealing with any class of work which was formerly done by hand, and rivets from ½ in. up to 1¼ in. diameter can be closed by suitable tools. It takes from 7 to 10 seconds to close a ¾ in. snap-headed rivet.

The air consumption of these tools is generally from 28 to 35 cubic feet of free air per minute, and the pressure may be from 80 to 100 lbs. per square inch, the latter figure being necessary for the heaviest class of riveting, and particularly for steam- or water-tight work.

Caulking Hammers.

The chipping or caulking hammer has now taken its place as an indispensable every-day tool, without which present rates of output could not be maintained. The average chipping hammer will remove mild steel at the rate of 3½ to 4 oz. per minute, and the air consumption is from 17 to 20 cubic feet of free air per minute: pressure from 80 to 100 lbs. per square inch.

Drills.

A pneumatic drill of average size can be relied upon to drill in mild steel plate 1 in. diameter holes at the rate of 1 in. to 1½ in. deep per minute, and larger and smaller holes in proportion, the air consumption varying from 30 cubic feet in the average drills to 50–60 in the larger tools. Usual air pressure 80 lbs. per square inch. Reamering and tapping are also among the varied uses to which these tools can profitably be put. Pneumatic drills are now available in a large range of sizes, from a tool weighing 8 lbs. for drilling $\frac{3}{16}$ in. holes up to a weight of 85 lbs. and a capacity for 3 in. drilling and tapping.

Pneumatic Tools for Salvage Work.

The chipping hammer and drill mentioned above can be fitted for use under water, for ship repairs, dock construction, and other similar work, and are already very largely used in this connection.

Air Mains and Receivers.

Air Mains for the distribution of compressed air are generally made of wrought-iron piping of steam quality, and should be so arranged that the rate of flow will not very much exceed 50 feet per second. At intervals along this main, or wherever tools are likely to be employed, a tee piece should be inserted which will take a connection for the flexible rubber hose which joins up to the tools. An important point to take care of is the provision of adequate means for draining any moisture from the air receiver or the pipe lines themselves, and it is advisable to arrange for the discharge of the compressor to deliver the air at the top of the receiver, and to connect the air main lower down. This assists in depositing any moisture or oil which may be in the air as delivered from the compressor. Efficient separators are available for the removal of water and oil from the compressed air. It is advisable to connect the separator in the pipe line as near as possible to the pneumatic tool to be operated.

TABLE GIVING THE THEORETICAL VOLUME OF EQUIVALENT FREE AIR, IN CUBIC FEET,

that will flow per minute at various pressures through straight pipes of different diameters, each 100 ft. long, without any reduction of pressure.

Initial and Terminal Gauge Pressure, Lbs.	Nominal Diameters of Pipes in Inches. 1	1½	2	2½	3	3½	4	5	6	7	8	10	12	15	18	20	22	24	Initial and Terminal Gauge Pressure, Lbs.
10	24·05	75·67	153·0	244·7	433·3	636·4	883·8	1585	2542	3701	5134	9212	14530	22530	36280	48180	61870	77880	10
20	28·48	89·54	117·9	288·8	512·9	753·0	1045	1871	3006	4381	6076	10900	17210	26640	42940	57000	73220	92150	20
30	32·35	101·6	202·0	329·1	582·5	855·0	1187	2124	3415	4975	6900	12380	19540	30270	48750	64760	83160	104630	30
40	35·82	112·5	223·5	364·2	644·7	946·4	1314	2352	3779	5507	7635	13700	21620	33500	53960	71640	92010	115800	40
50	**38·96**	**122·4**	**243·4**	**396·3**	**701·5**	**1030**	**1430**	**2558**	**4114**	**5993**	**8312**	**14910**	**23520**	**36450**	**58730**	**74300**	**100150**	**126030**	**50**
60	41·83	131·4	261·1	425·4	752·9	1105	1535	2747	4416	6433	8920	16000	25260	39120	63040	83660	107470	135240	60
70	44·53	139·9	278·0	452·9	801·8	1176	1634	2925	4701	6848	9499	17030	26900	41660	67110	89190	114420	144000	70
80	47·08	147·9	294·0	478·8	847·6	1244	1728	3091	4971	7240	10040	18000	28430	44050	70960	94180	120970	152230	80
90	49·54	155·6	309·3	503·8	891·8	1307	1817	3253	5230	7619	10560	18940	29920	46340	74660	99100	127300	160190	90
100	**51·88**	**163·0**	**324·0**	**527·5**	**933·8**	**1370**	**1904**	**3407**	**5477**	**7979**	**11050**	**19850**	**31330**	**48530**	**78190**	**103770**	**133300**	**167720**	**100**
110	54·10	169·9	337·8	550·1	973·9	1429	1985	3552	5712	8320	11530	20690	32670	50610	81540	108220	139000	174920	110
125	7·15	179·5	356·8	581·3	1028	1510	2097	3754	6034	8789	12180	21860	34520	53470	86140	114430	146850	184800	125
150	62·10	195·1	387·8	631·7	1117	1641	2280	4080	6558	9553	13240	23760	37520	58120	93610	124280	159600	—	150
175	66·71	209·6	416·5	678·4	1201	1762	2448	4382	7044	10250	14220	25520	40300	62420	100520	133460	—	—	175
200	**70·93**	**222·9**	**443·0**	**721·4**	**1277**	**1874**	**2603**	**4659**	**7489**	**10900**	**15120**	**27140**	**42850**	**66370**	**106800**	**141910**	—	—	**200**
250	78·70	247·3	491·4	800·4	1416	2079	2889	5170	8309	12100	16780	30110	47550	73640	118620	—	—	—	250
300	85·85	269·9	536·3	873·4	1546	2269	3152	5642	9063	13210	18320	32870	51890	80380	—	—	—	—	300
400	98·55	309·9	615·7	1002	1775	2606	3619	6477	10400	15160	21020	37730	59580	92270	—	—	—	—	400
500	109·7	345·1	685·3	1116	1976	2902	4031	7215	11590	16890	23420	42020	66340	102750	—	—	—	—	500
600	**119·9**	**377·3**	**749·5**	**1220**	**2160**	**3172**	**4405**	**7882**	**12670**	**18460**	**25600**	**45940**	**72520**	—	—	—	—	—	**600**
700	129·3	406·6	807·6	1315	2328	3419	4748	8496	13650	19890	27590	49500	78150	—	—	—	—	—	700
800	138·1	434·1	862·0	1404	2486	3650	5070	9072	14580	21240	29450	52840	—	—	—	—	—	—	800
900	146·3	460·0	914·4	1489	2636	3871	5373	9618	15460	22520	31230	56040	—	—	—	—	—	—	900
1000	154·3	485·1	963·5	1569	2778	4079	5664	10130	16290	23730	32920	59060	—	—	—	—	—	—	1000
1100	**161·3**	**507·1**	**1008**	**1641**	**2906**	**4264**	**5924**	**10590**	**17040**	**24820**	**34430**	—	—	—	—	—	—	—	**1100**
1200	168·6	530·3	1053	1716	3037	4457	6193	11080	17810	25940	35980	—	—	—	—	—	—	—	1200
1300	175·7	552·2	1097	1787	3163	4643	6448	11540	18550	27030	—	—	—	—	—	—	—	—	1300
1400	182·2	572·4	1137	1853	3280	4814	6686	11970	19230	28020	—	—	—	—	—	—	—	—	1400
1500	**188·2**	**591·5**	**1175**	**1914**	**3389**	**4973**	**6908**	**12360**	**19870**	—	—	—	—	—	—	—	—	—	**1500**

(The Consolidated Pneumatic Tool Co.)

Table Showing Loss in Air Pressure, in Pounds, Due to Friction in Pipes 100 Ft. in Length.

Gauge Pressure at Entrance to Pipe, 100 lbs.

Pipe Diameter.	Cubic Feet Free Air delivered per Minute.																												
Ins.	25	50	75	100	125	150	200	250	300	350	400	500	600	700	800	900	1000	1200	1500	1800	2000	2500	3000	3500	4000	5000	6000	7000	8000
1	0·3	1·18	2·66	4·75	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1¼	0·08	0·33	0·73	1·3	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1½	—	0·13	0·29	0·6	0·8	1·15	2·04	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
2	—	0·03	0·07	0·12	0·18	0·26	0·47	0·73	1·05	1·43	1·87	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
2½	—	—	—	0·03	0·05	0·08	0·14	0·21	0·31	0·42	0·54	0·85	1·22	1·66	2·17	—	—	—	—	—	—	—	—	—	—	—	—	—	—
3	—	—	—	—	—	0·03	0·05	0·08	0·11	0·15	0·2	0·31	0·45	0·61	0·8	1·01	1·25	1·8	—	—	—	—	—	—	—	—	—	—	—
3½	—	—	—	—	—	—	—	—	0·05	0·07	0·09	0·14	0·21	0·28	0·37	0·46	0·57	0·82	1·28	—	—	—	—	—	—	—	—	—	—
4	—	—	—	—	—	—	—	—	—	—	0·05	0·07	0·1	0·14	0·18	0·23	0·28	0·41	0·64	0·92	1·13	1·77	—	—	—	—	—	—	—
4½	—	—	—	—	—	—	—	—	—	—	—	0·04	0·06	0·08	0·1	0·13	0·16	0·23	0·35	0·51	0·63	0·98	1·41	—	—	—	—	—	—
5	—	—	—	—	—	—	—	—	—	—	—	—	0·03	0·04	0·06	0·07	0·09	0·13	0·2	0·29	0·36	0·56	0·81	—	—	—	—	—	—
6	—	—	—	—	—	—	—	—	—	—	—	—	—	—	0·02	0·03	0·04	0·05	0·08	0·11	0·14	0·22	0·32	0·43	0·56	—	—	—	—
7	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	0·02	0·02	0·04	0·05	0·06	0·1	0·14	0·2	0·25	0·4	0·57	—	—
8	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	0·02	0·03	0·03	0·05	0·07	0·1	0·13	0·2	0·29	0·39	0·51
10	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	0·01	0·02	0·02	0·03	0·04	0·06	0·09	0·13	0·16
12	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	0·01	0·01	0·02	0·02	0·04	0·05	0·06

(*The Consolidated Pneumatic Tool Co.*)

Air Receivers should be fitted with a manhole, in order that their interior condition may be examined from time to time, as it is possible for corrosion to take place, or on the other hand, if the lubrication of the compressor is not carefully attended to, a large deposit of oil will be formed, which may result in an explosive mixture being present in the receiver.

Air should not be throttled at connections. It is a good plan to introduce water traps in suitable places in a pipe system. Strainers should be carefully watched, they are liable to get choked; they may be cleaned by blowing air through them in the reverse direction.

Tubing and Tube Couplings.

To secure good, economical service, air hose must not kink. Kinking not only shuts off the supply of air at the time, but it creates a permanent injury in the wall of the hose, which later develops into a leak. To prevent accidents of this character, the thickness and number of plies must be proportioned to the size of the hose; the rubber must be of good quality; and the inner tube, fabric, and outer cover must be properly proportioned to each other

Sometimes customers specify wire winding as a protection against outside wear. The practice is not recommended. It makes the hose heavy and hard to handle. Once bent, the hose is hard to reshape; and a wire covering is more costly than the thick rubber cover recommended as the best protection for the outside of the hose.

A moulded hose is advisable for any length over 50 ft. This type of hose eliminates extra couplings, which retard the flow of air and decreases the pressure at the tool. This is an important feature. It has been found that a 15-lb. increase in pressure can make a 37 per cent. difference in the amount of work accomplished by the tool. The tool being designed to give its maximum output at a given pressure the efficiency rapidly falls with drop in pressure.

The pressure which the hose will stand is inversely proportionate to its diameter. A 1-in. 6-ply hose may have a bursting pressure as high as 800 lbs., while a 2-in. 6-ply hose made of the same materials would burst at 650 lbs. Hence, care always should be taken to specify greater number of plies for the larger sizes.

Flexible metallic tubing is being largely used for pneumatic tools. As the tubing withstands crushing and cannot kink, it is well adapted for conveying compressed air to pneumatic drillers and other machines, but care must be taken not to strain any part by bending at too small a radius, otherwise the metallic coil of the tube will become brittle and liable to burst.

Care of Pneumatic Tools.

(1)

Unless pneumatic tools are kept clean, well lubricated, and limited to the uses for which they are intended, they will not give the service for which they are designed. Wherever possible, the air supply should be drawn from some point outside the building where the air is most apt to be clean, cool and dry. In any event, a strainer should be placed over the exposed end of the inlet pipe. Such a strainer can be easily made by covering the opening with wire netting over which one or two layers of muslin have been placed. A strainer should also be placed in the pipe lines leading directly to the pneumatic tools, and a strainer inserted between the pneumatic hammers and the hose nipple.

Lubrication.—Valves and pistons for both hammers and drills require a light machine oil, and as the compressed air which comes in direct contact with these parts has a tendency to drive the lubricant out through the exhaust, it is advisable to oil such parts freely and often, about once every hour while the tool is in constant use. Such portions of the tools as the gear and crank cases to which the compressed air does not have direct access are best lubricated with grease. This grease may be forced into the crank cases through the dead air handle by means of a squirt gun or syringe. A filling every 10 hours while the drills are in constant use gives the best results. It is a good plan to immerse pneumatic hammers in a bath of benzene or kerosene over-night and then blow them out under pressure the following morning, after which they should be thoroughly lubricated with a light machine oil.

(2)

The greatest damage done to the working parts of air tools may be found in the compressor, or rather in the air delivered by the compressor, for not by any means do all users of air tools appreciate the actual necessity for clean, dry air. Until the advent of the air filter, dirty, gritty, moist air had been the cause of thousands upon thousands of pounds being spent in needless repairs on air tools, and on the compressor itself, especially that of the portable type. Besides seeing that clean air is delivered by the compressor, care should also be taken to keep dirt out of the hose and leaders. Open hose ends are often allowed to lie on dusty, dirty, and even sand-covered floors, and then be hooked up to motors and guns without a thought of the necessity for blowing out before connecting.

Motor and gun valves require regular inspection in order that impending trouble may be caught in time, but with clean air and regular, systematic oiling with the right kind of oil, the time needed for inspection can be reduced greatly. Of course, the utilisation of a known, valued oil is a positive requirement. An oil of good quality will remain for some time in the valves and around the plunger of a riveting or a chipping gun, and guns with the right kind of oil have been known to work perfectly when oiled once in three hours of continual service.

Pneumatic Riveting.

According to information received from one of the leading East Coast shipyards, the best day's work on a shell with pneumatic riveting is 700 rivets of ¾ of an inch diameter, and for hand work, under the same conditions, 430. An average day's work, however, is considerably less than this, and may be taken as being 510 rivets with pneumatic rivets, against 264 with a hand squad, the figures being taken for work on all parts of one ship. According to the *Board of Trade Journal* the following may be taken as indicating the difference in the day's work of a hand and pneumatic riveting squad :—On the shell, with ¾ in. rivets, hand-riveting, 260 to 360 rivets; with pneumatic riveting, 500 to 550; on the deck, with ⅝ in. rivets, hand-riveting 450, pneumatic 700; on deck with ¾ in. and ⅞ in. rivets, hand-riveting 350, pneumatic 550. Again, the pneumatic tool requires fewer men to work it. A hand riveting squad consists of three men and one or two boys. From two of these squads three pneumatic riveting squads can be formed, effecting a saving of 50 per cent. in labour alone. A third advantage of the pneumatic tool is that it gives both the other great advantages with less exertion and consequently less fatigue on the part of the men, providing that the nervous system of the operator is not too greatly affected by the vibration of the tool.

Cost of Compressing Air.

In a large shipyard on the north-east coast the actual cost of compressing 100,000 cu. ft. of free air to 100 lbs. per sq. in. pressure with electric current at 1*d*. per unit worked out as follows :—

	s.	*d.*
Power	20	1
Circulating water	0	3
Labour	6	0¾

(*R. W. Wilson. Paper read before Inst. of Mech. Engs., Feb. 8, 1924.*)

Repairing Worn Parts of Machinery by Deposition of Metal.

Electro-Chemical and Metal Spraying Processes.

Worn parts of machinery of practically any description can be repaired by building up the worn part to its original size, by a process of deposition of metal on the worn surface, and thereby making it suitable for further service.

The correct method of repair is to have sufficient metal deposited on the worn surface to make it over size, and then to reduce to correct size by machining or grinding.

By this process of repair, the worn parts are made equally as good as new, while the cost in many cases is much less than the cost of a replacement and generally takes far less time.

Formerly, the building-up process was by welding or brazing, but this has been largely discontinued, excepting in the case of rough work, because of the danger of heat distortion or deterioration of the surrounding metal.

The more modern method for repairing worn parts of machinery is :—

(1) Electro-chemical deposition of metal.
(2) Metal-spraying process.

Both these methods can be regarded as 'cold process,' where the danger of distortion or other heat-effects are entirely eliminated.

Electro-Chemical Process.

Nickel, chromium, cadmium, copper, lead, or a combination of these materials can be satisfactorily deposited on steel and other metals by the electro-chemical process.

In all cases where worn parts are to be repaired by deposition of metal, it is of the greatest importance that perfect adhesion is obtained between the part under repair and the metal deposited, and it should be noted that adhesion is in no way affected by the thickness of the deposited metal. It has been shown that with the Fescol process of electro-chemical deposition, any required thickness can be applied with safety, close blending of the metals being obtained.

Adhesion is of particular importance when the process of deposition of metal is applied to moving parts of machinery, such as piston rods, pistons, valves, shafts and hydraulic rams, etc., as any movement between the basis metal and deposit would be destructive. Adhesion is of equal importance when nickel and chromium are applied as a protective coating to parts exposed to weather conditions, such as motor-car fittings, or to prevent wear by corrosion and abrasion of gauges and other tool room measuring instruments, pistons, valves, shafts, etc.

Actual tests carried out go to show that when metal has been properly deposited, it is hardly possible to separate the deposit from the original metal, the interlocking or blending of the materials being perfect.

Results of tests for adhesion of nickel deposited on steel by electro-chemical process is given below.

*** The specimens tested were mild steel discs, forming a steel core, with a layer of nickel deposited on the periphery.**

Core Diam. Ins.	Outside Diam. Ins.	Depth. Ins.	Failing Load. Tons.	Tons per Sq. In. Max. Shear Stress Applied.
1·034	1·17	0·25	14·96	18·4
1·035	1·17	0·25	16·17	19·9

NOTE.—In each case examination of the fracture showed that failure occurred by shearing of the steel just inside the nickel coating, and not by failure of the adhesion between coating and the steel.

Microphotographs of specimens shows the blending of materials obtained by the Fescol process, and accounts for the high adhesion results obtained by the physical test.

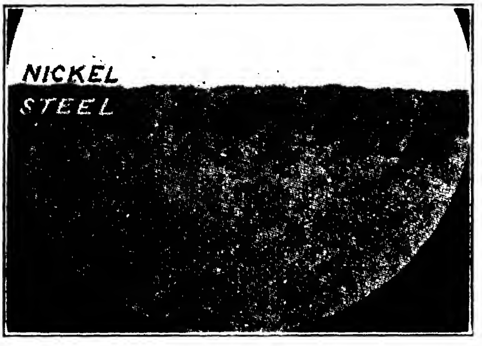


FIG. 26.—Showing interlocking of deposited nickel on steel. Magnification 50.

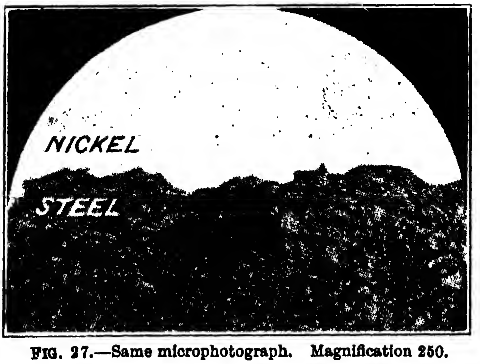


FIG. 27.—Same microphotograph. Magnification 250.

*** Test taken at the National Physical Laboratory on nickel deposited on steel by the Fescol process of electro-chemical deposition of metals.**

While the electro-chemical process of deposition can be applied to all external surfaces of reasonable dimensions, it is more limited when applied to internal surfaces, the ratio of length to diameter has to be taken into account, and also whether the cylinder or other part is open at both ends or only open at one end.

There are also difficulties in filling cracks or cavities by this process, and those should be removed in all cases before any attempt is made to build up any worn or damaged part. It is also necessary that the surface to be treated is clean and free from scale or rust.

The question of cost largely depends on the depth or thickness of the deposit to bring the part to its original size, and for this reason it is usually more economical to have worn surfaces repaired before excessive wear has taken place.

In addition to actually saving worn parts, the deposition of cadmium, chromium and nickel have been found to add considerably to the working life of some parts of machinery, for instance

Cadmium will resist the action of sea water, and the atmosphere.

Nickel resists wear, abrasion and corrosion.

Chromium deposited on printing rolls give a longer life to sharp impressions.

METAL SPRAYING PROCESS.

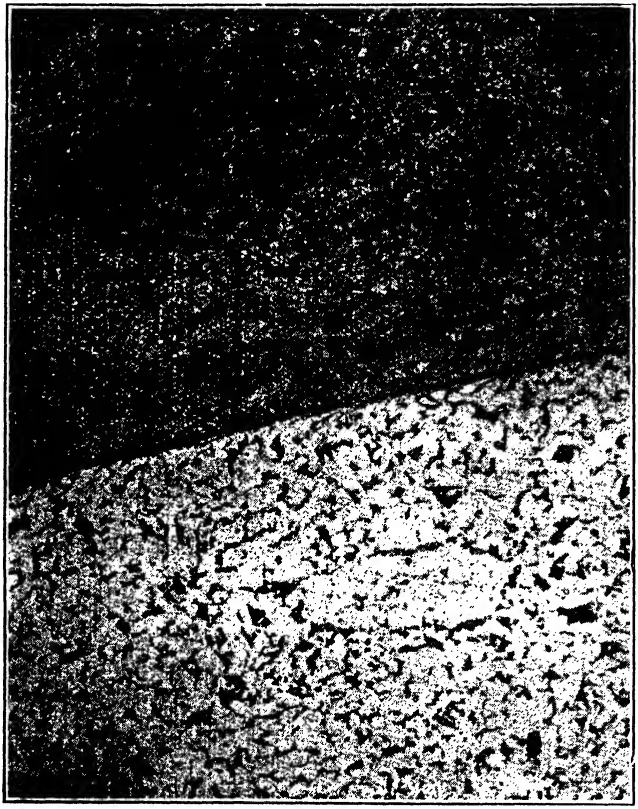


*FIG. 28.—0·09 per cent. carbon steel sprayed using dissolved acetylene, with nitrogen as impelling medium upon a bar of 0·30 per cent. carbon steel. Transverse section at junction of deposit and bar. Etched. × 150.

[1] R. B. Sillifant, British Oxygen Co. Ltd., paper read before Iron and Steel Inst., London, 1937.

Any metal capable of being drawn to a wire and melted in an oxy-hydrogen flame can be deposited on to other metal surfaces and also on to many non-metallic materials.

Any of the following metals can be sprayed either as a protective coating or to give a hardened surface to soft metals for wear-resisting purposes. The metals may also be sprayed to a thickness necessary to build up worn parts of machinery for repair work.

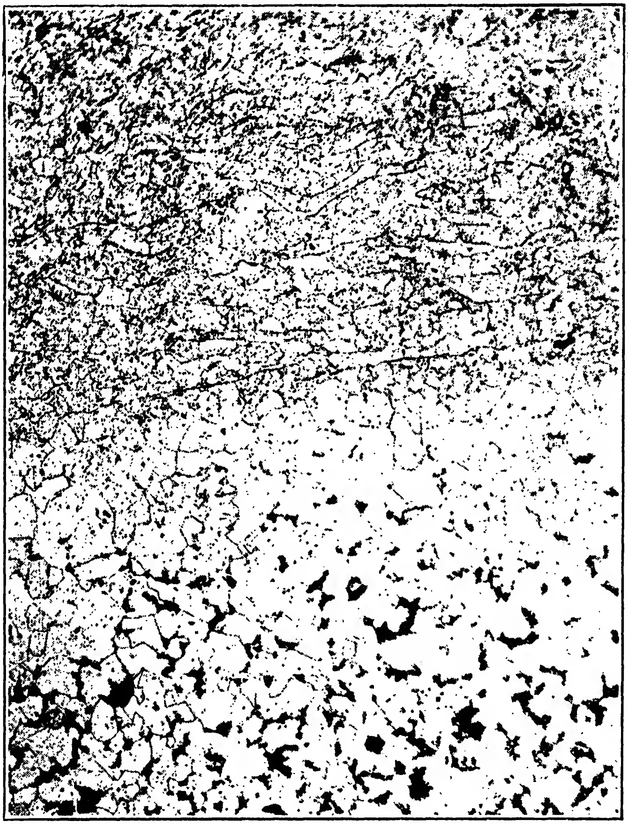


*Fig. 29.—Same as fig. 28, but after heat treatment at 900° C. for one hour. Etched. × 150.

The metals commonly used are :—

Stainless steel, high carbon steel, mild steel, iron, nickel, mono metal, copper, brass, aluminium, cadmium, tin, lead.

Carbon steels that have been deposited by the spraying process are usually capable of being improved with heat treatment, and results of experiments which have been made in this direction are seen in the microphotographs shown above.

* R. R. Sillifant, British Oxygen Co. Ltd., paper read before Iron and Steel Inst., London, 1937.

For the purpose of the experiment, 0·25 carbon steel was sprayed to a thickness of ⅛ in. on a 0·80 carbon steel bar.

Fig. 28 shows a section of the bar etched × 150 before heat treatment, and the line of contact between the deposited metal and the bar metal is clearly seen.

Fig. 29 is a section of the same specimen after heat treatment, etched × 150. The refined structure of the deposited metal, due to the heat treatment, is quite visible.

Mechanical tests have also been taken on specimens having metal collars built up on mild steel shafts by the metal spraying process.

Both tensile and torque tests were taken to determine the degree of adhesion established between the original shaft and the metal deposited in the form of collars by the spraying process.

Tensile or Pull Tests taken on Tensile Machine.*

Specification of Collars Sprayed on	Test Shaft. Diam.	Test Shaft. Length.	Finished Diam. of Collar. Sprayed on.	Finished Length of Collar. Sprayed on.	Pull required to separate Collar from Shaft.
	Ins.	Ins.	Ins.	Ins.	Tons.
Mild steel 0·06 % carbon.	0·99	12½	1·33	1	12·3
High carbon steel 0·8 % carbon.	0·99	12½	1·33		22·45
Foxes stainless steel	0·99	12½	1·33		20·95

Torsional Tests taken on Torsional Testing Machine.*

Test Shaft. Diam.	Finished Collar. Sprayed on. Diam.	Finished Collar. Sprayed on. Length.	Torque at Elastic Limit of Shaft. In./lbs.	Torque at Fracture of Shaft. In./lbs.	Torque at Fracture of Shaft. Tons/sq. in.	Position of Collar on Shaft with Max. Torque.
Ins.	Ins.	Ins.				
¾	1¼	1 13/16	1,700	4,600	24·8	Unmoved

NOTE.—To take the torsional test, the collar, which was built up on the test shaft by spraying process, was slotted and keyed into the adaptor of the torsional testing machine.

The preparation of metal surfaces to receive deposition of metal by spraying, is effectively carried out by sandblasting which should give a clean surface. This method can always be used for parts requiring only a protective coat of about $\frac{5}{1000}$ in. thickness. But in all cases where worn parts of shafts and the like have to be built up and then machined to original dimensions, it is preferable to rough machine the worn part, leaving a surface resembling a roughly cut screw thread.

*** Test report supplied by Metallisation Ltd., Dudley, Worcs.**

In a metal spraying outfit, the spraying pistol is the most important unit. This pistol is operated with compressed air at 50 lbs. gauge pressure. The heating medium is oxygen and a combustible gas. The gas may be either compressed coal gas, dissolved acetylene, or hydrogen.

The following tables give the approximate gas consumption per hour, the amount of metal sprayed per hour, and the speed of the metal wire passing through the spraying pistol, also weight of wire consumed by covering the surface of parts to a given thickness to form a protective coating.

Coal Gas and Oxygen.—Table of pressure and consumption of gases and metals, using coal gas at near constant pressure (calorific values 450 B.Th.U.) and oxygen.

	Wire.					Gases.		
						Coal Gas.	Oxygen.	
Metal.	Diam. Mm.	Speed. Ft. per min.	Metal Sprayed. Lbs. per hour.	Weight of $\frac{1}{1000}$ in. Metal per sq. ft. Ozs.	Area in sq. ft. covered per hour $\frac{5}{1000}$ in.	Press. Lbs. per sq. in.	Press. Lbs. per sq. in.	Cub. ft. per hour.
Tin . .	1·5	19	9·8	0·601	39	24	23	37
Lead .	1·5	21	17·5	0·938	45	24	23	37
Zinc . .	1·0	21	4·8	0·663	17	27·5	25	41
,, . .	1·5	12·5	6·4	0·663	23	27·5	25	41
Cadmium .	1·0	25	7·2	0·730	23	27·5	25	41
Aluminium	1·0	14	1·15	0·211	13	27·5	26	43
Copper .	1·0	8	2·3	0·732	8	27·5	26	43
Brass .	1·0	8·5	2·4	0·701	8·5	27·5	26	43
Bronze .	1·0	8·5	2·4	0·732	8	27·5	26	43
Nickel .	1·0	5	1·4	0·730	4·75	27·5	27	44
Iron . .	1·0	5·5	1·3	0·640	4·75	—	27	44

Dissolved Acetylene.—Where supplies of hydrogen or compressed coal gas are not available, dissolved acetylene in cylinders is adopted as the combustible gas, and the mixtures for the gases are as follows.

Metal.	Acetylene. Lbs./sq. in.	Oxygen. Lbs./sq. in.	Speed. Ft. per min.
Lead . .	11	11	19
Tin . . .	11	11	19
Zinc . .	9	9	24
Aluminium .	13	13	12
Brass . .	22	22	11
Copper . .	23	23	10
Bronze . .	23	23	10
Nickel . .	24	24	6·5
Iron . .	24	24	6·5

Hydrogen and Oxgyen.—Table of pressure and consumption of gases and metals using hydrogen and oxygen.

Metal.	Wire. Diam. mm.	Wire. Speed. Ft. per min.	Wire. Metal Sprayed. Lbs. per hour.	Weight of $\frac{1}{1000}$ in. Metal per sq. ft. Ozs.	Area in sq. ft. covered per hour $\frac{5}{1000}$ in. thick.	Gases. Hydrogen. Press. Lbs. per sq. in.	Gases. Hydrogen. Cu. ft. per hour.	Gases. Oxygen. Press. Lbs. per sq. in.	Gases. Oxygen. Cu. ft. per hour.
Tin . .	1·5	22	11·4	0·601	45	13	38	12	11
Lead . .	1·5	22	18·4	0·938	47	14	58	13	19
Zinc . .	1·5	13	6·6	0·663	24	21	72	18	25
" . .	1·0	24	5·5	0·663	20	21	72	18	25
Cadmium .	1·0	28	8·0	0·730	26	21	72	18	25
Aluminium	1·0	15	1·2	0·211	15	22	74	21	26
Copper .	1·0	10	2·8	0·732	10	23	77	22	27
Brass .	1·0	11	2·9	0·710	11	23	77	22	27
Bronze .	1·0	11	2·9	0·732	10	23	77	22	27
Nickel .	1·0	6·5	1·8	0·730	6	28	90	27	32
Iron . .	1·0	6·5	1·5	0·640	6	28	90	27	32

High Speed Nozzle.—Consumption table using 2 mm. nozzle, air pressure 50 lbs. per sq. in.

Metal.	Oxy-Coal Gas. Oxygen Pressure lbs. per sq. in.	Oxy-Coal Gas. Coal Gas Pressure lbs. per sq. in.	Oxy-Coal Gas. Wire lbs. per hour.	Oxy-Hydrogen. Oxygen Pressure lbs. per sq. in.	Oxy-Hydrogen. Hydrogen Pressure. lbs. per sq. in.	Oxy-Hydrogen. Wire. lbs. per hour.	Oxy-Acetylene. Oxygen Pressure lbs. per sq. in.	Oxy-Acetylene. Acetylene Pressure. lbs. per sq. in.	Oxy-Acetylene. Wire. lbs. per hour.
Tin .	26	29	22·6	23	23	22·2	20	18	24·5
Lead .	26	29	35·0	22	22	41·5	18	17	40·5
Zinc .	27	29	11·9	26	25	12·5	25	24	11·9
Copper .	29	29	7·5	27	26	9·0	26	25	7·5
Aluminium	28	29	2·4	30	30	3·5	26	25	2·8
Iron .	29	29	3·0	36	33	4·0	27	26	4·0

With the metal spraying process, blow holes and cracks in castings can be filled, and where necessary the process can be carried out *in situ*, whether required for protective purposes or for building up worn parts.

The process can also be used for depositing metal on non-metallic materials such as bakelite, wood, paper, porcelain and other substances.

Hardness of Materials.

Different Methods of Determining Hardness.

The test to which a material is generally subjected in order to obtain a measure of its hardness may be of two classes, viz. :—

(1) The resistance offered by the material under investigation, (2) the resistance offered by the material to abrasion or cutting.

Of these, the first finds most extensive favour in practice, and the indenting medium may be a chisel, cone, or sphere. Repeated use on hard materials is inclined to blunt the chisel edge or point of the cone. and the sphere is the agent commonly adopted.

In 1900 Brinell introduced a method of ascertaining the hardness of a material by indenting it with a hardened steel sphere subjected to a predetermined load. The ratio of the load on the sphere to the area of indentation produced gives the hardness figure. The ball commonly used is 10 mm. diameter. The load for the softer materials is usually 500 kilograms, and that for the harder metals 3,000 kilograms.

The load should be applied for at least 30 seconds in order to secure equilibrium between it and the resistance of the material. The diameter of indentation is observed by means of a special microscope fitted with a scale, and from this the depth of indentation can be calculated since :—

$$\triangle = \tfrac{1}{2}(D - \sqrt{D^2 - d^2}),$$

where,

$\triangle$ = depth of indentation; d = diameter of indentation; D = diameter of sphere.

The hardness figure or number N for a load P is given by the expression:—

$$N = \frac{P}{\frac{\pi}{2} D\left(D - \sqrt{D^2 - d^2}\right)} = \frac{P}{\pi D \triangle}.$$

The hardness number is therefore inversely proportional to the diameter of ball and depth of indentation, and for a ball of constant diameter is directly proportional to the load. The hardness numbers for diameters of impressions made by a ball 10 mm. diameter under a load of 3,000 kilograms are given in the table, page 1090.

The depth of indentation may be obtained directly by means of a micrometer microscope by observing the difference between the readings of the datum line on the piston when the parts are brought together and after the load has been applied and removed. This procedure precludes the possibility of including the compression of the piston, sphere, and other parts of the machine beyond the datum line.

As it is difficult to bring the parts together without indenting the specimen to some extent, a small initial load may be applied. Observations of the datum line are then made when the initial load is applied, and also after the full load has been applied, and when the load has been brought back again to its initial value.

By measuring the actual depth of penetration, the error due to the elastic deformation of the sphere (which is included when the diameter is observed) is avoided. This elastic deformation is relatively greatest when the load. and consequently the indentation, is least. As the indentation increases so does the lateral component of the resisting force, and this tends to keep the sphere to its true form.

In testing some cast irons the metal flows from beneath the ball and causes a ridge to be formed which stands proud of the original surface, whilst with bronze and some of the copper alloys the metal is drawn in. In the case of the former the diameter observed is greater than the true value corresponding to the depth of indentation, whilst in the latter case the diameter is less than its true value.

The Shore Scleroscope.

This instrument consists of a small hammer, weighing about $\frac{1}{12}$ of an ounce and fitted with a diamond which is somewhat spherical in shape at the striking end. The hammer is suspended on a stirrup piece fitted to a head on a vertical stand and upon which is also mounted a glass tube about 10 inches long and a graduated scale subdivided into 140 parts. The hammer is released and raised by a pneumatic device and moves within the tube. As the end of the glass tube is made to rest upon the material under investigation, the hammer always falls through a definite height and the measure of hardness is obtained by observing its rebound on the scale. The hammer on striking makes a permanent impression in the material, but as this is very small and irregular in outline it is impossible to measure the depth or diameter with any approach to accuracy. In a recent form a 3 mm. steel ball is soldered into the hammer in place of the diamond.

Abrasion Test.

An early conception was to scratch a smooth surface of the material under investigation with a sharp point of a mineral, a number of minerals being selected to constitute a scale of hardness. In their order of hardness these were :—(1) talc, (2) gypsum, (3) calcite, (4) fluor, (5) apatite, (6) felspar, (7) quartz, (8) topaz, (9) sapphire, (10) diamond. The method is only suitable for materials which differ widely in hardness.

The Sclerometer.

This instrument was developed by Prof. Turner to obtain a measure of hardness of materials which differ but slightly in value. It consists of a diamond carried at one end of a balance lever in a horizontal plane and which rests upon a knife edge fitted into a vertical pillar. The pillar is free to turn by hand, so that the diamond is made to travel across the smooth face of the metal being investigated. The end of the lever carrying the diamond is loaded, and the weight in grams necessary to produce a scratch of standard depth gives a measure of the hardness.

With apparatus similar to the above, some have taken the width of the scratch produced by a diamond under a constant load as the measure of hardness. As the scratch is very minute, it has to be measured by means of a microscope, and the surface must be smooth and polished in order to give a sharp definition. The test is therefore more adaptable to the laboratory than to the workshop.

Drilling Test.

This test can be made with an ordinary sensitive drilling machine having a vertically free spindle. A constant load is applied to the spindle, so that the drill is made to penetrate the material under observation. The revolutions made in drilling a hole of predetermined depth gives a measure of the hardness or tenacity.

With the object of observing the variation in the hardness as the drill proceeds, the machine is fitted with a drum which rotates with the descent of the spindle. A screw, attached to the drum support, carries a nut in which is fixed a pencil. The screw is driven from the spindle, and as the nut cannot rotate the pencil moves in the plane parallel to the drum axis. The combined motion causes a diagonal line to be traced on a piece of paper around the drum, the height and horizontal length of the line being proportional to the depth drilled and revolutions respectively. The slope of the line varies with the hardness, and a soft or hard patch in the material under test is shown on the graph by the line becoming steeper or flatter in the order mentioned at that part. The drill is periodically tested for sharpness on a piece of material of standard hardness. (*Dempster Smith.*)

BRINELL HARNDESS NUMBERS (for load (P) of 3,000 Kg.).

DIAMETER OF BALL (D) = 10 mm; $H = \frac{P}{D^2}\left(\frac{2/\pi}{1-\sqrt{1-(d/D)^2}}\right)$

Diam. of Impression. mm. d	Hardness Number. H	Diam. of Impression. mm. d	Hardness Number. H	Diam. of Impression. mm. d	Hardness Number. H	Diam. of Impression. mm. d	Hardness Number. H
mm.		mm.		mm.		mm.	
2·0	946	3·25	351	4·5	179	5·75	105
2·05	898	3·3	340	4·55	174	5·8	103
2·1	857	3·35	332	4·6	170	5·85	101
2·15	817	3·4	321	4·65	166	5·9	99
2·2	782	3·45	311	4·7	163	5·95	97
2·25	744	3·5	302	4·75	159	6·0	95
2·3	713	3·55	293	4·8	156	6·05	94
2·35	683	3·6	286	4·85	153	6·1	92
2·4	652	3·65	277	4·9	149	6·15	90
2·45	627	3·7	269	4·95	146	6·2	89
2·5	600	3·75	262	5·0	143	6·25	87
2·55	578	3·8	255	5·05	140	6·3	86
2·6	555	3·85	248	5·1	137	6·35	84
2·65	532	3·9	241	5·15	134	6·4	82
2·7	512	3·95	235	5·2	131	6·45	81
2·75	495	4·0	228	5·25	128	6·5	80
2·8	477	4·05	223	5·3	126	6·55	79
2·85	460	4·1	217	5·35	124	6·6	77
2·9	444	4·15	212	5·4	121	6·65	76
2·95	430	4·2	207	5·45	118	6·7	74
3·0	418	4·25	202	5·5	116	6·75	73
3·05	402	4·3	196	5·55	114	6·8	71·5
3·1	387	4·35	192	5·6	112	6·85	70
3·15	375	4·4	187	5·65	109	6·9	69
3·2	364	4·45	183	5·7	107	6·95	68

For other test loads the hardness numbers are proportional to those in the table.

British Standard Tables of Brinell Hardness Numbers.*

(No. 240—1937.) (*Abstract.*)

Definition.

*1. The Brinell hardness number is the quotient of the applied load divided by the spherical area of the impression. The Brinell hardness number is given by the following formula:—

$$H = \frac{P}{\frac{\pi D}{2}\left(D - \sqrt{D^2 - d^2}\right)} = \left(\frac{P}{D^2}\right)\left(\frac{2\pi}{1 - \sqrt{1 - (d/D)^2}}\right)$$

where,

P = Load (in kilogrammes); d = Diameter of impression (in millimetres);
D = Diameter of ball (in millimetres); H = Brinell hardness number.

Note.—The spherical area must be calculated from the average diameter of the impression obtained by taking two readings at right angles, and not from the depth of the impression.

Apparatus.

2. (*a*) The micrometer microscope or other measuring device used shall be capable of measuring the diameter of the impression to $\pm 0\cdot5$ per cent.

Note.—An accuracy of measurement of $\pm 0\cdot05$ mm. ($0\cdot002$ in.) may be accepted for impressions made with a ball of 10 mm. diameter.

(*b*) The balls used in Brinell hardness testing shall be of hardened steel or of some harder material.

Note.—When ordinary hardened commercial steel balls are used, Brinell hardness numbers exceeding 500 may be subject to error owing to permanent distortion of the ball. The error increases with increasing hardness of the material tested. With hardness numbers above 500 care should be taken to see that the balls are considerably harder than the material to be tested.

(*c*) The standard balls and loads used for Brinell hardness testing shall be as follows:—

Diameter of Ball.	Load.			
	$\frac{P}{D^2} = 1$	$\frac{P}{D^2} = 5$	$\frac{P}{D^2} = 10$	$\frac{P}{D^2} = 30$
mm.	Kg.	Kg.	Kg.	Kg.
1	1	5	10	30
2	4	20	40	120
5	25	125	250	750
10	100	500	1000	3000

The limit on the diameter of the balls shall be $\pm 0\cdot0025$ mm. ($0\cdot0001$ in.).

Note.—The same Brinell hardness number is given by tests on the same uniform material with balls of different diameters when the same value of the ratio $\frac{P}{D^2}$ is used.

Test Specimens.

3. (*a*) The centre of the impression shall be not less than two and a half times the diameter of the impression from any edge of the test specimen.

(*b*) The thickness of the test specimen shall be at least seven times the depth (t) of the impression as given by the formula $t = \frac{P}{\pi D H}$ or, alternatively, shall be such that no bulge or other marking showing the effect of the load appears on the side of the piece opposite the impression.

When tests are made on thin specimens, care should be taken to ensure that the under surface of the specimen is smooth and in good contact with a smooth supporting surface of hardened steel.

* By permission of the British Standards Institution.

(c) The surface on which the impression is to be made shall be polished if the diameter of ball used is 1 mm. or 2 mm. If the diameter of ball used is 5 mm. or 10 mm., it is still advantageous to use a polished surface, but, if a high order of accuracy is not required, the surface may alternatively be filed, ground, or smoothly machined.

4. (a) The value of the ratio $\frac{P}{D^2}$ shall be as follows :—

For steel and materials of similar hardness	30
For copper alloys and materials of similar hardness	10
For copper	5
For lead and tin and materials of similar hardness	1

It is advisable that the ratio of the diameter of the impression to the diameter of the ball shall not exceed 0·6.

(b) The load shall be applied slowly and progressively to the test specimen.

(c) The full load shall be maintained for at least 15 seconds when the ratio $\frac{P}{D^2} = 30$ and for 30 seconds when the ratio $\frac{P}{D^2} = 10$, 5 or 1.

Note.—For steel, experiment has shown that an approximate indication of the tensile strength in tons per sq. in. can be obtained by multiplying the Brinell hardness numbers by 0·22. In cold-worked material and in exceptional cases the ratio may differ widely from this figure.

The Specification gives twelve tables with diameters (D) of balls varying from 10 mm. to 2 mm. and loads (P) varying from 1,000 kg. to 4 kg.

Balls for Brinell Tests.

At a meeting (1927) of the American Society for Steel Treating, a paper was read on the use of an iron-carbon-vanadium alloy for balls for making Brinell tests. When Brinelling steels of such hardness as causes ordinary Brinell balls to deform, the alloy recommended for use in testing consists of 2·9 per cent. carbon and 13 per cent. vanadium, the remainder being iron. Balls made from this material, and heat-treated, are found, when tested on steels of approximately 700 Brinell hardness, to flatten only one-fifth as much as the ordinary balls used and only one-half as much as certain special balls.

The Herbert Pendulum Hardness Tester.

This instrument consists of an arched weight of 4 kg. pivoted on a fixed ball of 1 mm. diameter, the ball being of hardened steel, ruby, or diamond. The position of the centre of gravity of the instrument can be varied, but for all standard tests the centre of gravity is 0·1 mm. below the centre of the ball, and the instrument constitutes a very short compound pendulum. A curved spirit tube with bubble and scale graduated from 0 to 100 serves to measure the amplitude of the oscillations. On rocking the instrument the specimen is slightly indented, but the degree of hardness is not measured by the indentation but by the swing of the instrument; the readings are, therefore, expeditiously made.

Time Test.

This is the number of seconds taken by 10 single swings of the pendulum when placed gently on the specimen and caused to oscillate through a very small arc. The 'time hardness numbers' (seconds for 10 swings) vary from 3 seconds on lead to 20 seconds on mild steel, 85 on very hard steel, and 100 on glass.

The pendulum time test, like the Brinell test, measures 'indentation hardness' or resistance to indentation.

Work-hardening Test.

Another test is that which measures the 'work-hardening capacity' of metals, or the increase of hardness which can be induced in them by working them cold. In making this test the Pendulum is placed on the specimen in a tilted position with the bubble at 0 on the scale and is then released. The pendulum swings through a certain arc which depends upon the hardness of the metal, and the position of the bubble on the scale at the end of the first swing measures the 'work hardness' or resistance to working by rolling.

The next operation is to tilt the pendulum and complete the swing by hand. The bubble is then at 100 on the scale and is again released. The effect of tilting the pendulum is to elongate the impression and to harden the rolled surface by cold work. When the pendulum is released the ball rolls back along the hardened surface, and the position of the bubble on the scale at the end of the swing measures the hardness of the work-hardened surface. This process is repeated, the pendulum being tilted alternately to right and to left, and the hardness measured after each successive rolling by the ball until a maximum is reached. The difference between the first reading and the maximum reading measures the work-hardening capacity of the metal. Glass has zero work-hardening capacity. The first reading and the subsequent readings are the same. Manganese steel is shown to be soft by the Brinell test, soft by the pendulum time test, soft by the first reading of the work-hardening test, but after being rolled by the ball it immediately becomes as hard as hardened tool steel, which explains why it cannot be worked.

The time test and the work-hardening test have important applications owing to the ease with which they can be applied to heated specimens. The pendulum equipped with a diamond ball measures the hardness of hot specimens as easily as cold, and the tests are conveniently made while the specimen is actually in the electric furnace.

'Rockwell' Hardness Tester.*

There are several types of 'Rockwell' hardness testing machines designed to cover a wide variety and size of parts to be tested. The standard 'Rockwell' is made in a number of sizes which range in vertical capacity from 3½ ins. to 12 ins. All standard types are open at the sides. Another type of 'Rockwell' tester is known as the 'Universal,' and is made with a bridge-piece supported on threaded columns. It has a horizontal working clearance between the columns of 25 ins., and an effective vertical gap of 20 ins.

In all types the method of measuring the hardness of metals is the same, and, like the 'Brinell, is based on the resistance offered by the material under test to a definite applied load.

The 'Rockwell' hardness number is computed on the additional depth to which a test point is driven by a heavy load beyond the depth to which the same test point has been previously driven by a definite light load. This dual operation is accomplished without moving the piece being tested. The light or minor load is applied first and is immediately followed by the major or heavy load, the hardness number being automatically registered on a dial indicator mounted at the front of the machine. The whole operation can be carried out in from 7 to 10 seconds.

In order to obviate errors in measurement of depth of impression which otherwise might be occasioned by surface imperfections of the part tested, the 'Rockwell' machine is designed to make two superimposed impressions, the first with a load of 10 kgs., and the second with a load of 100 kgs., operating on a $\frac{1}{16}$-in. diam. steel ball penetrator, or 150 kgs. when the diamond test point penetrator is used.

There are thus two types of penetrators, either of which may be used on any size of machine. They are designated Penetrator 'B' and Penetrator 'C.' The first-mentioned 'B' is a hardened steel ball $\frac{1}{16}$-in. diameter, mounted in a special chuck to readily permit replacement or change of balls; this is usually used for testing unhardened steels and the softer metals such as bronze, brass, cast iron and many of the other alloys now in use. The standard major load for the 'B' penetrator with the $\frac{1}{16}$-in. steel ball is 100 kgs. The penetrator 'C' is a Brazilian diamond, termed 'Conical Brale.' The diamond is ground and polished very accurately to a cone of 120° angle and has a mechanically lapped spherical point of microscopic but exact radius, and is patented. It is used for testing hard steel whether tempered or not, and, as it makes a deeper impression than the ball, it is not used on sheet metal because the depth of the impression made would be out of proportion to the thickness of the part being tested. The standard major load for use with the 'C' penetrator diamond cone is 150 kgs.

Only a few seconds is taken to change over from 'B' penetrator to 'C' penetrator, or from 100 kgs. to the 150 kgs. major loads, also the weights which constitute the load are so formed as to preclude any possibility of error in applying them.

The diamond cone is used on very hard steel because of the possibility of the $\frac{1}{16}$-in. steel ball taking a flat, even with 100 kg. load. Also the diamond cone is more sensitive and retains its shape with a load of 150 kgs., and because its shape and mpression is deep in proportion to its diameter.

The 'Rockwell' machine operates on a system of dead weights and levers, and to ensure the load being applied slowly and smoothly to the penetrator, an adjustable oil dash pot is fitted so that the speed at which the load is applied can be governed as desired.

* See also B.S.S. No. 891—1940.

A direct-reading dial is fitted at the front of the machine. It is graded in two colours for the 'B' and 'C' scales. The 'B' scale in red applies with the 'B' penetrator and 100 kg. load. The 'C' scale in black applies with the diamond cone penetrator and 150 kg. load. Both by letters and colours the 'B' and 'C' scales are distinguished on the dial and all notation of readings should always be written down with the prefix 'B' or 'C.' This applies to all ordinary 'Rockwell' hardness testers and must be rigidly adhered to.

A later design of machine, which may be regarded as a special purpose machine, is known as 'Rockwell' Superficial Hardness Tester. It has all the essential features of the ordinary 'Rockwell,' but has been designed for use where the penetration into the work must be kept shallow, while its sensitivity is preserved. It is intended for use in testing nitrided steel, safety razor blades, and brass, bronze and steel sheet. This machine operates with much lighter minor and major loads, and has a more sensitive depth measuring system. Instead of the 10 kg. minor load and the 100 and 150 kg. major loads of the ordinary 'Rockwell,' the 'Superficial' applies minar load of only 3 kgs., and major loads of 15, 30 and 45 kgs. Whereas hardened tool steel is penetrated to a depth of about 0·0035 in. on an ordinary 'Rockwell C' scale test, the superficial machine on metal of the same hardness, and the 30 kg. major load, would only penetrate to about 0·0010 in. depth.

In recording readings obtained on the superficial hardness tester, the value in kilograms of the major load used should first be written, and second, the letter N if Brale N is used as penetrator, or T if the $\frac{1}{16}$-in. diam. steel ball is used as penetrator, and third, the dial reading. The scale pan alone applies 15 kg. major load, and each of the weights 15 kg. more, the load therefore may be 15, 30 or 45 kgs.

The established prefix symbols for the 'Superficial' tester are:

For Brale N.	For $\frac{1}{16}$-in. Ball.
15N	15T
30N	30T
45N	45T

APPROXIMATE COMPARISON OF HARDNESS SCALES.

(B.S.S. No. 860—1939.)

Foreword.

The most widely employed methods of measuring hardness are the 'Brinell,' 'Diamond Pyramid,' and 'Rockwell' tests. There are individual preferences for using these, but cases are increasing where the same material is tested in different works or laboratories by the three different methods. There has, in consequence, been a general demand for some means of correlating the most used A, B and C Scales of the Rockwell test with those of the Brinell and Diamond Pyramid tests.

Investigations show that there can be no general theoretical relationship between these scales, and empirical formulæ devised from experiments only hold closely for materials of approximately similar composition and in a given condition. *Variations* in the empirical relationships result if the conditions as regards composition, heat-treatment and cold work differ appreciably, and also if different loading ratios are selected in the Brinell test. On the other hand, groups of materials similar as regards composition, cold work, etc., when tested by any of the above methods, may give fairly close comparisons.

The following table is therefore a general approximation. It is issued solely as an indication of the order of the relationship between the three systems of Hardness Readings, and must not be used as a standard for the conversion of hardness values given on one scale in any British Standard to those of another scale.

The Diamond Pyramid Scale has been taken as the basis of reference and the most probable comparative values have been adopted from published experimental results.

This Table requires reference to the following British Standards :—

No. 240. Methods and Table for Brinell Hardness Numbers.

No. 427. Tables of Diamond Pyramid Hardness Numbers.

No. 891. Direct Reading Hardness Test (Rockwell Principle).

DIAMOND PYRAMID SCALE (B.S. 427—1931).	BRINELL (STEEL BALL) SCALE. (B.S. 240—1937).		DIRECT READING HARDNESS TEST (ROCKWELL PRINCIPLE). (B.S. .‡)					
			C. SCALE. 150 kg. Diamond Cone.		A. SCALE. 60 kg. Diamond Cone.		B. SCALE. 100 kg. $^1/_{16}$ in. Steel Ball.	
(1)	Variations.* (2)	Adopted Value.† (3)	Variations.* (4)	Adopted Value.† (5)	Variations.* (6)	Adopted Value.† (7)	Variations.* (8)	Adopted Value.† (9)
20	15–25	20						
40		40						
60		55						
70		65						
80		75						
90		85						
100	80–100	95				43	47–61	54
120		115				47		65
140		135				50		77
160		155				53		83
180		175				56		89
200	175–205	195			58–60	59	93–95	94
220		215				60		97
240		235	18–23	20		61		100
260		255		24		63		
280		275		27		64		
300	280–300	295	27–33	30	65–68	66		
320		310		32		67		
340		325		34		68		
360		345		36		69		
380		360		39		70		

* *I.e.*, range to be expected among individual cases. (See Foreword, paragraph 2.) The variations given apply to the line on which they occur, and intermediate values may be estimated by considering the next set of variations in the same column.

† See Foreword, paragraph 4.

‡ B.S.S. 891.

DIAMOND PYRAMID SCALE (B.S. 427—1931).	BRINELL (STEEL BALL) SCALE. (B.S. 240—1937).		DIRECT READING HARDNESS TEST (ROCKWELL PRINCIPLE). (B.S. .‡)			
			C. SCALE. 150 kg. Diamond Cone.		A. SCALE. 60 kg. Diamond Cone.	
(1)	Variations.* (2)	Adopted Value.† (3)	Variations.* (4)	Adopted Value.† (5)	Variations.* (6)	Adopted Value.† (7)
400	370–395	380	38–42	40	70–72	71
420		395		42		72
440		415		44		73
460		430		45		73
480		445		47		74
500	445–480	460	46–50	48	73–76	75
520		475		49		75
540		490		50		76
560		505		51		76
580		520		52		77
600	515–550	535	52–56	54	75–79	77
620		545		55		78
640		560		56		78
660		570		57		79
680		585		57		79
700	580–620	595	57–61	58	76–80	80
725		605		59		81
750		630		61		81
800			60–64	62	77–83	82
850				63		82
900			63–67	65	78–84	83
950				66		83
1 000			65–69	68		84
1 100				69		85
1 200				70		87
1 250					87–90	88
1 400				71	90–93	91

* *I.e.*, range to be expected among individual cases. (See Foreword, paragraph 2.) The variations given apply to the line on which they occur, and intermediate values may be estimated by considering the next set of variations in the same column.

† See Foreword, paragraph 4. ‡ B.S.S. 891.

See also Descriptive Section XXII, Part I.
Heenan & Froude, Ltd.
Edward G. Herbert, Ltd.
C. Winn & Co., Ltd.

SECTION XXII

PART II

GRINDING, ABRASIVES, AND POLISHING APPLIANCES.

ABRASIVE MATERIALS — NATURAL GRINDSTONES — ARTIFICIAL GRINDSTONES — ABRASIVE WHEELS AND BLOCKS — SELECTION OF WHEELS — SPEEDS OF WHEELS — MOUNTING ABRASIVE WHEELS — WHEEL GUARDS — LUBRICATION — DRY VERSUS WET GRINDING — GRINDING PROCEDURE — CYLINDRICAL GRINDING — EXTERNAL GRINDING — INTERNAL GRINDING — SURFACE GRINDING — GRINDING ALLOWANCES—POLISHING—SAND BLASTING.

Revised and amplified by Alfred B. Searle
(Consultant and Advisor on Abrasive Materials).

GRINDING, ABRASIVES, AND POLISHING APPLIANCES.

The appliances used for grinding and polishing may be divided into two chief groups:—

(*a*) The *abrasive materials*, which may be in the form of powders, liquids, wheels, blocks, ups, etc.

(*b*) The *machines* used in the application of such materials, and commonly known as 'Grinding and Polishing Machines.'

Abrasive Materials.*

The chief *natural* abrasive and polishing materials are sandstone, sand, emery, corundum, diamond dust, pumice, rotten stone, tripoli powder, rouge, lime, whiting, and chalk. The chief *artificial* abrasives are: (i) carbon silicide, which is known by various trade names, such as carborundum; and (ii) fused or crystalline alumina or artificial corundum, which is also known under a variety of trade names such as Alundum. Corindite, etc.

These materials are ground and then sifted so as to ensure the particles being of sizes suitable to the particular purpose for which they are to be used; this being a matter of great importance.

*** For further details see *Abrasives*, by A. B. Searle (Pitman); also *Chemical Industries*, xiv. 3 (1939).**

These abrasive materials may be used in the form of powders or pastes, or suspended in water, oil, or other convenient liquids (as liquid polishes), or they may be made into grindstones, wheels, blocks, etc.

Natural Grindstones.

Natural grindstones are made by cutting blocks of suitable stone to the desired size and shape. The stones used for this purpose are shown by the following tables:—

FOR GRINDING MACHINISTS' TOOLS.

Name of Stone.	Kind of Grit.	Texture of Stone.	Colour of Stone.
Nova Scotia	All kinds, from finest to coarsest	All kinds, from hardest to softest	Blue, or yellowish-grey
Bay Chaleur (New Brunswick)	Medium to finest	Soft and sharp	Uniformly light blue
Liverpool or Meling	Medium to fine	Soft, with sharp grit	Reddish
Yorkshire and Derbyshire grit stone	Medium to fine	Hard to soft, as selected	Biscuit to grey

FOR WOOD-WORKING TOOLS.

Name of Stone.	Kind of Grit.	Texture of Stone.	Colour of Stone.
Wickerolz	Medium to fine	Very soft	Greyish-yellow
Liverpool or Meling	Medium to fine	Soft, with sharp grit	Reddish
Bay Chaleur (New Brunswick)	Medium to finest	Soft and sharp	Uniformly light blue
Huron, Michigan	Fine	Soft and sharp	Uniformly light blue

FOR GRINDING BROAD SURFACES, AS SAWS OR IRON PLATES.

Name of Stone.	Kind of Grit.	Texture of Stone.	Colour of Stone.
Newcastle	Coarse to medium	Hard	Yellow
Independence	Coarse	Hard to medium	Greyish white
Massillon	Coarse	Hard to medium	Yellowish white

FOR PULPING WOOD IN PAPER MANUFACTURE.

Name of Stone.	Kind of Grit.	Texture of Stone.	Colour of Stone.
Grindleford Grit	Coarse	Medium	Biscuit

Maximum Speed of Natural Grindstones.

Except under expert advice, grindstones should not be run at a higher peripheral speed than 3,400 ft. per min. This corresponds to the following speeds of rotation:—

Diameter of Stone. Ft. Ins.	Revolutions per Minute.	Diameter of Stone. Ft. Ins.	Revolutions per Minute.	Diameter of Stone. Ft. Ins.	Revolutions per Minute.
8 0	135	6 0	180	4 0	270
7 6	144	5 6	196	3 6	308
7 0	154	5 0	216	3 0	360
6 6	166	4 6	240		

To True Natural Grindstones.

Drive at a moderate speed and true up with a piece of iron tube. Keep turning the tube, which should be held at right angles, and turn it as the edge grinds away. The stone should be *dry*, not wet.

Artificial Grindstones.

Artificial grindstones are made by converting abrasive powders into a plastic paste, compressing this in moulds of the desired shape and size, and then heating (or cooling) the product until it becomes sufficiently strong for use.

The abrasive materials used are those mentioned on p. 1097. They are united by a bond composed of (i) vitrified or porcelain bond; (ii) sodium silicate; (iii) shellac or similar material; or (iv) magnesium oxychloride. Each bond gives the wheel special advantages for certain purposes and care should be taken when selecting wheels to stipulate the bond (porcelain, silicate, etc.).

Wheels with oxychloride bonds should not be used in contact with water.

Wheels with a porcelain bond are the most resistant and powerful, but they are also the most expensive. Wheels with other bonds are satisfactory for 'mild' work. See British Standard Specifications Nos. 620 and 671—1935.

Abrasive Wheels and Blocks.

The particles of abrasive material are held together by clay or by a mixture of materials in which clay is an important ingredient, the wheels being moulded from a plastic state and afterwards burned in kilns at a temperature of about 1,400° C. In other abrasive wheels the binding materials are shellac, rubber, or silicate of soda.

Abrasive wheels, blocks, etc., are classified according to the nature of the abrasive (corundum, carborundum, emery, etc.), the size of the abrasive particles, the hardness or tenacity of the bond, and the bond.

Much economy, speed, and quality can be obtained by using abrasive wheels in the place of files, especially when the wheels are used on the varied machines to which they are applied. The great profit to be derived from the general introduction of abrasive wheels is due to saving in wages, files, increased output, and improved work.

The *grade* of a wheel is represented by a letter, A and Z being the hardest and the softest. The term 'grade' is confined to the relative hardness of the wheel or block as a whole, and does not refer to the hardness of the abrasive particles. This distinction is important, because the cutting or abrasive power depends quite as much on the bonding material as on the abrasive particles.

No wholly satisfactory method of grading abrasive wheels, etc., has yet been devised, various mechanical devices of testing having proved unsatisfactory. The method adopted by manufacturers consists in gouging the wheel with a tool resembling a screw-driver, the 'feel' indicating the grade with fair accuracy to an experienced tester. It is, however, almost impossible to compare accurately the grades of abrasive wheels made with different bonds, as they do not work equally satisfactorily with any material.

The term '*grit*' is used to indicate the sizes of the abrasive particles present, and it is usually expressed by figures indicating the number of meshes per linear inch in a sieve through which the abrasive particles will pass completely. Thus, in a 60-grit wheel or block the abrasive particles will pass completely through a sieve having 60 holes per linear inch. It should be observed that the grit only indicates the largest particles, and has no reference to their average diameter nor to the proportion of much smaller particles present. It is understood, however, that there is only a small proportion of particles sufficiently fine to pass through a sieve which is represented by the next grit figure.

Grit Sizes of Grinding Wheels.

Class of Work.	Emery.	Alundum.	Corundum.	Carborundum.
Glass and marble polishing	150	200	240	—
Fine hardened steel work	40–80	46–100	50–120	60–150
Sharpening milling cutters	40–50	46–50	46–60	50–80
General machine shop use	20–24	20–30	20–36	24–40
Brass castings	16–20	20–30	24–36	30–40
Agricultural implements	14–20	16–24	16–30	16–36
Rough castings	10–16	12–16	12–20	14–36
Marble	—	—	—	2–4

'Alundum' and 'carborundum' are both 'trade mark' names.

Wheels for Different Materials and Operations.

Material.	Operation.	Abrasive.	Grit.	Grade.	Bond.
Aluminium	Finishing	Carbide	24–30	P–R	Vitrified
Bits, Auger	"	Alumina	60–80	E–K	Rubber
" Woodworking	"	"	36–80	M–P	Vitrified
Brass Castings, large	Smoothing	Carbide	16–20	H–K	"
" " small	"	"	36–50	H–I	"
Bronze Castings, large	"	"	16–20	H–I	"
" " small	"	"	36–50	H–J	"
Cams, unhardened	Roughing	Alumina	16–24	D–K	"
" hardened	"	"	24–40	K–P	"
" hardened	Finishing	"	36–50	G–K	"
Copper	Roughing	Carbide	16–30	3–4	Shellac
"	Finishing	"	60–80	5–8 or L–P	" Vitrified
Dies, Cast Iron	Smoothing	"	46–60	L–P	"
" Chilled Iron	"	"	24–30	H–J	"
" Steel	Roughing	Alumina	36–40	H–K	"
" "	Finishing	"	60–80	P–S	"
Edge Tools	Sharpening	Al. or Carb.	36–46	M–P	Shellac
Iron, Cast	General	Carbide	16–20	G–K	Vitrified
" "	Surfacing	"	24–30	K–M	"
" Malleable Castings	"	Alumina	10–20	G–H	"
" Wrought	Roughing	"	16–24	F–H	"
Knives	Finishing	"	60–100	4 or J	Sh. or V.
Milling Cutters, Carbon Steel	Grinding	"	50–60	M–N	Vitrified
" " High Speed Steel	"	"	36–40	M–O	"
Piston Rings	Finishing	Carbide	36–40	K–P	"
Saws, metal (hot)	Grinding	Alumina	360	K	"
" " (cold)	"	Al. or Carb.	40–60	H–P	"
" wood (band)	"	Alumina	360	K–L	"
Steel Castings, large	Smoothing	"	8–12	E–G	"
" " small	"	"	10–16	E–G	"
Stove parts	Surfacing	Carbide	60–80	H–K	"

NOTE.—These figures are only intended as a rough guide. It is usually desirable to consult the manufacturers of the wheels (see p. 1101).

The *suitability* of an abrasive wheel or block for any given purpose depends on the grade or hardness of the wheel as a whole, on the grit or size of the abrasive particles, and on the *bond* which unites them together.

For grinding steels and the harder materials, *i.e.* materials with a high tensile strength, wheels or blocks composed of an aluminous abrasive (corundum emery) are preferable. For cast iron, chilled iron, brass, soft bronzes, aluminium. copper, granite, marble, leather, wood, and other materials of lower tensile strength, wheels or blocks composed of a carbide abrasive (carborundum, crystolon, etc.) should be used. The binding agent which unites the particles of abrasive also determines the suitability of the wheel or block for any given purpose. Thus—

(i) Rubber-bonded abrasives are very tough and resistant, and so are specially suitable where a bond of exceptional strength is required in conjunction with a thin wheel or ring, as in the case with various cutting operations, such as slotting iron plates, sawing stone, cutting glass and saw teeth, or sharpening milling cutters of awkward shape, as well as for cutting metals, grinding chilled iron rolls, and for other work where a fine finish is required. Shellac-bonded wheels are too soft for this purpose. They are also used where it is necessary that the shape of the abrasive article should be retained as long as possible. This is particularly the case with the thin cutting discs used by watchmakers, dentists, etc.

(ii) Shellac-bonded wheels are used for cutter and reamer grinding, pointing drills, sharpening knives, saws, and other edge tools, and for similar purposes as rubber-bonded wheels, but as they are more elastic they can be used to give a finer finish. They will not stand high temperatures, but can be used for wet or dry grinding. They are less porous and less free cutting than wheels with a vitrified bond. They are singularly free from 'glazing,' as the baked shellac has lost the stickiness characteristic of the natural material. *Bakelite* is a better bond than shellac.

(iii) Silicate wheels are very close in texture, and as they cut freely they are very useful for grinding tools and cutlery.

(iv) Abrasive articles made with a vitrified bond are characterised by a high porosity, a great resistance to heat and acids, and great freedom in cutting. They may be used for almost any purpose if care is taken to select them of suitable grit and grade, though when very thin cutting wheels are required it is often preferable to use a shellac—or rubber-bond may be

In the **Selection of an Abrasive Wheel**

the most important points to be kept in mind are *suitability for the particular class of work*, and *safety from accident*.

It is obvious to everyone that a wheel specially manufactured for fettling iron castings is not the one to be used for needle pointing, but few people, except those connected with the trade, have any conception of the number of kinds or classes into which such wheels have been divided and subdivided, each main group (designated by symbols, B, S, SS, G, R, M, etc.) being varied in hundreds of ways to suit the work.

The general practice is to state the purpose for which the wheels are required and leave the selection to the judgment of the makers; it is to their interest to give satisfaction. The other particulars to be given when ordering are—quantity required, diameter, thickness, and size of centre hole.

For information on the *dimensions* of abrasives wheels and mode of attachment, see British Standard Specification No. 620—1935, also pp. 1102–1105.

Speed of Wheels.*

This varies somewhat according to the make and the purpose for which they are used, but no abrasive wheels should be operated at a speed exceeding that recommended by the manufacturer thereof. The following speeds are equivalent to a peripheral speed of 5,000 ft. per minute :—

SPEED TABLE.

Diam. of Wheel in Ins.	Speed. Revs. per Min.	Diam. of Wheel in Ins.	Speed. Revs. per Min.	Diam. of Wheel in Ins.	Speed. Revs. per Min.
1	19,099	18	1,061	42	455
2	9,549	20	955	44	434
3	6,366	22	868	46	415
4	4,775	24	796	48	397
5	3,820	26	733	50	383
6	3,183	28	683	52	369
7	2,728	30	637	54	354
8	2,387	32	596	56	341
10	1,910	34	561	58	330
12	1,592	36	531	60	319
14	1,364	38	503		
16	1,194	40	478		

The peripheral speed of a grinding wheel—which is a function of its diameter and the number of revolutions per minute which the wheel makes—has a marked effect on the apparent hardness of the wheel. Hence, the hardness or grade of an abrasive wheel or cylinder appears to vary with the speed at which it rotates. Thus, by running a soft wheel above its normal speed it appears to become harder, and by reducing the speed of a hard wheel it appears to become softer. Hence, by varying the speed within reasonable limits a wheel which is normally too soft or too hard may be used satisfactorily.

As a wheel wears, its peripheral speed is reduced and the wheel appears to soften, but by increasing the number of revolutions per minute so that the original peripheral speed is obtained the wheel appears to regain its former hardness. Many workers suppose that the faster the work revolves the greater will be the output, and so they use very hard wheels which require maximum driving power and cause a maximum wear of the machine. Yet, in most cases, they produce work of inferior quality which requires a longer time and greater effort to produce than if softer wheels and slower work speeds were used.

The following speeds are typical, but should be modified to suit particular cases :—

Application.	*Peripheral Speed in Feet per Min.*
Cylindrical grinding	5,500 to 6,500
Snagging and general off-hand grinding on bench and floor stands .	5,000 ,, 6,000
Surface grinding	4,000 ,, 5,000
Knife grinding	2,000 ,, 3,000
Hemming cylinders	2,100 ,, 2,400
Tool grinding	3,000 ,, 4,000†
Vertical surface grinding machines	4,000 ,, 4,500
Elastic and rubber cut-off wheels	9,000 ,, 12,000
Polishing wheels	about 7,000

* Some of the following tables and recommendations are taken from a booklet issued by the Royal Society for the Prevention of Accidents, 52, Grosvenor Gardens, London, S.W. 1.

† Some large users of grinding wheels regard as maximum peripheral speeds for machinist's tools 9–1,000 ft. per min., and for carpenter's tools 550–600 ft. per min.

For speeds higher than the above, slightly softer grades should be used to off-set the harder cutting action; for lower speeds slightly harder grades than would be ordinarily supplied should be used.

It is impossible to state exactly the speed at which any article or part to be ground should rotate. It is largely a matter of experiment, as the speed should be suited to the abrasive and to the nature of the material to be ground. With Norton cylindrical grinders a peripheral work-speed of 60 to 80 ft. per minute is often used for roughing, and 30 to 40 ft. per minute for finishing. It is customary to rough-grind at a higher surface speed of work than on finish-grinding.

In grinding cylinders the speed at which the article to be ground rotates is of great importance, as by running the grinding wheels at a constant speed and varying the speed of rotation of the article to be ground it is easy to secure uniform results from wheels having a variation in speed of 2,000 ft. per minute, and a variation from true grade within practical wheel-making limits. Thus, by varying the speed of the article to be ground, the necessity for variable revolutions for the wheel, or close limits of speed and grade for the wheel, are made unnecessary, and variable speed for the work accomplishes all that variable wheel spindles can accomplish, and much more. in a more simple way, at less expense.

The following grinding-wheel speeds are recommended for different types of grinding, in a recent number of *Grits and Grinds*, published by the Norton Co. of America :—

Type of Grinding.	Recommended peripheral speed (feet per minute).	Maximum peripheral speed (feet per minute).
Cylindrical grinding	5,500	6,500
Internal grinding	5,000	6,000
Snagging and general off-hand grinding on bench and floor stands	5,000	6,000
Surface grinding	4,000	5,000
Knife grinding	3,500	4,000
Hemming cylinders	2,100	2,400
Wet tool grinding	5,000	6,000
Cutlery grinding	4,000	5,000
Shellac and rubber cutting-off wheels	9,000	12,000

Every care must be taken to guard against the possibility of wheels being run at too high a speed. Thus, in the case of variable speed motor drive the speed control should be enclosed in a locked case, and cone pulleys should not be employed for speed regulation unless belt locking devices are provided.

New wheels should be run at working speed for 1 minute before applying the wheel to the work, the operator standing to one side. A heavy cut should not be taken with a cold wheel, but the wheel should be allowed to warm up slowly.

Storage of Wheels.

Extreme care should be exercised in the storage of wheels. They should be stored in a dry place and supported on edge in racks, except straight-sided shellac and rubber-bonded wheels of ¼ in. or less in thickness, which should be laid flat on a straight surface to prevent warping.

Wheels used for wet grinding should not be allowed to stand partly immersed in water, as this would tend to throw the wheel badly out of balance.

Mounting Abrasive Wheels.

If there is a large area of contact the method of mounting and safeguarding abrasive wheels and the condition of the grinding machine are of great importance.

British Standard Specification No. 620—1935 deals with the *dimensions* of grinding wheels and *methods of attachment*.

Sizes of Spindles.

The minimum sizes of spindles in inches (at that part where the wheel is located) for the various diameters and thicknesses of grinding wheels should not be less than the following table :—

SIZES OF SPINDLES.

Diam. of Wheel.	Thickness of Wheel in Inches.																	
	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
Ins.																			
6	½	½	½	½	½	½	⅝	⅝	¾	¾	¾	¾	¾	¾	¾	¾	1	1	1
7	½	½	½	½	⅝	⅝	⅝	¾	¾	¾	¾	¾	¾	1	1	1	1	1	1
8	⅝	⅝	⅝	⅝	⅝	⅝	¾	¾	¾	1	1	1	1	1	1	1	1¼	1¼	1¼
9	⅝	⅝	⅝	⅝	¾	¾	¾	¾	1	1	1	1	1	1¼	1¼	1¼	1¼	1¼	1¼
10	¾	¾	¾	¾	¾	¾	¾	¾	1	1	1	1¼	1¼	1¼	1¼	1¼	1¼	1⅜	1⅜
12	¾	¾	¾	¾	¾	1	1	1	1	1	1	1¼	1¼	1¼	1¼	1¼	1⅜	1⅜	1⅜
14	⅞	⅞	⅞	⅞	1	1	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1⅜	1⅜	1⅜	1⅜	1⅜	1⅜
16					1¼	1¼	1¼	1¼	1¼	1¼	1⅜	1⅜	1⅜	1⅜	1⅜	1½	1½	1½	1½
18					1¼	1¼	1¼	1⅜	1⅜	1⅜	1⅜	1⅜	1⅜	1⅜	1½	1½	1½	1⅞	1⅞
20						1⅜	1⅜	1⅜	1⅜	1⅜	1⅜	1⅜	1⅝	1⅝	1⅝	1⅞	1⅞	1⅞	1⅞
24						1⅜	1⅜	1⅜	1¾	1¾	1¾	1¾	1¾	1¾	2	2	2	2	2
26							1½	1⅜	1¾	1¾	1¾	1¾	2	2	2	2	2¼	2¼	2¼
30								1¾	1¾	2	2	2	2	2	2¼	2¼	2⅜	2⅜	2⅜
36									2	2¼	2¼	2¼	2¼	2¼	2¼	2⅜	2⅜	3	3

The spindle should be threaded left or right hand as required, so that the wheel clamping nuts will tend to tighten as the spindle revolves, and the threaded portion should be long enough to provide a bearing for the entire nuts when the wheel and flanges are in place.

The diameters of the spindle bearings for machines with overhanging wheels should not be less than the wheel-location diameters in the above table, and for single wheel machines (where the wheel is mounted between two bearings) should not be less than 80 per cent. of the diameters in the above table.

The wheels should be an easy fit on their spindles or locating spigots, and the soft metal bushing should not extend beyond the sides of the wheel.

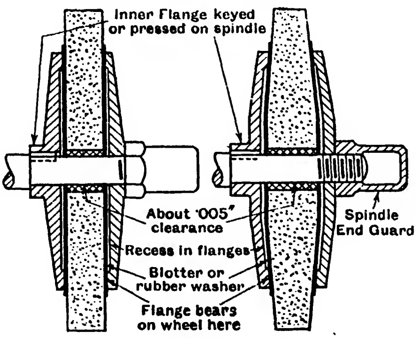


FIG. 1.

A grinding wheel should be fixed carefully and securely between recessed steel or iron plates or flanges about half the diameter of the wheel, with a washer of compressible material such as a sheet of thin rubber or one or two thicknesses of brown paper, on each side between the wheel and the flanges (fig 1). These washers should be an easy fit on the spindle and not larger in diameter than the flange. If flanges with Babbit or lead facings are used, the thickness of the Babbit or lead should not exceed ⅛ in.

Spindle end-nuts should be tightened only sufficiently to hold the wheel firm, otherwise undue strain may be set up in the wheel.

Care should be taken in screwing up very thin wheels that too much force is not applied. See British Standard Specification No. 1089.

FLANGES.

No abrasive whee used on the periphery should be mounted except with suitable flanges. When wheel guards are fixed, straight, dovetail, or taper flanges should be employed. Where guards cannot be used, and also on swing frame and portable grinders, taper flanges should be used with taper wheels.

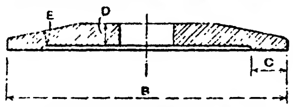
Flanges over 6 ins. diameter should be of steel of not less than 30 tons tensile. Smaller flanges may be made of cast iron. All flanges should be machined on the bearing surface and balanced. Every flange should be recessed at the centre at least $\frac{1}{16}$ in. on the side next to the wheel for a distance specified in the tables. The inner flanges should be keyed, shrunk, or pressed on to the spindle, and the bearing surface should run true with the spindle.

The minimum taper flange dimensions for taper wheels are given in the table, p. 1105. Such flanges should have a taper of $\frac{3}{4}$ in. per ft. No taper wheel over 24 ins. in diameter should be run with the wheel projecting from the flanges more than 4 ins. In the case of wheels 24 ins. diameter and under, the projection should not exceed 3 ins.

Dovetail flanges of the raised or countersunk type should be at least equal in diameter and thickness to the taper flanges. The taper provided for gripping the wheels should not be less than $\frac{3}{4}$ in. per ft. The recess should have ample clearance and be well radiused.

Cup, cylindrical, and sectional ring wheels should either be provided with guards or enclosed in protection chucks. Not more than a quarter of the height of the wheel should protrude beyond the guard or chuck. Where the thickness of the rim is less than 2 ins. the maximum distance that the wheel should project is 1 in. If the thickness of the rim is 2 ins. or more, a maximum projection of 2 ins. is allowable.

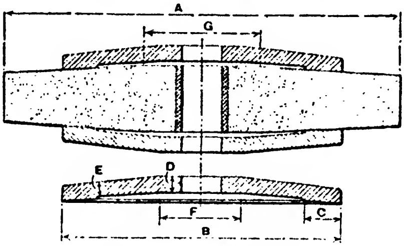
STRAIGHT FLANGE DIMENSIONS.



(All Dimensions in inches.)

A. Diameter of Wheels.	B. Minimum Outside Diam. of Plates.	C. Radial Width of Bearing Surface. Minimum.	Maximum.	D. Minimum Thickness of Plate at Bore.	E. Minimum Thickness at Edge of Recess.
1	½	$\frac{1}{16}$	⅛	$\frac{1}{16}$	$\frac{1}{16}$
2	1	⅛	¼	$\frac{3}{16}$	$\frac{3}{32}$
3	1½	$\frac{3}{16}$	⅜	¼	⅛
4	2	¼	½	⅜	$\frac{3}{16}$
6	3	¼	½	⅜	$\frac{3}{16}$
8	4	$\frac{5}{16}$	⅝	½	$\frac{5}{16}$
10	5	½	1	½	$\frac{5}{16}$
12	6	½	1	⅝	⅜
14	7	⅝	1¼	⅝	⅜
16	8	¾	1¼	⅝	$\frac{7}{16}$
18	9	¾	1¼	⅝	½
20	10	⅞	1½	¾	½
22	11	⅞	1¾	¾	¾
24	12	1	2	⅞	¾
26	13	1⅛	2¼	⅞	¾
28	14	1¼	2½	1	⅞
30	15	1⅜	3	1	⅞
36	18	2	3½	1⅛	1
40	20	3	4	1¼	1⅛

Taper Flange Dimensions.



(All Dimensions in Inches.)

A.	B.	C. Radial Width of Bearing Surface.		D. Min. Thickness of Plate at Bore.		E. Min. Thickness at Edge of Recess.		F.	G.
Diam. of Wheels.	Min. Outside Diam. of Plates.	Min.	Max.	For Double Taper Wheel.	For Single Taper Wheel.	For Double Taper Wheel.	For Single Taper Wheel.	Max. Flat Spot at Centre of Plate.	Approx. Diam. of Flat at Centre of Wheel.
8	4	5/16	3/4	3/8	3/8	1/4	1/4	0	1
12	6	1/2	1	3/8	5/8	5/16	5/16	4	4 1/2
16	10	3/4	1 1/2	1/2	7/8	3/8	7/16	4	6
20	14	1 1/4	2 1/2	1/2	1	1/2	9/16	4	6
24	18	1 1/2	3	5/8	1 1/8	9/16	5/8	4	6
30	24	2	4	3/4	1 1/4	5/8	3/4	4	6
36	30	3	4	1	1 3/8	7/8	1	4	6
42	36	3	6	1 1/4	1 1/2	1	1 1/4		
50	42	3	6	1 1/2	2	1 1/4	1 1/2		

Both flanges should be of equal diameter, to avoid cross-straining of the wheel. The inner flange should be keyed to the shaft and abutting against a shoulder on the same. Various firms have special mountings which have proved satisfactory.

Wheel Guards.

All abrasive wheels should have a guard or protection hood made of rolled steel. The hood should cover as much of the wheel as possible and also the spindle rod. The hood should be constructed to act as a dust hood which, when connected with an exhaust system, will eliminate the injurious effects of wheel dust.

To meet the requirements of factory inspectors, a number of wheel guards, good, bad, and indifferent, have been introduced by manufacturers. The features to be looked for in an appliance of this kind are:

(1) That it is of ample strength; (2) that it can be readily adjusted to follow the wear of the wheel. (3) The maximum angular exposure for guards used on bench and floor grinders should not exceed 90°, the exposure beginning at a point not more than 65° above the horizontal plane of the wheel-spindle. When the work demands contact with the wheel below the horizontal plane of the spindle the exposure should begin not more than 65° above, and extend not more than 60° below, the horizontal spindle plane. (4) On cylindrical grinders the maximum wheel exposure should not exceed 180°, beginning at a point not more than 65° above the horizontal spindle plane. (5) The maximum wheel exposure on surface grinders should not exceed 150°, and on swing frame and portable grinders 180°, the top half of the wheel being protected at all times. (6) Guards should be so constructed that the peripheral member may be adjusted to preserve the maximum wheel exposure. The distance between the wheel and the peripheral member should not exceed 1/4 in. The connection between the peripheral and side members or between the parts of a section guard should have a strength in a radial direction at least equal to the strength of the material of which the guards are constructed.

It is essential that the guard should be kept close to the periphery of the wheel in order to limit the flight of any portion which may break away.

Face or Cup Wheels **ought to be protected by being well and properly secured inside a Safety Adjustable Chuck, designed to allow only a small portion of the wheel to project whilst at work, and constructed so as to secure and support the wheel as close to the grinding face as possible.**

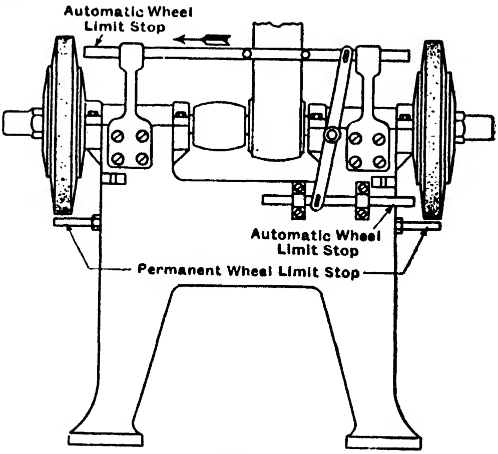


FIG. 2.

Machines fitted with two or more speeds should be provided with safety devices to prevent the maximum speed of the wheel being exceeded. In fig. 2 is shown a two-speed machine fitted with stops to limit the diameter and to prevent the belt being shifted to the high-speed pulley when the wheel exceeds a predetermined diameter.

Wheel Dressers:

Before using, the wheels should be trued up on their own spindles with a truing tool, and they should also be turned up when they get out of truth through wear.

Wheel dressers are used for 'truing' worn wheels; they should be equipped with rigid guards of sheet or cast metal over the tops of the cutters, to prevent flying pieces of broken cutters or wheel-particles from injuring the operator.

Dust Collection.

Dust generated by grinding should be suppressed or removed; preferably by localised exhaust ventilation. The hoods may be combined with the wheel guards or quite separate. They should be adjustable and should enclose the wheels as far as practicable. Hoods and ducts should be of substantial construction.

Except for very light dusts a pressure type of fan is a necessity, and special attention should be given to (*a*) the fan capacity; (*b*) the correct proportions and shapes of the ducts and branches; and (*c*) the means of collecting the dust after it has passed the fan. The efficiency of an exhaust installation is greatly increased by separating by gravity the heavy dust, burrs, fins, and miscellaneous material from the light dust, so that only the latter is directly dealt with by the fan. For this purpose some form of dust-separator (usually a cone) should be inserted in the duct between the hoods and the fan.

For testing Dust Collectors, see British Standard Specification No. 893—1939.

Rests.

Grinding wheels should revolve towards the rests, and the rests should be moved close to the wheels so as to prevent work being jammed between the rest and the wheel. The rest must never be moved while the wheel is in motion.

Lubrication.

The rate, effect, and efficiency of the grinding are also affected by the water or other lubricant applied to the wheel when in use. The function of the water is to carry away the heat resulting from the friction between the abrasive and the article being ground, and to wash away any loose particles dislodged by the grinding.

The water supply for all external grinding operations should be ample in volume without force. Using, say, a 14 in. diameter wheel, not less than 5 gallons and up to 10 gallons per minute should be used; on larger wheels 20 to 50 gallons per minute.

For many purposes, solutions of soap or emulsions of oil and water are preferable to plain water. A solution of carbonate of soda containing 2 oz. per gallon has the advantage of not producing rust, especially if ¼ or ½ pint of lard oil is mixed with it.

In grinding aluminium it is advisable to use a lubricant of lard, oil, kerosene, or a mixture of kerosene and gasolene or lard, or oil and gasolene.

Dry *versus* Wet Grinding.

In certain circumstances water may—or must—be dispensed with. Whenever it is necessary to see the edge of the wheel distinctly, to be sure of reaching some particular spot, water cannot be used, as it obstructs the view. Work revolving in the chuck or on the faceplate is also generally ground dry, as water is awkward to deal with on ordinary machines under such conditions. Cutters are also usually ground dry, although special care has to be taken to prevent 'drawing the temper' of the cutting edges. The amount to be removed is, in most cases, small, so that the sacrifice of time is balanced by the greater convenience of dry grinding.

More material can be ground away by dry internal grinding than by wet, and it is fairly common practice to run water on the work externally to dissipate as much heat as possible whilst grinding dry internally. On such classes of work as aeroplane cylinder grinding, or ring gauges, where a high degree of accuracy and finish is essential, water must be used, but it should be in such volume as to prevent as far as possible the chips being carried round to the wheel and interfering with the cut.

In *wet grinding*, care should be taken that the water supply is steady and adequate, and particular attention must be paid to this in the case of tool grinders, where it is of the utmost importance that the tool should be effectively covered with water, to obviate any loss of temper in the steel during the process of grinding. Some grinding machines provide for this; *e.g.* in one such patent machine the water is forced by a centrifugal pump through a series of small holes in the tool rest itself in such a manner as to completely envelop the tool being ground. A noticeable feature of this grinder is that there is total absence of splashing.

Grinding Procedure.*

Cylindrical Grinding—External.

The best average *surface speed* of modern grinding wheels, as made to-day from artificial abrasives, is about 6,000 ft. per minute for external cylindrical grinding, and the useful speed range is from 6,500 to 5,500 ft. per minute; below this speed excessive wheel wear is liable and very probable, and grinding machines should be arranged so that the effective life of the wheel falls within this range. It is of the utmost importance that the speed of the wheel be maintained during the cutting operation, no matter what the speed may be, and the drive should be sufficiently powerful to prevent slowing down during momentary heavy cutting. Not only is the wheel wasted by being allowed to slow down, but the wheel face is destroyed and more frequent truing up is necessary. It is more economical, from the point of view of wheel wear to metal removed, to keep the speed up so that softer and freer cutting wheels can be used, the speed enabling them to stand up to heavy cutting without undue wear, and, further, such wheels have very little tendency to glazing.

The *grade* of the wheel is determined by the material to be ground and the rigidity or otherwise of the work, and whether it is of large or small diameter. No definite rule can be laid down for the adoption of any particular grade, but it is unusual in general machine work to

* Abstract of Paper read by H. H. Asbridge before the Manchester Association of Engineers.

require more than two wheels to a machine, one to grind all classes or steel work and the other to deal with all cast metals. Closer grading can be done where the work is all of one class, as, for instance, a machine may be employed wholly on cast-iron pistons, crankshafts, or piston-rods, etc., in which cases particular grades of wheels can be employed.

The *diameter* of the wheels should be as large as possible.

The direct result of using large diameter wheels is economy of operation, the cost of wheel per cubic inch of material removed being less than in the case of the smaller wheel.

The *wheel width* has a direct result on the output, providing it is made use of, and to utilise a wide wheel to its full extent the table-travel must correspond.

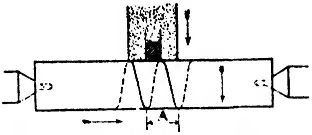


FIG. 3.—A = traverse per rev. of work = ⅔ width of wheel. Maximum wear of wheel tends to concentrate on shaded portion.

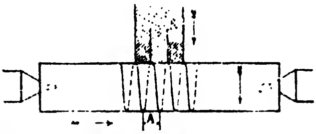


FIG. 4.—A = traverse per. rev. of work = ⅓ width of wheel. Maximum wear of wheel tends to concentrate on shaded portions.

Table Travel.—If the traverse of the work per revolution is less than half the width of the wheel, then the cutting face of the latter will gradually wear convex, but if the traverse per revolution of the work is over half the width of the wheel, then the wheel will preserve a flat face. (See figs. 3 and 4.)

The ideal traverse per revolution of the work is about two-thirds the width of the wheel, but it should not, except for finishing, be less than half the width.

The table should travel as rapidly as can be used. (See '*cross feed*,' page 1110.)

The main factor governing production on external cylindrical grinding machines is the combination of wide wheels with fast table speeds, as, other things being equal, the machine which possesses these advantages is the most efficient tool.

The *work speed* has only an indirect effect on the output. With a good wheel it makes little difference to the finish obtained whether the work surface speed be, say, 30 or 60 ft. per minute, except that, if the lower speed is persisted in, it limits the table travel, and so in turn limits the output. For external cylindrical grinding the practice is to work on an average surface speed of 60 ft. per minute as a basis. Low work speed is directly conducive to distortion and inaccuracy. The heat created by the cutting action of the wheel must be dissipated or distributed as quickly as possible, and can only be done by a higher work speed and table travel aided by an ample flow of water.

Vibrations in the work itself and disintegration of the grinding wheel are more liable to occur in large diameter and heavy work than in small work. It is much easier to control vibration in small diameter work, and it would be almost impossible to rotate it at such a high speed as to disintegrate the surface of the wheel, but the tendency would be to cause unnecessary wheel wear, and if the table speed was such that the traverse was less than half the width of the wheel the later would wear more at the corners than in the centre.

Vibrations on heavy work are much more difficult to control, and much depends on the machine and method of supporting the work. No definite rule can be laid down, as so much depends on circumstances. For instance, it would be both permissible and advisable to run an 8-in. tube at a higher surface speed than an 8-in. solid shaft, the limiting factor in the tube being a surface speed such as would cause disintegration of the wheel face, there being no heavy body of metal to set up vibration, whereas in the shaft the limiting factor would be a surface speed sufficient to set up an oscillatory motion difficult to control, and which would prevent the shaft from being ground to a perfect cylinder. This same defect could also be obtained by using a too slow surface speed, due to insufficient dissipation of the heat created by the grinding wheel. Between these two extremes there is a wide margin of surface speed at which grinding can be successfully carried on.

A good rule is to get a wheel suitably graded for the work, and when once a suitable grade is found it should be recorded, as wheels such as are made from artificial abrasives can be duplicated within very fine limits.

The *cross feed* to the grinding wheel should operate at each reversal of the table, that is, at the end of each stroke; this is necessary to distribute the work over the whole width of the grinding wheel. Production is not governed so much by the amount of cross feed as by other conditions. There should be as much cross feed as the work, the machine, or the grinding wheel will stand; this will be from ·0005 in. to ·0015 in. on the work diameter at each reversal of table. Under these conditions, a machine with a fast table travel will give maximum output with a comparatively light cross feed, as compared with a machine having only a slow table speed, and will have the advantage also of not stressing the work to the same extent. The wheel will remove more material for a given wheel wear, and the machine will be more economical in power.

Over-run of work should be avoided (see p. 1110).

The *time required* in external grinding cannot be calculated with any approach to accuracy on the usual basis of cutting speed and feed, etc.

The finish grinding of a shaft of a given diameter and length is not a constant operation with an abrupt termination, as in other methods of machining, but is an operation starting at zero, rising quickly to the point at which possibly the maximum cutting capacity of the wheel is demanded, and then falling gradually to zero or finished size, and the finer the limit fixed for the finished diameter, the more gradual will be the closing stages of the operation. Grinding times must, therefore, so far as the wheel is concerned, be based on a mean of its cutting capacity, much lower than its maximum.

The formula, as deduced from the above method, is as follows:

Diameter of work in inches × length in feet × constant = grinding time in minutes for the removal of $\frac{1}{32}$ in. in diameter and finishing to commercial limit.

TABLE OF CONSTANTS.

Diameter of Work.	Size of Wheel.	Constant.
4-in. shaft and upwards.	26 ins. by 3 ins.	1·3
3-in. ,,	,,	1·4
2-in. ,,	,,	1·8
1½-in. ,,	,,	2·2
1-in. ,,	,,	3
3-in. shaft and upwards.	14 ins. by 2 ins.	2·2
2-in. ,,	,,	3
1½-in. ,,	,,	3·7
1-in. ,,	,,	5
3-in. shaft and upwards	12 ins. by 1 in.	3
2-in. ,,	,,	3·8
1½-in. ,,	,,	4·5
1-in. ,,	,,	6·3

For the removal of $\frac{1}{64}$ in. in diameter allow $\frac{2}{3}$ of time calculated. For work below 1 in. diameter the grinding time tends to increase, depending entirely on rigidity of support afforded to same. Extra time should be allowed to special limits, such as may be required for drive fits, etc., or for special finished surfaces, such as are necessary for spindles, gauges, etc.

Internal Grinding.

The idea that internal grinding cannot be satisfactorily accomplished at spindle speeds less than 30,000 to 50,000 r.p.m. is erroneous. Provided that there is rigidity in the spindle employed, the grinding-wheel diameter has little effect on production. The effective speed range of a good grinding wheel is very much greater than in any other form of grinding, and ranges from 1,000 to 4,000 ft. per minute, and much of the most successful grinding is done at the surface speed of from 1,500 to 2,500 ft., but a very rigid spindle is necessary.

There are two types of internal spindles, the 'tube' type having a bearing immediately behind the wheel (this type is not to be recommended for holes, say, less than 6 ins. in length, as holes of this length, and longer, are generally insufficiently large in bore to admit of a substantial spindle), and the 'adaptor' type. The construction of the latter type is such that the spindle bearings never enter the hole being ground, the wheel being carried on an adaptor fitted into the main spindle and held by a draw bolt, or, if the work to be ground is of limited range, the adaptor portion is formed integral with the main spindle. For general commercial work the loose adaptor is preferable, as long or short, heavy or light adaptors can be used suitable for the particular work in hand. The fact that the main spindle in this instance never enters the hole permits of it being made of liberal dimensions. The Churchill Machine Tool Co. fix the higher limit of such spindles at 1,500 r.p.m., and so long as the rigidity is there behind the adaptor no higher speed is necessary for holes from $\frac{1}{4}$ in. upwards.

The *wheel width* has a direct result on the output.

Given a spindle with the necessary rigidity, the use of wide wheels and a high table speed are the governing factors in obtaining maximum production. The standard practice of the above firm is to use all wheels from $\frac{1}{2}$ in. to 3 ins. diameter $\frac{3}{4}$ in. wide, above 3 ins. diameter 1 in. wide. Such wheels are of no advantage unless combined with high traverse speeds, in order to utilise as much of the wheel width as possible, and also to dissipate the heat generated evenly.

Table traverse speeds up to 9 ft. and 10 ft. per min. are commonly applied to internal grinding machines.

The *work speed* in internal grinding can be varied considerably above or below the speed which has been previously determined to be the best for the particular work in hand. Such variation would have very little effect on production, but might affect the finish obtained. In standard practice a work surface speed of from 100 to 120 ft. per minute is used as a basis or starting point, and varied according to circumstances.

The *cross feed* requires special care in internal grinding.

Whatever type of internal grinding spindle is used (where the work to be ground is carried on a rotating spindle) there is bound to be overhang from its supporting bracket, with more or less unavoidable spring, and whilst the slide carrying the spindle can be set to a predetermined position with great accuracy, it does not follow that the hole being ground will have reached the desired size, owing to the spring in the spindle. This fact makes the use of automatic cross feed almost impossible of accomplishment if the amount of the feed has to equal the amount of cut which the wheel will take at each traverse of the table. Hand-operated cross feed fitted with an adjustment dead stop is therefore to be preferred.

In another type of internal grinding, the work is stationary and the grinding wheel has imparted to it a planetary motion whilst rotating at a high speed on its own axis, the planetary spindle being eccentrically adjustable whilst grinding is proceeding. This type of machine has been employed for grinding motor-car cylinders when cast *en bloc*, and the possibilities of this type of machine for finishing holes are great.

Over-run of work should be avoided; that is, the wheel should never be allowed to leave the work, the traverse being only just sufficient to allow of not more than one-third of the width of the wheel projecting beyond the end of the work. If the wheel is allowed to run off, the result in external grinding will be to have the face of the wheel sheared away by the sharp ends of the work; and in internal grinding the hole will be bell-mouthed.

Surface Grinding.

There are two methods of surface grinding—one using the side of the grinding wheel,* and the other the edge of the wheel, carried on either a vertical or horizontal spindle, the more widely used being the vertical spindle type. The two methods are further subdivided according to the manner in which the work is moved, machines being made with the work carried on a reciprocating table, or on a circular revolving table, briefly classified as follows:—

1. Horizontal spindle surface grinding machine, where the work is reciprocated under the grinding wheel.

2. Horizontal spindle ring and surface grinding machine. In this machine the work is carried on a circular revolving table, generally made in the form of a magnetic chuck, and rotated under the grinding wheel.

3. Vertical spindle ring and surface grinding machine, where the grinding wheel is of cup form, carried on a vertical spindle, and the work is carried on a circular rotating table, generally in the form of a magnetic chuck and rotated under the wheel.

4. Vertical spindle surface grinding machine, where the wheel is of cup form carried on a vertical spindle, and the work is carried on a table reciprocating under the wheel.

The first and fourth types are the most commonly used, and intended for the grinding of flat surfaces of all kinds and in all materials.

* Grinding on the flat sides of straight wheels is often hazardous, and should not be allowed. Cup or ring wheels should be used.

The second and third types are widely used for grinding the sides of disc work, such as circular knives and saws, piston rings, disc valves, etc., using the rotary table. The second type of machine with the horizontal spindle leaves the work with a concentric finish, whereas the third type of machine with the cup wheel leaves the work with a radial finish, and which in some classes of work is objectionable. particularly, for instance, in piston rings, where the concentric finish is preferred. So far as production is concerned the vertical spindle ring and surface grinding machine will give a greater output then the horizontal spindle ring and surface grinding machine. Both machines can be arranged to grind flat, concave, or convex surfaces.

The most largely used machine for regular commercial purposes is the machine with the vertical spindle, owing to its greater productive capacity.

The best *wheel speed* for all-round surface grinding is about 4,000 ft. per minute, with very little margin of speed—that is, if a wheel is working successfully at 4,000 ft. per minute, it cannot be varied up or down from this speed without affecting production, consequently the vertical spindle machine with the cup wheel has a great advantage in this respect, due, of course, to the wheel retaining its diameter as it wears.

The grade of the wheel for surface grinding demands far closer attention than in any other type of grinding. The only remedy for a wheel that is too hard or too soft is to change it for a wheel of the correct grade.

A high *table speed* is the main factor governing output, using suitable grinding wheels, but the machine which will so work that the temperature is even throughout the whole ground surface will produce the most accurate results.

The vertical surface grinding machine with the cup wheel, which covers the full width of work at each stroke of the table, has a distinct advantage over the horizontal spindle machine using the periphery of the wheel, and which would have to be traversed transversely in order to cover a work surface wider than the width of wheel.

Quite apart, however, from the question of heat distribution, the vertical spindle machine has the greater productive capacity, and for grinding wide surfaces or groups of small articles this type of machine has the advantage of the horizontal spindle machine. The vertical surface grinding machine is now being adopted in all classes of engineering works, and wherever flat surfaces have to be produced, such as in machine-tool work, there is no method that can be compared with it for cheapness of production.

A machine of this type added to the equipment of any works, and used for the production of flat surfaces, has, more than any other type of grinding machine, a beneficial effect throughout the works, and in this respect raises automatically the standard of work produced. In this way, given an accurate machine, it is so very easy to true up straight edges and similar tools and keep them all in a constant high state of perfection ; and by having a machine of this type always in readiness to produce flat surfaces the result is that more accurate work is automatically produced throughout the works.

The *time required* for grinding a given surface can be ascertained with fair accuracy once a basis is found, based on area ground and amount of metal removed. Under normal working conditions, a grinding wheel has a certain capacity for the removal of metal, based on its cutting area or its diameter and width.

Grinding Allowances.

(For Norton and Heald Grinding Machines.)

Frequent inquiries as to the amount of stock which should be left on turned and bored work to be ultimately finished to size by grinding had led to the answer that this depends on several factors which must be ascertained before a correct allowance can be determined. The object aimed at should be to so proportion the work between the lathe and the grinding machine that the total time taken by both in producing the desired degree of finish and accuracy is reduced to the minimum.

In a desire to turn out a 'good job' the turner sometimes overlooks the fact that a smooth piece of work having only a small allowance may take as long to finish in the grinding machine as a rough-turned job on which more metal is left.

The chart, fig. 5, shows the grinding allowances which are recommended for turned or bored work up to 12 ins. diameter. Although the amounts specified may, to some, appear excessive, they may be allowed with confidence in establishments possessing a grinding department run according to modern ideas of efficiency and lathes sufficiently powerful to take the coarse feeds recommended.

Curves A and A' show maximum and minimum amounts to be left on the diameter of rough turned work over 12 diameters long.

Curves B and B' show maximum and minimum amounts to be left on the diameter of rough turned work under 12 diameters long.

Curves C and C' show maximum and minimum amounts to be left on diameter of smooth turned and bar-lathe work over 12 diameters long.

Curves D and D' show maximum and minimum amounts to be left on diameters of smooth turned and bar-lathe work under 12 diameters long.

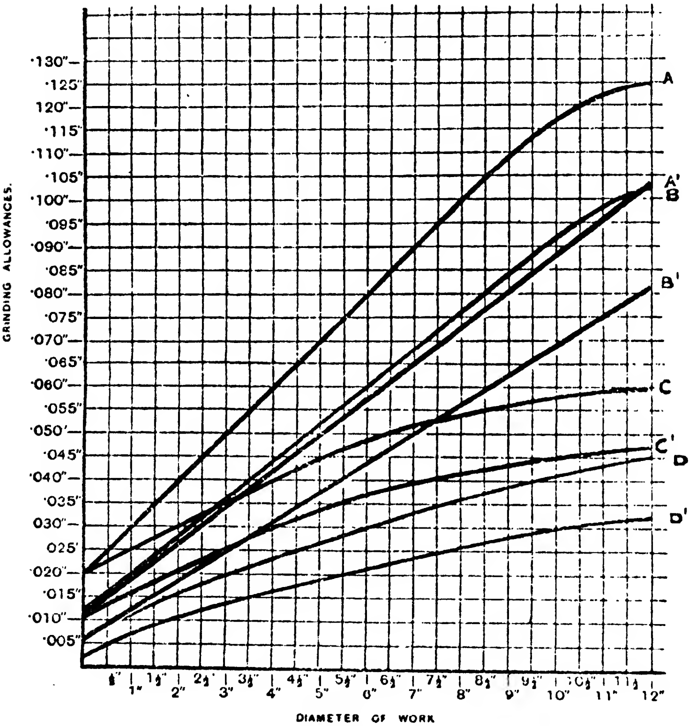


FIG. 5.—Grinding Allowances for Norton Cylindrical Grinding Machines.

NOTE.—By 'rough turned,' is meant work having coarse traverse marks, the measurements being taken on the top of the ridges. The following are the finest feeds which should be employed :—

For work up to 2 ins. diameter	30 per inch.
For work from 2 ins. to 4 ins. diameter	20 „
For work from 4 ins. to 8 ins. diameter	16 „
For work from 8 ins. to 12 ins. diameter	8 „

Case-hardened work is an exception, because the finish must be fairly smooth to allow regular penetration of the casing compound. Use curves O and O' for all such work.

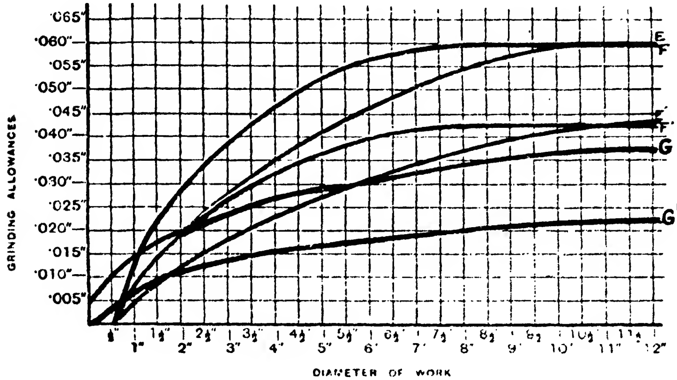


FIG. 6.—Grinding Allowances for Heald Internal Grinding Machines.

Curves E and E' (fig. 6) show maximum and minimum amounts to be left in the bore of mild steel work that is to be case-hardened before grinding.

Curves F and F' show maximum and minimum amounts to be left in the bore of carbon or high-speed steel work that is to be hardened before grinding.

Curves G and G' show maximum and minimum amounts to be left in the bore of soft work, whether steel, cast iron, or brass.

NOTES.

The following explanatory notes will serve to show the theory followed out, and will assist in determining allowances for the exceptional jobs which will occasionally present themselves :—

Taking first *external work* (fig. 5), as being the more common, it will be noted that a larger allowance is made on articles which exceed twelve diameters length than in the case of shorter jobs. This is to save lathe work.

A job over twelve diameters long will spring considerably under a good cut, and there is a possibility of bends making their appearance. The largest possible allowance is therefore made on this work, so that the turner need not waste the time which would be taken in producing articles more nearly parallel and straight.

Then, again, a distinction is made between rough-turned work and that which is produced by a fine feed or made on turret lathes with a box tool.

For all work turned between centres, with the exception of that which is to be carbonised for case-hardening, the use of a coarse traverse is advocated, the finest feed per inch that should be used for various diameters being specified on the chart. Work which is to be carbonised must be turned with a fairly smooth finish, so as to allow regular penetration of the casing compound ; therefore the allowance specified for smooth work over twelve diameters long should be used for case-hardened work whatever its length. If such work is hollow it may distort after quenching, therefore great care must be exercised in hardening, as the allowance above specified cannot be exceeded or the hard skin would be entirely removed in grinding to size.

The allowance for carbon and high-speed steel work which is hardened will be the same as that provided for soft work, as any distortion which occurs must be corrected in the straightening press before grinding.

Internal work (fig. 6) is divided into three classes as follows :—

1. Mild steel case-hardened.
2. Carbon or high-speed steel hardened.
3. Soft work, whether steel, cast iron, or bronze.

Hardened and case-hardened work in which the hole is less than half an inch in diameter should as shown in the chart, be bored or reamed to standard size, as there will be a sufficient shrinking when the quenching takes place to provide a grinding allowance. It will be noted that the amount allowed in case-hardened work increases very rapidly up to 6 ins. diameter; this is to allow for distortion in hardening. Above this size the increase is slow, and care should be taken to avoid excessive distortion, as no larger allowance can be made, otherwise the hard skin would be removed by the grinding operation. In determining the allowance for soft work it should be remembered that the class of finish given to the hole will decide whether the high or low limit should be worked to. Thus, in holes which are rough drilled or bored, the allowance should approach the maximum, whilst in reamed holes the minimum allowance should be left.

Abrasive Cloths and Papers.

It is usually advisable when ordering Abrasive Cloths and Papers to stipulate that they must comply with the corresponding British Standard Specifications.* These specify the abrasive, backings, adhesive, workmanship, form (sheets or rolls), sizes, grades, tensile strength and the Marks whereby material corresponding to the B.S. Specification may be identified. Each sheet or roll must also show the (*a*) manufacturer's name and trade mark so that the manufacturer may be readily determined and (*b*) the Grade Number.

Material.	Uses.	Abrasive.	B.S. Specification No.	Identification Mk.
Alumina Cloth	Metal work	Aluminium oxide	872—1939	872F
Emery Cloth	Engineering and general	Emery	871—1939	871A
Emery Paper	—	—	—	—
Flint Cloth	Wood and general	Flint, quartz or quartzite	871—1939	871E
Flint Paper	"	"	871—1939	871D
Flint Paper (Waterproof)	Paint, rub down, and general	"	872—1939	872B
Flint Rolls	Wood and general	"	872—1939	872A
Glass Cloth	"	Glass	871—1939	871C
Glass Paper	"	"	871—1939	871B
Garnet Cloth	"	Garnet	872—1939	872E
Garnet Paper	"	"	872—1939	872D
Silicon Carbide Paper (Waterproof)	Wet rubbing down of filler, paint and lacquer surfaces and general work	Fused Silicon Carbide†	872—1939	872C

Polishing.

The term 'polishing' is applied to two distinct processes: (i) the removal of coarse irregularities on a surface, and (ii) the production of a smooth, reflective surface. For the first, coarse abrasives or even abrasive wheels or blocks are employed, but for a glossy surface milder abrasives and sometimes solutions of waxes are used.

For polishing slabs of *marble*, *stone*, *glass*, and similar materials steel discs, mechanically operated, scour the surface with the aid of wet sand, emery carborundum or other abrasive powder, which is gradually replaced by finer abrasive particles, the final polish being imparted by whiting and sometimes by a mixture of whiting and oil.

For *smoothing wood* and similar materials, garnet, emery, sand or glass, mounted on paper or cloth, are largely employed, garnet being especially suitable for hard woods. For finishing wooden handles, sand mounted on endless belts is largely used (see 'Re-polishing').

For work in which an *exceptionally hard abrasive* is required, including the truing of abrasive wheels, diamond bort or carbonado are employed.

* No. 871—1939, for papers and cloths for general purposes, and No. 872—1939, for papers and cloths for technical and other special purposes. Published by the British Standards Institution, 28 Victoria Street, London, S.W. 1.

† Silicon Carbide is the technical name of the abrasive; the trade-mark names for two well-known brands are 'Carborundum' and 'Silundum.'

For polishing *steel articles*, blocks of carborundum or aluminous abrasive or 'mops' made of cloth or paper covered with emery or corundum are largely employed.

For polishing *other metals, glass, etc.*, loose powders are employed. Such powders may be used in the dry or wet state, and may be rubbed on to the material to be polished, or they may be applied by means of a wheel, 'mop' or 'dolly' composed of discs of cloth.

For polishing *brass and other softer metals*, very fine silica, quartz flour, emery, silicon, carbide or alumina, crocus, rouge, or whiting are employed. For *silver* and *gold*, well-slaked lime is largely used. The polishing powder is sometimes treated with rouge to disguise its nature and to give white metals a slightly ruddy tint, which improves their appearance.

Re-polishing is the process employed to restore a brilliant surface to one which has previously been polished but has become dull. For re-polishing a mixture of (*a*) a detergent such as oxalic acid or soap, (*b*) a mild abrasive such as very fine (air-floated) silica or whiting, and (*c*) a vehicle is used. The vehicle may be water, but paraffin or other organic liquid is generally used.

For re-polishing most *metals* it is sufficient to apply a little of the polish and to rub vigorously with a soft cloth, but if the surface is very seriously tarnished it may be necessary to give a preliminary treatment with a slightly coarser abrasive (emery) by means of a 'mop' or 'dolly.'

When re-polishing wood or similar materials, the surface is first cleaned in any suitable manner and the pores are filled with wax or other suitable 'stopping.' The surface is then coated with a solution of shellac or other 'gum' in alcohol or other solvent. At the present time several 'gums' and solvents are available.

For *leather*, the most suitable polishes consist of a solution of one or more waxes (with or without a detergent) in a convenient solvent, of which many, sold under trade names, are available.

Lacquers are now increasingly used instead of polishes, those consisting of celluloid in a suitable solvent being quite satisfactory. These lacquers are usually sold under trade names.

Power required for Emery Polishing.

A calico mop, 12 ins. diameter by 2 ins. wide, run at a speed of 10,000 feet per minute, requires about 3½ b.h.p. to apply it with the requisite force. A 10-in. mop requires about 3 b.h.p.

Sand Blasting.

The process of sand blasting consists in projecting a stream of sand, suspended in air, against a surface either to smooth it or to impart a slight roughness such as that on 'ground glass.'

The air pressures used for different materials are, approximately, as follows:—Steel castings or forgings, 80 to 100 lbs.; malleable iron, 70 to 85 lbs.; cast iron, 60 to 70 lbs.; and brass and aluminium, 35 to 50 lbs.

For some classes of work a small nozzle opening, that is, a fine strong jet, may be desirable; for other work a broader stream, covering a larger surface but working at a lower pressure may be best. Table I. gives the relation between air flow in cubic feet of free air per minute at varying pressures, and the horse-power required to produce them by single-stage compression, for nozzles ranging in diameter from ⅛ to ⅝ in. inclusive.

In cases where there is much moisture in the air or sand it is well to make provision for its elimination by installing a moisture separator or a suitable heater, as moist sand 'balls' badly and chokes the jet.

Sand is the most commonly used abrasive on account of its relatively low price. Ordinary sand is inferior to sharp sand and crushed silica, as the two latter possess greater cutting power and are therefore more effective. Abrasives such as steel grit and shot are now largely used. For classes of work such as electro-plating or galvanising, the metallic dust adhering to the work prohibits the use of carborundum, because it prevents perfect galvanising, although no difficulty is experienced in this respect with sand. For some purposes, steel grit appears to have special advantages over sand and other abrasives.

All abrasives should be screened each time before using, to remove particles large enough to clog the nozzle, and also to eliminate fine particles which only produce dust and have no abrasive quality, but which consume some of the pressure. Screen separators, frequently operated by compressed air, may be used for this purpose.

Table I.—Flow of Free Air for Different Sizes of Nozzles.

Diameter of Nozzle. Ins.	Pressure (cub. ft. per min.) and Corresponding Horse-power Required.							
	20 lbs.	H.P.	30 lbs.	H.P.	40 lbs.	H.P.	50 lbs.	H.P.
$\frac{1}{8}$	7·70	0·63	10·00	1·03	12·30	1·50	14·50	1·99
$\frac{3}{16}$	17·10	1·40	22·50	2·32	27·50	3·26	32·80	4·49
$\frac{1}{4}$	30·80	2·53	40·00	4·12	49·10	5·99	58·20	7·97
$\frac{5}{16}$	48·17	3·95	62·89	6·48	76·60	9·36	90·70	12·43
$\frac{3}{8}$	69·00	5·66	90·00	9·27	110·00	13·42	130·00	17·81
Diameter of Nozzle. Ins.	60 lbs.	H.P.	70 lbs.	HP.	80 lbs.	H.P.	100 lbs.	H.P.
$\frac{1}{8}$	16·80	2·57	19·00	3·19	21·20	3·86	25·73	5·33
$\frac{3}{16}$	37·50	5·74	43·00	7·22	47·50	8·65	57·88	11·98
$\frac{1}{4}$	67·00	10·25	76·00	12·77	85·00	15·47	103·00	21·32
$\frac{5}{16}$	105·00	16·07	119·00	20·00	133·00	24·10	161·00	33·82
$\frac{3}{8}$	151·00	23·10	171·00	28·73	191·00	34·76	232·00	47·90

SECTION XXII

PART III

STEAM-HAMMERS.

Energy and Velocity of Blow.

If

a = area of piston, in square inches : p = average pressure of steam on piston during downward stroke, in pounds per square inch : S = stroke of piston, in feet : W = falling weight in pounds : E = energy of blow after full stroke and before striking, in foot-pounds, friction being neglected ;

$$E = (ap + W)S.$$

If

P = total pressure on piston = pa : F = total force causing downward acceleration = P + W = pa + W : g = acceleration due to gravity = 32·2 : V = velocity after full stroke and before striking, in feet per second, friction again being neglected ;

$$V^2 = \frac{2FgS}{W}.$$

No commercial use is made of these calculations, either for the naming of sizes of hammers or for any other purpose.

The question is sometimes asked : *What weight of blow does the hammer strike ?*

The force of a blow cannot be stated in terms of weight at all, because the pressure of a weight is continuous, whereas the force of a blow is expended in a moment. It has, however, been ascertained, by careful experiments, that the maximum blow of a 5-cwt. double-acting steam-hammer, with moderate steam pressure, produces a crushing effect upon a piece of hot iron as great as that produced by a load of about 30 tons, and a ½-cwt. double-acting steam-hammer a crushing effect equal to that produced by a load of about 2½ tons.

Naming Sizes of Steam-Hammers.

In naming the sizes of steam-hammers, no account is taken of the pressure of steam upon the piston. Thus, a 5-cwt. hammer is one in which the falling mass weighs at least 5 cwt., but of course the force of the blow is greatly increased by the pressure of steam above the piston accelerating the fall.

DIMENSIONS, WEIGHTS, AND MEMORANDA OF OVERHANGING FORM STEAM-HAMMERS WITH SLIDES (*B. & S. Massey.*)

Standard sizes	½ cwt.	1 cwt.	1½ cwt.		3 cwt.
Intermediate size				2 cwt.	
Diameter of cylinder	4½″	5½″	6″	6″	7½″
Longest stroke	11″	12″	13″	13″	17″
Clear space between slides	5½″	6″	6″	6″	8¾″
Overhang to pallett centres	7½″	9″	10″	10″	11″
Total height from floor	5′ 6″	5′ 9″	6′ 9″	6′ 10″	8′ 1″
Steam and exhaust pipes	1″ & 1¼″	1″ & 1¼″	1¼″ & 2″	1¼″ & 2″	2″ & 2½″
Approximate weight	1 ton	1½ tons	2 tons	2¼ tons	3½ tons
Without anvil block	¾ ton	1 ton	1½ tons	1½ tons	2½ tons
Without anvil block and baseplate					1⅝ tons
*Diameter of bar worked efficiently	1½″	2″	2½″	3″	4″
†Approximate steam consumption (lbs. per hour)—					
(*a*) On hard blows	115 lbs.	215 lbs.	315 lbs.	410 lbs.	420 lbs.
(*b*) Average on mixed blows	19 lbs.	35 lbs.	50 lbs.	64 lbs.	73 lbs.

Standard sizes	4 cwt.	5 cwt.		7 cwt.	
Intermediate sizes			6 cwt.		8 cwt.
Diameter of cylinder	8″	9″	9″	10″	10″
Longest stroke	19″	21″	21″	24″	24″
Clear space between slides	9½″	10½″	10½″	11¼″	11¼″
Overhang to pallett centres	11″	13″	13″	14″	14″
Total height from floor	8′ 9″	9′ 4″	9′ 4″	10′ 1″	10′ 1″
Steam and exhaust pipes	2″ & 2½″	2½″ & 3″	2½″ & 3″	2½″ & 3″	2½″ & 3″
Approximate weight	3¾ tons	5½ tons	6 tons	7½ tons	7¾ tons
Without anvil block	2¼ tons	3½ tons	3½ tons	4½ tons	4½ tons
Without anvil block and baseplate	1¾ tons	2½ tons	2½ tons	3½ tons	3½ tons
*Diameter of bar worked efficiently	4¾″	6″	6½″	7″	7½″
†Approximate steam consumption (lbs. per hour)—					
(*a*) On hard blows	520 lbs.	620 lbs.	710 lbs.	800 lbs.	880 lbs.
(*b*) Average on mixed blows	89 lbs.	105 lbs.	119 lbs.	134 lbs.	147 lbs.

Standard sizes	10 cwt.		15 cwt.	20 cwt.	25 cwt.
Intermediate size		12 cwt.			
Diameter of cylinder	12	12″	13⅜″	14½″	15¾″
Longest stroke	27″	27″	30″	33″	36″
Clear space between slides	13″	13″	14½″	16″	17′
Overhang to pallett centres	16″	16″	18″	20″	22″
Total height from floor	11′ 1″	11′ 1″	12′ 3″	13′ 1″	13′ 9″
Steam and exhaust pipes	3″ & 3½″	3″ & 3½″	3″ & 4″	4″ & 4½″	4″ & 4½″
Approximate weight	10 tons	10¾ tons	14¼ tons	18½ tons	23 tons
Without anvil block	6 tons	6 tons	8½ tons	10¾ tons	13½ tons
Without anvil block and baseplate	4½ tons	4½ tons	6 tons	7¾ tons	9½ tons
*Diameter of bar worked efficiently	8″	8½″	10″	12″	13½″
†Approximate steam consumption (lbs. per hour)—					
(*a*) On hard blows	1040 lbs.	1190 lbs.	1420 lbs.	1750 lbs.	2100 lbs.
(*b*) Average on mixed blows	170 lbs.	195 lbs.	228 lbs.	275 lbs.	330 lbs.

*** All these figures are necessarily rough approximations.**

They are meant to apply to average engineers' smithwork, and are for efficient results on this class of work, with a steam pressure of about 60 lb. per sq. in.

Even for this they may for occasional jobs be considerably exceeded.

There are many classes of work which require much larger hammers, and some classes which require smaller hammers than those indicated, owing chiefly to the varying amount which has to be done at a heat.

† Steam Consumption. **A steam pressure of about 60 lb. per sq. in. is assumed.**

During ordinary forging there are many blows of less than full force, and there are also necessary pauses (holding up) for adjusting work or tools during a heat.

While line (*a*) gives the consumption on continuous hard blows only, line (*b*) gives the consumption as ordinarily reduced by these circumstances. Further, under average smithy conditions, a hammer may be assumed to be actually in motion about 15 minutes in every hour, and the figures given in line (*b*) are based on this assumption.

Wt. of falling parts.	5 cwts.	7 cwts.	10 cwts.	15 cwts.	20 cwts.	25 cwts.	30 cwts.	40 cwts.	50 cwts.	3 tons.	4 tons.	5 tons.	6 tons.	7 tons.	8 tons.	10 tons.
Dimensions, etc.:																
Cylinder diam. ¶ ins.	9	10	12	13¾	14½	15¾	17	18	19½	21	24½	27½	29	31	31	34
Stroke (maximum) ins.	21	24	27	30	33	36	39	45	49	54	60	66	66	74	74	84
Tup pallett face ins.	10×7	11×8	12×9	13½×9½	15×11	16×11	17½×12	19½×14	21½×14	21½×16	27×16	28×16	28×18	30×18	32×18	35×20
Space between standards . . ft. ins.	3 10½	4 6	5 2	5 9	6 3½	6 10	7 9½	8 6	9 1	11 1	12 2	12 8	12 9	13 10	14 0	15 9½
Space between slides ins.	10½	11½	13	16	17	17	20½	22½	24	25½	28	29	30	33½	35	38
Anvil pallett face to slides . ins.	10½	11¾	14	17	18	18½	21	25	27	33	36	36	36	40	40	42
Total height from floor (approx.) . ft. ins.	8 9	9 9	10 10	12 0	12 10	13 6	14 5	15 10	16 8	18 3	20 7	22 8	22 8	23 10	23 10	26 6
Baseplate (floor space) (approx.) . ft. ins.	8 0 × 4 6	9 0 × 4 10	10 0 × 5 6	11 0 × 5 10	12 0 × 6 4	13 4 × 6 9	14 0 × 7 4	15 3 × 7 6	15 10 × 10 2	18 3 × 11 2	20 2 × 11 6	21 1 × 12 0	21 2 × 12 0	23 4 × 12 6	23 6 × 12 6	26 0 × 13 6
Supply and exhaust pipes . . ins.	2½ & 3	2½ & 3	3 & 3½	3 & 4	4 & 4½	4 & 4½	4 & 5	4½ & 5	4½ & 6	5 & 6	6 & 7	6 & 7	7 & 8	7 & 8	7 & 8	8 & 9
Weights (approx. gross)																
Complete . tons	5½	7½	9¾	14½	18¾	23	28	37½	43	51½	67½	81	90½	114½	124½	163
Without anvil block tons	3½	4¾	6	8¾	11	13½	16½	21¾	23½	28½	36½	42½	43¾	60	62	84¾
Without anvil block and baseplate tons	2½	3½	4	5¾	7½	9	10¾	14½	16¾	20½	26¾	31¾	33½	45¾	47¾	65½
Steam consumptions ¶:																
Maximum lbs. steam per hour	800	1050	1390	1420	1750	2100	2400	3000	3520	4050	5050	5950	6850	7700	8500	10100
Average * lbs. steam per hour	123	158	203	228	275	330	375	450	525	600	750	875	1000	1100	1200	1420
Boiler capacity †‡ lbs. steam per hour	600	770	1000	1090	1340	1580	1800	2220	2620	3000	3680	4340	4950	5550	6100	7200
Air consumptions ¶:																
Maximum cu. ft. free air per min.	290	375	500	505	630	750	860	1070	1260	1450	1800	2130	2450	2760	3050	3600
Average * cu. ft. free air per min.	44	56	72	81	100	118	134	169	191	220	270	315	360	400	440	510
Compressor ‡§ cu. ft. free air per min.	240	310	400	440	540	640	750	900	1100	1230	1550	1800	2100	2350	2600	3100

* Assuming the hammer to be at work for 15 mins. in each hour, on a mixture of light and heavy blows, with necessary pauses for holding up to adjust work and tools.

§ For single hammer. For two or more the sum of the figures for the individual hammers may be reduced by the following amounts: 2 hammers—20%. 3 hammers—30%. 4 hammers—35%. 5 or 6 hammers—40%. 7 to 9 hammers—45%. 10 or more—50%.

† Normal continuous evaporative capacity. ‡ In conjunction with receiver having volume equal to the one-minute supply from the compressor.

¶ Assuming a pressure of 60 lbs. per sq. in. = 4·2 kgs. per cm.^{2}.

In cases in which one boiler is put down solely to drive one hammer, the capacity of such boiler should be fixed according to line (a), page 1118, because otherwise the pressure would fall too low during a normal run of hard blows.

The standards are normally made of cast iron, but for specially severe work cast steel standards are recommended. As a further alternative the standards may be built up of steel plates and angles firmly riveted together.

Foundations.

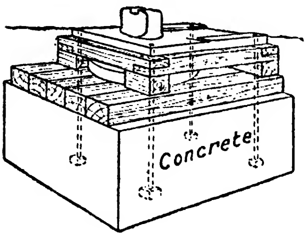


FIG. 1.—Typical Foundation for Steam-Hammers of the Overhanging Form.

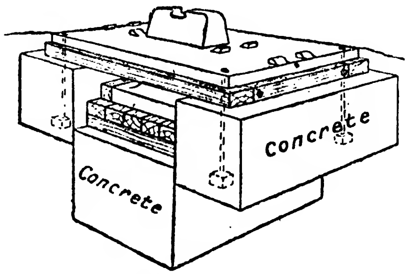


FIG. 2.—Typical Foundation for Steam-Hammers of the Arch Form.

(B. & S. Massey, Ltd.)

PNEUMATIC POWER HAMMERS.

These hammers are arranged for belt or for direct electric drive. They are similar to steam-hammers in many respects, and are about equal to them in working capacity, assuming the steam-hammers to be working under ordinary conditions of pressure, etc.

DIMENSIONS, WEIGHTS, AND MEMORANDA OF PNEUMATIC POWER-HAMMERS.

OVERHANGING FORM WITH SLIDES.

Standard sizes	½ cwt.	1 cwt.	2 cwt.	3 cwt.	5 cwt.	7 cwt.	10 cwt.	15 cwt.	20 cwt.
Longest stroke (inches)	11	12	14	17	21	24	27	30	33
Driving pulley diam. and width of belt (inches)	21×2½	24×3	30×3	33×4	39×5	45×7	48×10	27×6	30×9
Blows per min.	280	240	200	160	140	125	110	100	85
Size of bar worked efficiently (inches square)	1½	2	3	4	6	7	8	10	12
Maximum power (full blow) B.H.P.	4	7	10	16	21	25	35	48	60
Mean consumption in B.T.U. (kwt. hours), approximate	·6	·9	1·5	1·9	2·7	3·3	4·2	5·5	6·6
Size of motor (continuous rating)	3	5	7½	12	15	20	27	35	40
Approximate weight of hammer (tons)	1¼	1¾	3	4½	6¼	9½	12½	18¾	23½
Ditto, without anvil block (tons)	—	—	2¼	3	4¼	6¾	8½	13	15½

(B. & S. Massey, Ltd.)

DROP STAMPS.

The practice of Stamping, in Dies, parts which have to be made in large numbers, such as motor-car parts, railway wagon parts, etc., etc., has become very general, and the tool usually adopted for this purpose in this country is the 'drop stamp.' Latterly, the friction lifted type of drop stamp (of the strap or of the board type) has tended to supersede the steam lifted type, on account

of the improved control, the greater efficiency and the smaller cost of up-keep. A usual plan is to have two, three, or four drop stamps of different falling weights arranged in a battery.

The strap lifters themselves resemble a band brake in construction. A steel brake band lined with suitable material, and connected with the lifting arm, is made to grip a constantly revolving drum by means of a lever actuated by hand through a pulling cord, and the tup is thereby raised. As soon as the tension on the cord is released, a spring disengages the band, and the tup falls under the influence of gravity. In the board type the strap is replaced by a long plank which is raised by being pinched between steel rollers.

Progress in drop stamping in recent years has been very considerable, owing to the fact that the manufacturer of equipment has had to keep pace with an ever-increasing demand from the user of drop hammers, for increased production, in order to meet particularly keen competition. One of the outstanding developments is the application of dies containing multiple impressions, instead of the usual single impression. By this means drop forgings can be produced direct from the bar in the same hammer, from start to finish. The saving of time is very considerable, and there is no necessity to employ a separate hammer for dummying. Multiple dies can, of course, be used with all types of hammers, assuming they are of the latest design, providing the necessary strength to resist the eccentric shock created by the impact of the tup coming on to one side, instead of centrally.

The Table below—kindly prepared by Brett's Patent Lifter Co., Ltd., of Coventry—gives the production obtained for plain and also intricate forgings, by means of this type of equipment, together with the size of stock required on the basis of the projected area. Whilst it is advisable to take each stamping on its merits, it may be said that, in a general way, this information is authentic, and based on actual results.

DROP FORGINGS PER HOUR.

Weight to hammer. (lbs.)	Size of steel.	Projected area of forging.	Thin flat forgings, flat stock.	Ordinary forgings, square or round stock.	Intricate forgings.
600		2½ sq. in.	380	300	170
		3½ " "	300	260	150
800	⅝ in. round to ⅞ in. square	2½ " "	380	345	215
		3½ " "	300	260	170
1000	⅞ in. round to 1¼ in. square	2½ " "		215	195
		3½ " "		195	170
		4½ " "	300	215	150
		6 " "	215	150	130
1200	1 in. square to 1½ in. square	4½ " "	260	215	150
		5 " "	260	215	170
		6½ " "	235	215	170
1600	1½ in. square to 2½ ins. square	5 " "	215	195	170
		6 " "	190	170	150
		9 " "	170	150	125
2000	1½ in. square to 3½ ins. square	5 " "		170	150
		6 " "		160	135
		7 " "	190	190	150
		9 " "	190	190	130
2500		5 " "			130
		6 " "			130
		7 " "	190	215	150
		9 " "	170	170	120
		10 " "	170	155	85
		13 " "	125	130	65
3000	2 ins. square to 4 ins. square	9 " "	150	130	70
		11 " "	135	85	70
		13 " "	125	85	65

Drop forging mainly applies to the production of articles required in very considerable quantities, but this system of using multiple impression dies should be applied where possible, even in connection with a modest production, that is to say, in railway shops—for example, where orders are only given out for 200 or 300 at a time of each particular pattern. In such cases the design of the guide rods of the hammers must be sufficiently strong to give the required assistance and accuracy of adjustment.

The Table on page 1122 gives an idea of the space, etc., required by drop stamps.

See also Descriptive Section XXII, Part III:
Alldays & Onions Ltd.
Brett's Patent Lifter Co., Ltd.

DIMENSIONS OF BRETT BILLINGS AUTOMATIC BOARD DROP HAMMERS.

Weight of hammer. (lbs.)	Floor space R. to L. F. to B.	Height to operate hammer.	Longest stroke.	Shortest stroke.	H.P. to drive.	Motor revs. per min.	Motor pulley diam.	Counter-shaft revs. per min.	Counter-shaft pulley diam.	Diam. of driving pulleys.	Hammer pulley revs. per min.	Lifting speed ft. per min.	Blows per min.
	ft. in. ft. ins.	ft. ins.	ft. ins.	ft. ins.			ins.		ins.	ins.		ft.	
400	3 6×2 3	15 6½	4 6	3 0	7·5	680	8	309	17½	16	190	497	80
500	3 6×2 3	15 6½	4 4	2 10	7·5	680	8	309	17½	16	190	497	80
600	4 0×2 6	16 0	4 0	2 6	10	565	10	255	22	21	180	471	75
800	4 0×2 6	16 0	3 9	2 3	10	565	10	255	22	20	170	445	75
1000	4 9×3 4	17 8½	4 8½	2 8½	15	680	10	232	29	20	145	456	75
1200	4 9×3 4	17 8½	4 7	2 7	15	680	10	222	30	23	145	456	75
1400	5 0×3 4	17 10	4 9	2 7	20	565	10	210	27	28	140	440	70
1500	5 1×3 4	17 10	4 8	2 6	20	565	10	210	27	28	140	440	70
1600	5 1×3 4	17 10	4 7	2 5	20	565	10	210	27	29	140	440	70
1800	5 6×3 4	18 7	4 11½	2 9	25	565	12	210	32	26	115	362	60
2000	5 7×3 5	18 7	4 9½	2 7	25	565	12	210	32	26	115	362	60
2500	7 0×4 6	18 7	4 7½	2 5	30	570	13	200	37	26	110	346	55
3000	7 0×4 6	19 7	5 0	2 9	30	565	13	200	37	25	90	330	50

SECTION XXIII

PART I

METALLURGY (pp. 1125–1265)

Revised by Edwin Gregory, Ph.D., M.Sc., M.I.E.I., F.I.M., F.R.I.C., and W. J. Driscoll, B.Sc. (Eng.), A.M.I.Mech.E.

PART II

WELDING AND CUTTING (pp. 1267–1296)

By D. Richardson, Wh.Exh., A.M.I.Mech.E.

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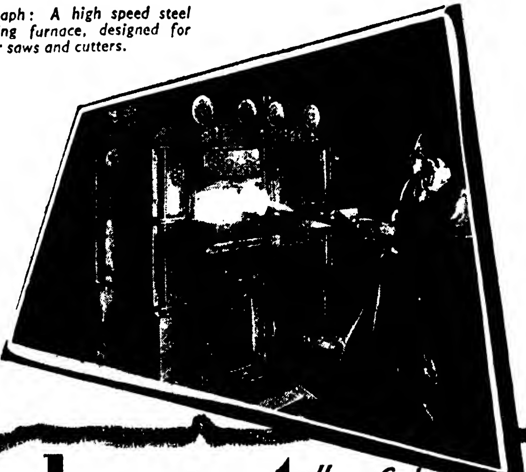
Photograph: A high speed steel hardening furnace, designed for circular saws and cutters.



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SECTION XXIII
PART I
METALLURGY

Revised by Edwin Gregory, Ph.D., M.Sc., M.I.E.I., F.I.M., F.R.I.C.

FURNACES.

CLASSIFICATION OF FURNACES.

FURNACES IN WHICH THE CHARGE AND THE FUEL ARE INTERMINGLED.

1. Hearth Furnaces.

(*a*) With blast. Forge for smithing. Grate for zinc-white manufacture. Furnaces (Swedish) for the manufacture of wrought iron of high quality.

(*b*) Without blast. Heap and stall for coking and roasting. Trough for liquidation and drossing. Siemens-Martin open-hearth furnaces for steel manufacture.

2. Shaft Furnaces.

(*a*) With forced draught. Blast furnaces for smelting ores of iron, copper, lead, etc. Cupolas for iron-foundry practice. Bessemer converters.

(*b*) With induced draught. Only rarely employed.

(*c*) With natural draught. Furnaces for smelting readily fusible ores. Kilns for calcining, roasting, etc.

FURNACES IN WHICH THE CHARGE AND THE FUEL ARE SEPARATE.

1. Reverberatory Furnaces (Radiation).

(*a*) With forced draught. Furnaces for the refining of metals and the smelting of ores.

(*b*) With natural draught. Furnaces for the roasting of ores. Furnaces for the manufacture of puddled iron, etc.

2. Crucible, Retort, Muffle, etc., Furnaces (Conduction).

(*a*) With solid, liquid or gaseous fuel.

(*b*) With electricity.

FURNACES IN WHICH THE CHARGE SUPPLIES ITS OWN FUEL.

(*a*) Preheated furnaces with solid charge. Kilns for ore-roasting. Coke ovens.

(*b*) Preheated furnaces with liquid charge. Converters for refining, etc.

(*c*) Non-preheated furnaces with solid fuel. Blast-roasting furnaces with up and down draught.

ELECTRIC FURNACES HEATED BY RESISTANCE EFFECTS.

(*a*) The charge is included in the electric circuit. Héroult aluminium furnace. Kjellin induction furnace.

(*b*) The charge is heated by means of an electrically-heated conductor.

(*c*) The charge is heated by eddy currents induced in it by means of a high-frequency inductor coil.

ELECTRIC FURNACES HEATED BY ARC EFFECTS.

(*a*) Direct heating. The charge forms one or both poles of an electric arc. Héroult, Siemens-Schuckert furnaces, etc.

(*b*) Indirect heating. The charge is enclosed in or below a space heated by an electric arc. Stassano and Detroit rocking arc furnaces.

ELECTRIC FURNACES HEATED BY COMBINED RESISTANCE AND ARC EFFECTS.

Snyder, Girod, Greaves-Etchells, etc., steel furnaces.

BLAST FURNACES.

(C. Livingston.)

A blast furnace should be designed to suit the materials used in smelting; successful designs can best be accomplished by keeping in view the best practice with particular materials. No general formula for the design of all blast furnaces, for a given output, is possible. The following points are common to all successful blast-furnace plants, and should be kept in view, in new construction :—

The aim of all blast-furnace work is obviously the production of a good marketable iron at the lowest cost. Low costs rest chiefly on low fuel consumption ; low fuel consumption is governed by the quantity of fuel, dryness and temperature of blast, with correct furnace lines for the ores smelted.

Blast temperature is entirely one of stove capacity ; it is essential that stove capacity should be on the generous side.

Output will depend on capacity of furnace, diameter of hearth, and will vary according to materials used, quantity and pressure of blast, dry or wet blast, temperature of blast ; blast pressure should be able easily to penetrate to centre of furnace.

Bosh line angles are found to vary very little ; short steep bosh lines are a decided advantage with most materials. Leading American blast-furnacemen attribute some of their remarkable results, in part, to low steep boshes and large hearths. Furnaces with hearths 20 ft. 9 ins. in diameter are in successful operation.

In general practice, furnace shafts should have a batter of at least 0·625 in. in 12 ins. ; where fine ores are smelted this could be considerably increased with advantage. Diameter of bell should be 4 ft. to 4ft. 6 ins. smaller than diameter of throat.

The furnace should be entirely enclosed in a strong steel shell, from the hearth level up.

All parts of furnace top work should be securely fixed to main shell structure.

In designing blast-furnace details the main considerations should be (1) efficiency, (2) strength, (3) simplicity.

The number of gas offtakes should be such that the gas leaves furnace as slowly as possible, so as to avoid carrying dust over to flues.

The height of furnace varies from 80 to more than 100 feet ; with the materials used, each district has come to fix the height found most suitable for its particular conditions.

Sufficient relief area, by bleeders or explosion doors, should be provided on top of furnace to prevent injury from slips ; the relief should, if possible be directly upwards.

For hearth well and boshing, nothing but the best quality of fire-brick should be used.* The bricks should not be too large, so that they may be thoroughly burned. For the upper lining a fire-brick with more wear resisting qualities can be used. Care should be taken that all bricks when laid are properly bedded and tight built, with fireclay joints as thin as possible. In relining a furnace, the bear in old hearth need not be removed, but simply levelled to start building well on.

Some means should be taken to preserve the upper lines of furnace from wear where stock from bell strikes brickwork ; a satisfactory way of doing this is to insert with brickwork alternate layers of cast iron or steel plates. In normal working, the stock level should be kept just high enough to allow bell to swing freely when full open.

The number of tuyeres is not so important (if sufficient area for output is provided) as equal distribution and penetrative power of blast. There should always be a reserve of blowing power available after providing for normal maximum requirements.

Where blast-furnace gas is fully utilised and gas economy a necessity, the blowing should be done by gas engines. Where there is a surplus of gas the turbo-blower can be installed with advantage, and will prove an ideal blowing engine. Reciprocating steam-blowing engines are still being installed, but only in cases where conditions make it unsuitable or inconvenient to discard existing plant.

Cooling.—Apart from tuyeres, jumbos, and hearthcooling, any cooling appliances provided for the bosh or upper lining should be carefully considered ; external or spray-cooling has the great advantage that leakage of water into the furnace is prevented, or rather more easily detected. The trouble caused by water leakage from blocks inserted in the building is considerable, and can never be fully known ; it is often when the damage is done the leak is located. Modern practice is to spray-cool the bosh and sometimes the entire outside shell ; by this means a much thinner lining can be used, the furnace preserves its original lines longer, and the lining has a longer life.

Stoves.—Should be as simple in design and equipment as possible, with ample facilities for cleaning and inspection, and have a good chimney draught.

Using unwashed gas, chequers should not have less than 9-in. by 9-in. openings ; with washed gas smaller chequer openings can be used, giving greatly increased heating surface for same size of

* Carbon or ' graphite ' blocks are now being extensively employed in bosh linings.

shell, and higher blast temperatures and lower chimney temperatures for same gas consumption. Openings 3½ ins. × 3½ ins. with 2-in. thick walls have proved satisfactory in practice.

In latest practice for heating stoves the gas and air is forced by special fan into combustion chamber, the stove burns more gas, which gives higher and more regular temperatures, with a saving up to 25 per cent. in stove construction for a given production.

Preheating the blast before entering the stove proper is in successful operation with increased average temperatures. It is no advantage to go beyond 1,550° F. in stove heats, any fuel economy being more than off-set by stove repairs.

It is decided economy to clean blast-furnace gas, as better results are obtained.

Better results with certain ores will sometimes be obtained by running on lower blast heats; this may seem to conflict with theory, but in blast-furnace work it is often the case that sound theory follows successful practice.

Uniform ore mixtures and quality of fuel and flux should be kept in view, rather than frequent changes of ore mixture, with varying qualities of fuel and flux. It is an advantage to break hard dense ores, especially magnetites.

The approximate saving in lbs. of coke per ton of iron by increase in blast temperature is as follows:—

1,000° to 1,100° F. (540°–595° C.) . .	50 lbs.
1,100° „ 1,200° „ (595°–650° C.) . .	40 „
1,200° „ 1,300° „ (650°–705° C. . .	30 „
1,300° „ 1,400° „ (705°–760° C.) . .	20 „

Dry Blast.—Average results with dry blast give a saving of 10 per cent. in fuel consumption with 10 per cent. increase in output, more regular grade of metal, and more regular working conditions.

A blast furnace in its working and results responds readily to proper treatment; proper regulation of flux has much to do with this, and the formation of a correct slag is all important.

The slag should not be too basic or limey; this causes the furnace to become sticky and gob up. If the slag is too acid the furnace will drive harder, but trouble is sure to result with grade and quality of pig.

When the lime content of slag is high it will prove profitable to make such slag into cement; the slag from furnaces making Bessemer iron will generally be found suitable for this purpose. Most blast-furnace slags can be easily made into first-class building bricks and large quantities of these slags are used in road-making.

The slightest indication of irregular furnace working should be promptly attended to and put right; irregular conditions will probably show first in change of slag or irregular settling of stock. The appearance and degree of fluidity of the slag is the furnaceman's best indication of working conditions.

Irregular working at tuyeres is generally caused by water leakage from jumbos and tuyere blocks. It is quite possible for tuyeres to be shut by water leakage, in spite of very careful watching. No tuyere in a regular working furnace will close while the blast is on except by water leakage.

Irregular working at tuyeres is sometimes caused by an alteration in burden; in this case every tuyere will give trouble, which will continue till the trouble has been put right.

The best way to open tuyeres that may have been hard closed by slips, irregular working, or water leakage, is by the application of oxygen; the apparatus used is simple and cheap: the worst stuck tuyere can be opened in a few minutes.

CUPOLAS.

Blast-Quantity and Pressure in Cupola Working.

An average of 120—in terms of quantity of air in pounds, multiplied by $\sqrt{\text{pressure in ounces}}$ is required to melt one ton of ordinary phosphoric iron.

The maximum output of metal melted per hour is more nearly comparative to diameters of cupolas than areas of same.

The maximum output of a cupola will be obtained when the pressure and quantity of the blast in ounces and pounds per minute respectively are so adjusted that $\frac{W\sqrt{P}}{D} = 330$, being the diameter of the melting zone in feet. Any further increase in the product $W\sqrt{P}$ results in excessive oxidation of the metal in the cupola.

Iron melted in a cupola is affected in hardness directly as the pressure of blast used, so that the total area of the tuyeres employed should be regulated according to the class of iron being melted, small areas being used when greater hardness is required.

The height of the cupola from the top tuyeres to the charging-door sill should *not be less* than two and a half to three times the diameter across the melting zone.

The coke consumption in a well-designed cupola need not be more than 0·10 lb. per pound of ordinary phosphoric iron when melted, the coke contains from 90 to 92 per cent. of carbon.

Centrifugal Fans for Cupolas and Smith's Fires.

To give the high pressure required to supply air to Cupolas and Smith's Fires the fans must be of the 'centrifugal' type and suitably designed. They usually have smaller inlets and outlets than those used for ventilating, as pressure, not volume, is the essential feature.

Care is required in the choice of a suitable fan, as the efficiency varies within wide limits with the design.

REVERBERATORY FURNACES.

In a reverberatory furnace the material to be heated is contained in a long horizontal or slightly sloping hearth, the heat being supplied by flames and hot gases which travel over it. A considerable amount of heat is radiated on to the hearth from the arched roof of the furnace. The proportions of the various parts of the furnace are important, as if the roof is too high the gases will not 'lick' the hearth, but will travel close to the roof. Care must be taken to have a sufficient volume of gases in the furnace, as otherwise the upper part of the hearth will be too cool. There is a strong tendency to open the exit damper too wide. When in constant use, a reverberatory furnace heated by producer gas is more economical than one heated by solid fuel.

KILNS.

A kiln is a furnace which is filled almost completely with the articles to be heated. Usually the flames and hot gases from the fuel circulate among the goods and raise them to the required temperature by conduction. When contact with the kiln gases would spoil the goods, they may be enclosed in fireclay cases or a *muffle furnace* may be used.

MUFFLE FURNACES.

A muffle furnace is one in which the articles or materials to be heated are protected from direct contact with flames and hot gases from the fuel, but the term is sometimes used loosely for any kind of furnace heated by town's gas instead of solid fuel. The latter kind of furnace is extensively used for annealing, normalising, case-hardening, tempering, forging, and other heat treatments of iron, steel, and the non-ferrous metals.

The following are figures from actual practice :—

Operation.	Load.	Duration.	Cub. ft. per ton.
Annealing H.S. steel	3½ tons	15½ hrs.	6,000
" steel	3½ tons	8 hrs.	3,000
" C.I. parts packed in filings in boxes	2,000 lbs.	3 hrs.	3,360

CRUCIBLE MELTING FURNACES.

'Lift-out Crucible' or Pit Type Furnaces.—These are supplied where requirements of molten metal are small, or in cases where it is preferable to melt in relatively small capacity crucibles. *Natural Draught Coke Furnaces* can be either connected up to one main flue and chimney or can be provided with smaller collecting flues, each flue having a chimney to serve a small number of fires, or again, each fire can have its own small stack.

Forced Draught Coke Furnaces can be similarly arranged as regards provision for exhausting the waste gases, and as they are not dependent upon chimney pull the heights of stacks are considerably less than those required for natural draught furnaces.

It is usual to supply air from a blower or fan of sufficient capacity to feed any number up to the whole of the plant. Forced draught offers the advantages of (*a*) independence of atmospheric conditions, and, therefore, absence of variation in chimney pull; (*b*) quicker heating qualities of furnaces; (*c*) a greater degree of heat intensity than is possible under all conditions in natural draught furnaces.

Oil-fired Furnaces.*—These can be fitted with burners to work with compressed air at 20-30 lbs. per square inch, depending to a large extent upon the temperature desired, the higher pressure effecting a more perfect atomisation of heavy grades of fuel, or with low pressure burners which run with a pressure of only ⅔ lb. per square inch for all temperatures up to that necessary for melting mild steel and nickel.

The former require an air compressor, which is a relatively expensive machine, and may require 5 to 10 h.p. per burner, whereas the latter needs only the ordinary open high-speed fan, and the burners will take 1½ to 2½ h.p. according to size of the furnace, etc. Almost any grade of commercial fuel oil can be used with either type of burner, and this is fed to the burners by gravity.

Producer Gas can be applied to lift-out crucible furnaces if these are made in the multiple form, viz. several crucibles in one common melting chamber, or several crucibles in separate chambers, but exhausting into one air or gas heating regenerator.

For ordinary low-power producer gas, 135 to 175 B.Th.U. per cubic foot, it is essential to preheat the air or gas supply to about 300° C. for efficient melting of copper alloys; for iron and higher temperature melting, both the gas and air should be supplied at about this temperature.

When using high-power gas, such as water gas of roughly 280 B.Th.U. value per cubic foot, no preheating is required for melting temperature up to about 1,400° C.

Tilting Furnaces *for Coke Fuel* are always of the forced draught pattern, for one of their chief features, quick heating, would be greatly impaired if worked on natural draught. This type of furnace is more efficient than the pit type, and it has the important advantage of melting in greater bulk than can be handled in a lift-out crucible—up to ½ ton of metal at one charge—and the further advantage of making it possible to mechanically tilt the furnace on a fixed front trunnion, so that the metal can be poured direct into ladles or moulds. In the portable pattern the body portion holding the crucible can be transported by means of an overhead crane to the point at which the molten metal is wanted.

Furnaces for Oil Fuel.—The remarks made above in connection with pit furnaces apply also to the tilting pattern. The economy in oil when melting, say, ½ ton of metal in one crucible with one burner, will be very marked, compared with, say, five burners for five 200-lb. size pit furnaces, and the consequent saving in power for air blowers in equal proportion.

Producer Gas can be applied to tilting furnaces as for pit furnaces, but tilting furnaces being of greater capacity than the latter it becomes possible to run single furnaces with producer gas, the heat in the waste gas being utilised to warm the incoming air or gas supply. For copper alloys it is usual only to preheat the air.

The Morgan Crucible Company give the following details regarding their oil-fired tilting furnaces:—

METAL TO BE MELTED.

	Aluminium.		Brass.		Bronze.		Copper and Nickel Brass.		Cast Iron.	
Melting capacity of furnace (lbs.) (Brass basis) . .	150	1,120	150	1,120	150	1,120	150	1,120	150	600
Weight of charge (lbs.) . .	50	375	150	1,120	150	1,120	150	1,120	120	525
No. of melts per 8-hour shift .	15	9	10	6	9	5	8	4	6	3
Oil consumption (lbs. per 100 lbs. metal)	19	12	14	8	16	9	18	12	33	24

For the melting of steel and pure nickel the values for oil consumption are obviously greater. For a single crucible furnace the oil consumed is about 100 lbs. per 100 lbs. of metal, but about 65 lbs. only with larger furnaces where the melting chamber will accommodate four crucibles.

ELECTRIC FURNACES

Electric Furnaces Heated by Resistance Effects.

Furnaces of the type in which the charge is included in the electric circuit (fig. 1) are employed largely in the manufacture of aluminium, carborundum, calcium carbide, graphite, electrodes, etc.

Furnaces of this type in which the charge is heated by means of an electrically heated conductor find extensive employment in laboratories, and are coming into more general use for the annealing and thermal treatment of steel and the melting and heat-treatment of non-ferrous alloys.

* Oil-firing has now superseded producer-gas firing in many open-hearth steel-making plants; quicker smelting and increased production have resulted.

Smaller furnaces, such as are used in laboratories, are often made after the style of gas-heated muffle furnaces, but in place of muffle jets, wire of nickel-chrome alloy (Nichrome) or aluminium-iron alloy (Kanthal, p. 1220), heated by the passage of an electric current, is wound round the body of the chamber. Other types incorporate a helix or spiral of carbon, or a series of 'Silit' (carborundum) rods, through which the current flows. Industrial forms of resistance furnaces of these types are available for the heat-treatment of steel and other alloys.

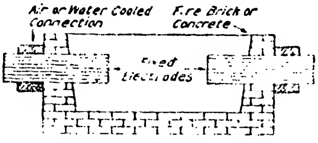
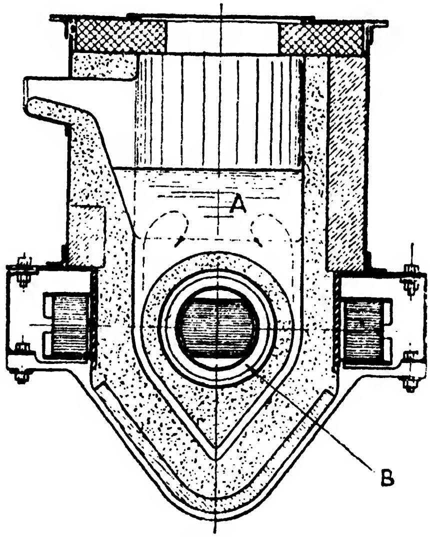


FIG. 1.—Cross Section of Resistance Furnace.
Current passes from one electrode through the charge to the other electrode.

These furnaces are provided with separate heating elements, which do not come into contact with the charge, but are made incandescent by the current passing through them, and thus heat the interior of the furnace chamber.

Induction furnaces, which may be classed under this head, find employment in the manufacture of certain steels and non-ferrous alloys. Low or normal frequency furnaces are not now much used in steel-making, but furnaces of the Ajax-Wyatt type (fig. 2) are extensively employed



Electric Furnace Co., Ltd.
FIG. 2.

in brass-melting. B is a copper coil, or core, through which alternating current is passed, and this acts as the primary of a transformer, the narrow V-shaped slot constituting the secondary. The induced current in the secondary generates heat in the metal and sets up a vigorous circulation, so that a head of molten metal is soon established above the slot. The molten metal in the slot is, under all circumstances, hotter than that above it, so that convection currents are induced in the molten charge. The average melting costs per ton for 60/40 brass in a 600 lb. furnace in 1938 were given as follows :—

	s.	d.
Electricity, 224 kW.h. at $\frac{1}{4}d$.	9	4
Lining of furnace and repairs	1	6
Labour, 1 man per shift at £5, making 12 tons per week per shift	8	4
	19	2

The principal advantages of this type of furnace are low melting losses, low power consumption, and complete mixing of the charge. The power factor lies between 0·80 and 0·85, according to the frequency of the power supply.

One great disadvantage of this type of furnace is the need for the maintainence of molten metal in the slot at all times. The low frequency furnace is thus unsuitable for intermittent working.

The high-frequency or 'coreless' induction furnace is the latest development in the methods for the melting of metals and alloys. It is essentially a crucible furnace (fig. 3), in which the

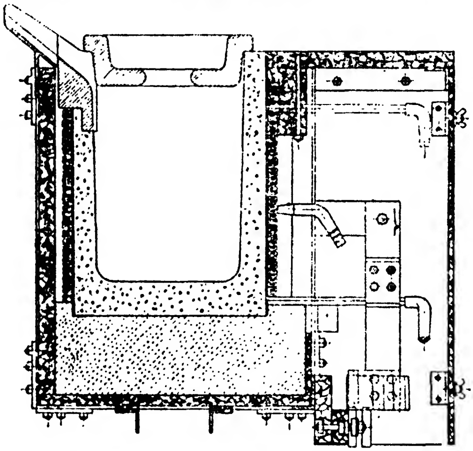


FIG. 3.—Ajax-Northrup High-Frequency Furnace.

charge is melted by the eddy currents induced in it by the passage of a high-frequency current through a water-cooled copper coil surrounding the crucible or container of the molten metal. Owing to the unreliability of crucibles, fritted linings are more commonly used in industrial units. The furnace is lined by tamping dry refractory grist between the coil and a hollow metal 'former,' having the external shape desired for the interior of the lining. When current is passed the former is heated and eventually melts, but long before this occurs the surface layers of the lining are fritted into a hard coherent mass capable of holding molten metal. The refractory near the coil remains as an unchanged grist or powder and acts as an effective heat insulator. By this means an acid, basic, or neutral lining may be built up as circumstances demand.

To some extent the frequency depends on the capacity of the furnace. With small experimental furnaces capable of melting a few grams only, the frequency is of the order of 1,000,000 cycles at 7,000 volts pressure. With more commercial types, capable of melting between 10 lbs. and 5 tons, the furnaces operate with frequencies between 500 and 2,000, according to their design. For efficient melting it has been shown that the ratio r/β must be greater than 3, where r is the radius of the charge in centimetres and β is given by the expression

$$\beta = \sqrt{\frac{\rho}{4\pi p}}$$

ρ is the resistivity of the charge and p the pulsatance (2π times the frequency) in radians per second. In all cases, very high voltages are essential so that a step-up transformer becomes a necessary unit. The induced eddy currents in the charge create a stirring of the molten metal the lower the frequency and the greater is the circulation for a given coil diameter. This results in the centre of the molten charge being raised above the outer layers, a phenomenon described as the fountain effect. The automatic stirring of the molten charge results in uniform dissemination of its constituents, an important point in the manufacture of alloys such as stainless and high-speed steel. The estimated cost of melting one ton of steel in a furnace of 5 cwt. (150 kW.) capacity pre-war was about £2 5*s*., the current consumption being about 700 kW. hours. With furnaces of larger capacity the power consumption per ton is lessened considerably. Furnaces up to 7 tons capacity have now been installed.

Induction furnaces are able to hold their own against other types of steel-melting furnaces on account of their steady current consumption and low power consumption in the melting of cold stock, but suffer from certain disadvantages of a well-defined character. The heat being generated in the metal and only conducted to the slag, renders it difficult to work a fluid basic slag, with the result that desulphurisation of the charge is not so efficiently carried out as in the case of furnaces of the arc-type. Nor can dephosphorisation be so perfectly attained, since the design of these furnaces militates against ready and complete removal of the oxidising slag needed for the removal of phosphorus. A further disadvantage arises as a result of the continuous circulation of the molten metal in the bath which leads to the erosion of the lining.

Electric Furnaces heated by Arc Effects.

These fall into two classes (*a*) indirect arc furnaces, where the current arcs from one electrode to another above the charge, which is thus melted by radiation and (*b*) direct arc furnaces, where the arcs are struck between the extremities of vertically suspended electrodes and the liquid or solid charge (fig. 4). In some furnaces of the latter type electrodes are built into the furnace

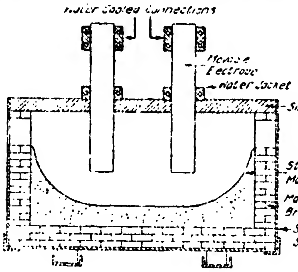


FIG. 4.—Cross Section Direct Heating Arc Furnace.
Current arcs from one electrode to charge and from charge to other electrode.

bottom so that current is forced to flow through the bath (*e.g.* Greaves-Etchells furnace) by which it is claimed that more uniform heating of the bath is obtained. Indirect arc furnaces are not now used to any great extent in the melting of steel, although the single-phase Detroit furnace (rocking arc) is extensively employed for melting brass, cast iron, etc. For the melting of steel, 3-phase current is largely employed, and the electrodes, which are of graphite or amorphous carbon, are controlled, either by hand or electrically, so as to yield the desired arc-length for melting, refining or heating of the charge.

Acid-lined furnaces are employed in the melting of high-grade materials for the manufacture of steel castings. By far the largest number of arc furnaces operate with basic linings, however, and thus the proportions of sulphur and phosphorus in the final product may be reduced, if initially not more than 0·06 per cent., to little more than traces. In the ordinary process, where it is desired to lower the percentages of both elements, the charge consists almost entirely of steel scrap. This is melted under a basic oxidising slag, which is then removed by tilting the furnace; the bath, if necessary, is then recarburised by the addition of crushed electrode, anthracite coal, etc., and finally refined under a '*white*' *slag*, composed of lime, fluorspar, coke-dust and crushed ferro-silicon, which removes sulphur and thoroughly deoxidises the molten metal. When in proper condition this slag, on extraction from the furnace, undergoes an enormous change in density on cooling and disintegrates into a fine white powder. It is then described as a '*falling*' *slag*. Unlike most of the other steel-making processes the final additions of ferro-manganese, ferro-silicon, etc., are added at the end of the arc-furnace process merely to bring the steel up to specification values, and not for purposes of deoxidation. In consequence, arc furnace steel is clean and free from non-metallic inclusions. The steel is thus of excellent quality and its composition can be very accurately controlled. For the manufacture of high-speed steel, stainless and heat-resisting steels the furnace is mainly used for melting selected high-grade materials. No slag removal then becomes necessary but automatic desulphurisation and deoxidation occurs during the finishing stage of the process.

Furnaces with capacities of from ½ to 30 tons are in operation in this country chiefly of the Héroult, Greaves-Etchells, 'Efco' or 'Lectro-Melt types. The current consumption varies with the capacity and furnace design, and with the method of working. Average figures, starting with cold scrap charges, are as follows :—

		3½ tons Héroult.	*25 tons high-voltage Arc furnace.*
Duration of Heat .	**Melt**	**2½ hrs.**	**3 hrs.**
	Refining	**1 to 1½ hrs.**	**2 hrs.**
Current consumption .	**Melt**	**550 kW. hours per ton**	**450 kW. hours per ton**
	Refining	**280 kW. ,, ,, ,,**	**230 kW. ,, ,, ,,**
Voltages . . .	**Melt**	**90 at 5,000 amps.**	**180 at 15,000 amps.**
	Refining	**78 at 2,000/5,000 amps.**	**104 at 2,000/8,000 amps.**

(*Gregory.*)

The chief **losses** in electric furnaces are due to radiation from the walls and conduction through the electrodes. Since the radiation losses increase as the square and the capacity as the cube of the linear dimensions of furnaces, it is obvious that large installations are more economical than small ones. Conduction losses can be reduced to a minimum by suitably designing the electrodes.

Although, with coal at 14*s.* per ton* and electrical energy at ½*d.* per kW.-hour,* the latter costs twenty-eight times as much as the former per British thermal unit purchased, the efficiency with which the electrical energy is applied is so much higher that the difference is largely neutralised, and in the case of crucible steel is completely wiped out. In addition, the labour cost is greatly reduced with electric furnaces and there is a considerable economy of space and buildings, while the steel produced is of higher quality and uniformity than by any other method.

A useful method of calculating heat flow through the furnace walls, etc., has been developed by Carl Hering on the lines of Ohm's law. If—

W = heat flow in watts ; T = drop in temperature in degrees C. ; R = thermal resistance in 'thermal ohms,'

then,

$$W = T/R.$$

A thermal ohm is that thermal resistance which requires a drop of temperature of 1° C. to produce one watt of heat flow and = 4·2 gramme-calorie units ; 1 gramme-calorie unit of thermal resistance = 0·24 thermal ohm.

Assuming that the electrodes are of uniform section, and that no lateral loss of heat takes place—conditions which are only approximately fulfilled in practice—Hering finds that the total loss of heat through the cold end of the electrode equals the sum of the loss by heat conduction alone (*i.e.* when no current is flowing) and half the T^2R loss. This combined loss is least when the loss by heat conduction alone equals half the T^2R loss ; the total loss then equals the T^2R loss, and no heat will be conducted out of the furnace by the electrodes. This minimum loss is dependent only on the material of the electrodes, the current, and the temperature, but not on the absolute dimensions ; it fixes the ratio of cross-section of electrode to length. In the case of a furnace working at 1,400° C., the cool end of the electrode being at 100° C., the length of the electrode 20 ins., and the current 10,000 amperes, the following dimensions were given by Hering's method of calculation :—

Carbon, 12·5 ins. diameter, loss of power in electrode 23·4 kW. ; Graphite, 7·1 ins. diameter; loss of power 15·1 kW. ; Iron, 4·8 ins. diameter, loss of power 4·6 kW. (*Stansfield.*)

The power, etc., provided in connection with Héroult furnaces used by the United States Steel Corporation are given in the following table, which applies to cold charges ; if the charge is put into the furnace in a molten condition about half this power suffices :—

POWER REQUIRED FOR ELECTRIC FURNACES.

Size of Furnace, tons.	Transformer capacity, kVA.	Amperes, per phase.	Square inches Copper, per phase.
1	375	2,250	3
2	600	3,600	5
3	750	4,500	6
4	900	5,400	7
6	1,200	7,200	9
10	2,000	12,000	16
15	3,000	18,000	24

* These are pre-war values and cannot properly be amended until the true results of nationalisation are stabilised.

The voltage used in arc furnaces depends upon the current and the length of the arc. In three-phase furnaces it ranges between 65 and 180 volts, and in single-phase furnaces between 150 and 200 volts. In resistance furnaces the voltage at starting, with a cold charge, may be 400 volts, decreasing as the charge heats up to 35–60 volts. The consumption of the electrodes varies widely according to the conditions obtaining. For steel production, starting with cold scrap charges, average values are between 8 and 10 lbs. per ton, using graphite electrodes and 14 to 18 lbs. per ton with amorphous carbon electrodes. The furnace bottoms last from 1 to 3 years, the walls from 100 to 300 heats, and the roof from 50 to 200 heats. To make ordinary carbon steel of crucible quality with a 6-ton furnace requires from 550 to 750 kW.-hours per ton, starting cold. In smaller furnaces the input per ton is greater.

The power factor of the arc resistance furnace varies from 75 per cent. to more than 90 per cent.: that of the resistance type from 90 to 97 per cent.: that of the induction furnace varies from 50 to 60 per cent. only and that of the coreless induction furnace is less than 15 per cent.

The leads from transformers to electric furnaces should be as short and direct as possible, and close together, as the voltage drop due to their self-inductance and the very large currents is considerable, resulting in a low power factor. The conductors should be wide thin bars, or tubes, to prevent loss of energy due to the tendency of alternating current to flow in the skin of the conductor.

ELECTRICAL RESISTIVITY OF MATERIALS, OHMS PER INCH CUBE.

Material.	Cold.	Hot.	Temperature ° C.
Graphite	0·00034	0·00031	100–2000
Carbon	0·0018	0·0014	100–2000
Silicate slag	∞	1 to 2	Molten
Fire bricks (various)	∞	8 to 24	850
		24 to 300	815

(*Stansfield.*)

FURNACE CONSTRUCTION.

The following are points requiring special attention in the construction of furnaces :—

1. Sections of furnaces such as are exposed to differing physical and chemical actions should be united in such a manner as to allow of free and independent movement and as to be subject to independent condition.

2. Such sections as are exposed to high temperatures should be made thin and should, when possible, be water-cooled; such sections as are exposed to low temperatures should be made of such a thickness as will reduce thermal losses due to radiation and conduction to a minimum.

3. The interiors of furnaces should be constructed so as to resist mechanical, physical and chemical wear. With this end in view choice must be made of suitable materials of construction and of efficient design. In this connection note should be made of the following factors :—

(*a*) The action of heat.

(*b*) The erosive and corrosive actions of flame, slag, etc.

(*c*) The mechanical wear due to abrasion by the solid charge, etc.

4. The exteriors of furnaces should be supported and strengthened in such a way as to allow of the free expansion and contraction of the body. The foundations of all furnaces must be of a strong and rigid character.

MELTING POINTS OF REFRACTORY MATERIALS.

	° C.
Alumina (pure)	2050
Bauxite brick	1565 to 1785†
Chromite	1900 to 2200
Fireclay brick	1550 to 1750
Ganister	1700 to 1800
Magnesia brick	2500 to 3300
Silica—Pure	1830*
Pure, with 14·5 per cent. alumina	1690*
Pure, with 63·0 per cent. alumina	1890*
Brick	1700 to 1780†
Zirconia	2800 to 3400

The melting points have been considered to be those temperatures at which the material in question was first seen to flow easily.

* Boudouard. † Kanolt—'Iron and Coal Trades Review,' vol. lxxxv. p. 309.

FURNACE EFFICIENCY.

The thermal efficiency of furnaces may be stated either absolutely or relatively :—

$$\text{Absolute furnace efficiency} = \frac{\text{Net thermal effect produced}}{\text{Calorific power of the fuel}}.$$

$$\text{Relative thermal efficiency} = \frac{\text{Net thermal effect produced}}{\text{Actual calories realised}}.$$

Owing to the large variation in the actual number of calories realised in individual furnaces it is general to employ the former of the above ratios. In order that a determination of the absolute efficiency of a given furnace may be made, a knowledge of the heat-absorbing and the heat-producing factors that are likely to be encountered in practice, of which a number are tabulated below, is required.

(*A*) Heat-absorbing Factors.

1. Heat absorbed by fume, gases, etc.
2. Heat removed by such solids and liquids as are periodically or continuously removed from the furnace.
3. Heat lost by radiation and conduction.
4. Heat absorbed owing to endothermic reactions within the furnace.

(*B*) Heat-producing Factors.

1. Heat supplied to the furnace by the combustion of fuel either inside or outside the furnace.
2. Heat added by such solids and liquids as are periodically or continuously charged into the furnace.
3. Heat gained owing to exothermic reactions within the furnace.

The thermal efficiency of the furnace is represented by the ratio of the heat-absorbing factors to the heat-producing factors.

Heat Economy by Insulation.

A considerable reduction in the loss of heat by radiation (see *A*, 3, above) may be effected by building a 'wall' of insulating bricks between the lining and the exterior of the furnace.

Latent Heat of Fusion of Metals.

The latent heat of fusion is the number of gram calories required to convert one gram of substance from solid into liquid without change of temperature :—

Mercury	2·8 calories.	Platinum	27·1 calories.
Lead*	5·5 ,,	Sodium	31·7 ,,
Bismuth*	10·2 ,,	Palladium	36·3 ,,
Cadmium*	10·8 ,,	Manganese*	36·7 ,,
Tin	14·3 ,,	Magnesium	46·5 ,,
Potassium	15·7 ,,	Antimony*	38·9 ,,
Gold*	15·9 ,,	Copper	50·5 ,,
Iron	49·0 ,,	Nickel	73·0 ,,
Silver*	26·0 ,,	Cobalt*	58·2 ,,
Zinc*	23·0 ,,	Aluminium	93·0 ,,

Heat and Electrical Energy required for Fusion of Metals.

The following table (after Stansfield) gives the heat and electrical energy required to melt various metals, per lb. :—

	Calb.†	Watt-hours.		Calb.†	Watt-hours
Tin	28	15	Copper	162	85
Lead	16	8	Cast iron (gray)	245	129
Zinc	68	36	Tool steel (1 % C.)	300	158
Aluminium	258	136	Wrought iron	343	181
Brass	130	69			

TEMPERATURE CONTROL OF FURNACES—PYROMETRY.

Determination of Calorific Intensity.

The calorific intensity of a substance is the theoretical maximum temperature attained during the perfect combustion of the same in air at 0° C. and 760 mm. pressure, and is inversely proportional to the time occupied in attaining this temperature. The calorific intensity of a substance may be determined by calculation or by experiment.

* Values after Wüst, *Stahl und Eisen*, 1918, 38, 177.

† *Calb.*—The amount of heat required to raise one pound of water through one degree Centigrade.

CALCULATION.

Let C = the calorific power of the substance,
T = the rise in temperature during combustion,
$w, w', w'',$ — = the weights of the products of combustion,
$s, s', s'',$ — = the specific heats of the products of combustion,

then,

$$T = \frac{C}{ws + w's' + w''s'' + \text{etc. etc.}}$$

for the fuel at 0° C. and 760 mm. pressure.

The calorific intensity of a fuel (by formula) will be seen to be directly proportional to the calorific power of the fuel and inversely proportional to the weights and the specific heats of the products of combustion.

The calorific power can be raised by
(a) Superheating the fuel.
(b) Superheating the air needed for the combustion of the fuel.
(c) Using a minimum of excess air.
(d) Increasing the proportion of oxygen in the air.
(e) Reducing the proportion of incombustible material in the fuel.

Expansion Type Thermometers.

The mercurial thermometer is most important for use at about atmospheric temperatures, largely on account of the wide range of temperatures through which mercury is liquid, −39° C. to 357° C. The Beckmann thermometer with a range of 6 degrees may be read accurately to 0·001° C. By the use of inert gases (nitrogen or carbon dioxide) under pressure in the tube of a thermometer, the boiling point of the mercury may be raised to allow temperatures up to 540° C. to be determined, at which temperature the glass softens. Quartz thermometers containing mercury under pressure can be used up to 700° C. By using tin instead of mercury the maximum limit may be raised to 1000° C. Steel tube mercurial thermometers, in which the capillary is attached to a form of a Bourdon pressure gauge, are not easily broken, and may be relied upon up to 540° C. These are particularly suitable for oil, lead and salt tempering baths, tinning and galvanizing baths, etc.

Fusion Pyrometers.

Observations of the fusion of metals, alloys, salts or mixtures of salts enable the observer to judge the temperature reached in a furnace at the moment of fusion, given that the temperature of fusion of the material employed is known prior to its introduction into the furnace. Metals and alloys are not much used now, but salts and mixtures of salts are still employed.

The following are the melting points of a number of metallic salts :—

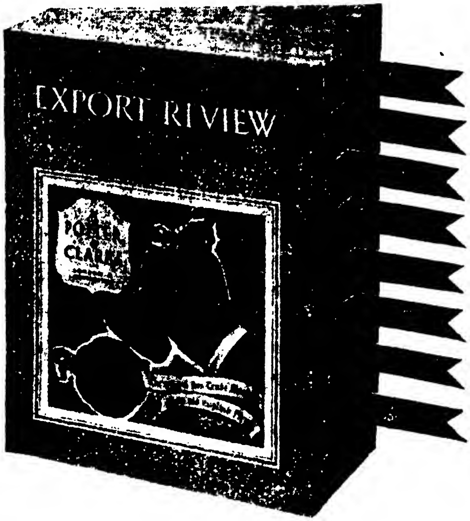
SALT.	MELTING POINT.	
Sodium nitrate	310° C.	590° F.
Potassium nitrate	338 ,,	640 ,,
Calcium nitrate	561 ,,	1042 ,,
Strontium nitrate	570 ,,	1058 ,,
Barium nitrate	593 ,,	1099 ,,
Potassium iodide	705 ,,	1301 ,,
Potassium bromide	750 ,,	1382 ,,
Potassium chloride	790 ,,	1454 ,,
Sodium chloride	800 ,,	1472 ,,
Sodium carbonate	849 ,,	1560 ,,
Sodium sulphate	880 ,,	1616 ,,
Barium chloride	960 ,,	1760 ,,
Potassium sulphate	1050 ,,	1922 ,,

MIXTURES OF SALTS.

Under this heading may be included two important types of fusion pyrometers :—
(a) Séger cones.
(b) Sentinel pyrometers.

SÉGER CONES.

These are small three-sided pyramids, each about 2½ ins. high and about ½ in. wide at the base The term ' cone ' is rather misleading, as they are more pyramidal than conical. They are made of a finely ground mixture of felspar, china clay, and flint, the composition of the mixture depending on the temperature to be indicated. For the lower members of the series other mixtures are used.



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Their shape and composition are such that when heated to their indicating temperature at a reasonably slow rate and under suitable conditions they bend over slowly from the upright position possessed by them when placed in the kiln or oven and gradually form an arch. When the top has almost reached the same level as the base the indicating temperature is reached. A further rise in temperature will cause the cone to collapse and form a shapeless, glazed mass.

It is customary to employ three different cones at a time; the first will indicate a temperature about 20 degrees Cent. below the one required to be known, the second cone will indicate the desired temperature (such as the finishing temperature of a glaze) and the third cone will serve as an indicator that the required temperature has not been exceeded by 20 degrees. The composition of the cones is such that the difference in temperature shown between two successive cones is about 20 degrees, though they are not quite regular in this respect.

It is important to remember that Séger cones do not serve as strict indicators of temperature in the sense that a thermometer does. If heated too rapidly a much higher temperature may be reached than is normally required to bend a cone, and which they are supposed to indicate. Under ordinary conditions of use, however, it is easy to find which cones correspond to the temperatures or conditions of heating which it is required to control, and Séger cones are, therefore, invaluable when they are rightly used. They serve as excellent indicators of the effect of heat on pottery and similar materials, and as it is this, rather than the mere momentary temperature which is important, they have advantages not possessed by a pyrometer or high temperature thermometer.

Sentinel Pyrometers (*Brearley*).

These are small cylinders, measuring about ¾ in. by ½ in. and manufactured from salt mixtures of definite melting points. Only such mixtures can be utilised as are formed of salts that neither dissociate nor corrode when in the molten state. Brearley states that mixtures of salts, after being melted together and ground to a very fine powder, may be made into an adhesive paste with vaseline, and applied to many purposes for which no ordinary pyrometer is available. In the smith's hearth, for example, a tool may be heated to low redness and smeared near the point with a small portion of a paste whose melting point is, say, 770° C. The tool is then reheated directly in the fire or protected from the coke by a piece of scrap, wrought or cast iron piping, until the white mark left by the thin layer of the salt mixture disappears. This indicates that the steel itself has nearly reached the desired temperature and may be quenched.

The following are the melting points of a number of mixtures of salts:—

Salt.	Parts by Weight.	Salt.	Parts by Weight.	Deg. C.
Sodium nitrate	55	Potassium nitrate	45	205
Potassium carbonate	50	Potassium chloride	50	580
Sodium chloride	30	Ditto	70	625
Ditto	42	Ditto	58	655
Potassium sulphate	20	Sodium sulphate	80	825
Ditto	30	Ditto	70	830
Ditto	50	Ditto	50	850

Heat Radiation Pyrometers.

In this type of instrument the relation between the effect of radiant heat on suitably chosen thermal elements and the temperature of the radiating body is observed. The Féry Radiation Pyrometer is an example of this class of instrument.

In this pyrometer the rays from the hot body are received on a concave mirror, which can so be adjusted that the rays form a heat image which covers a small sensitive element permanently fixed on a line through the axis of the mirror. The sensitive element consists of a delicately adjusted thermo-couple, which delivers current to a millivoltmeter, calibrated, if desired, to read the temperature of the body, whence the heat rays are emanating, directly. In both cases suitable means are provided for ensuring that the sensitive element is completely covered by the image of the radiant body.

In order that correct results may be obtained when using pyrometers of the radiation type it is necessary that the hot body sighted should be enclosed within a vessel whose walls are approximately at the same temperature as the body itself. If this precaution is not observed the reading obtained will be lower than the true temperature of the body, the degree of inaccuracy being dependent almost entirely on the nature of the surface of the body in question. When sighted on to bodies in the 'open' the error may amount to as much as several hundreds of degrees.

Optical Pyrometers.

Approximate estimations of the temperatures of incandescent bodies may be made by visual observation. Numerous colour scales have been proposed. but those due to Howe and Taylor and White are generally accepted as the most accurate. The following table is of value in this connection.

Taylor and White.		Howe.	
Name of colour.	Degrees C.	Name of colour.	Degrees C.
—	—	Lowest visible red in the dark	470
—	—	Lowest visible red in daylight	475
Dull red, blood red, low red	566	Dull red	550–625
Dark cherry red . . .	635	—	—
Cherry, full red . . .	746	Full cherry	700
Light red, bright cherry, light cherry . . .	843	Light red	850
Orange	899	—	—
Light orange	941	—	—
Yellow	996	Full yellow	950-1000
Light yellow. . . .	1079	Light yellow. . . .	1050
White	1205	White	1150

The design of optical pyrometers is dependent on the fact that the intensity of the light rays emitted from an incandescent 'black body' is directly proportional to the temperature of the 'body.'

The Wanner photometric pyrometer compares the intensities of the lights emitted by the incandescent body and a standard lamp respectively. In this instrument spectra of the two sources of light are, firstly, broken up into two polarised planes at right angles to one another; secondly, adjusted so that only the narrow red bands of each spectrum are transmitted further; and thirdly, so brought into vision that the intensity of either field, both being in visual contact, can be strengthened or weakened, until they are equally intense, by rotation of a Nicol prism. The amount of rotation, which can be observed on a suitably situated scale, serves to measure the intensity of the emitted light, and hence to define the required temperature. As with the total heat radiation pyrometer, the optical pyrometer only indicates true temperatures when the hot body is enclosed in a heated chamber whose walls are at the same temperature as the body, and when the instrument is sighted on to a body in the open low readings are obtained. The error is considerably less, however, than with a total radiation pyrometer, and corrections may be applied more readily.

The Cambridge Optical Pyrometer is, in principle, the same as the foregoing, and can be used for temperature measurements between 700° and 4000° C.

A most convenient optical instrument is the Disappearing Filament Pyrometer. A hair-pin filament lamp is used as a light standard, the electric current passing through it being adjusted until the image of the filament, viewed through an eyepiece, just disappears into the field illuminated by the hot body whose temperature is sought. The temperature is then indicated on a milliammeter or voltmeter placed in the circuit and which is graduated in degrees C. For calibration and checking the formula

$$C = a + bt + ct^2$$

is employed, where C is the current flowing through the filament (or the voltage drop across its terminals), t is the temperature in °C., and a, b, and c are constants. Three determinations at known temperatures thus serve to evaluate these constants and hence establish the current or voltage-temperature relationship.

Thermo-Electric Pyrometers.

This type of pyrometer is dependent for its action upon the fact that a current is produced within a closed circuit formed of two dissimilar metals when the junction between the two metals is heated, the electromotive force being dependent on the temperature of the junction.

The essential features of this pyrometer are two thin insulated rods or wires of dissimilar metals coupled at one of their ends and connected at their other ends to separate leads. The connection between the two metals and the leads, which is known as the 'cold-junction,' is enclosed within suitably designed apparatus that can be retained at a known constant temperature. The leads from the 'cold-junction' pass to some form of electrical instrument capable of registering or recording changes in the P. D. produced in the now closed circuit when the thermo-couple is heated or cooled.

The number of metals and alloys available for the manufacture of thermo-couples are few. The following are among those most generally employed :—

Couple.	Range.
Platinum-platinum rhodium (10 per cent.)	Up to 1400° C.
Platinum-platinum iridium (10 per cent.)	,, 1000° C.
Nickel chromium (90–10)-nickel chromium aluminium (88–10–2)	,, 1300° C.
Silver-constantan	,, 800° C.
Copper-constantan	,, 500° C.
Iron-constantan	,, 700° C

Union between couple *wires* is effected by fusion in the electric arc or oxyhydrogen flame in the case of couples having a platinum base, by soldering or brasing in the case of base-metal couples or by twisting. Union between the wires and the leads can be effected as above or by means of electrical clamps.

The couple wires, when in use, are generally protected by tubes of nichrome, fireclay, silica or glazed porcelain, the last being particularly useful when it is desired to defend the wires against the deleterious action of hot gases. When necessary, protection is afforded these tubes by enclosing them in steel jackets.

Heavier types of thermo-electric pyrometers consist of a *rod* of one metal fitted tightly into one end of a cylinder concentric with the rod and of another metal and insulated from the same by suitable means.

Calibration of thermo-electric pyrometers is effected by immersing the protected couple in substances or the vapour of substances whose melting or boiling points are known. The couple, being subject to the like thermal changes as the substance, generates an electromotive force, the fluctuations of which are registered by a millivoltmeter within the circuit. For example, while solidification of a metal in which the protected couple is immersed is in progress a constant reading is obtained on the instrument. This value is observed, and is recorded on a graph against the melting point of the metal as ordinate. A series of such points may be obtained by employing a number of metals in similar manner to that already described, and, these points being united, a curve is obtained from which the temperature corresponding to any given reading of the instrument may be abstracted.

The following are the melting points on the thermo-dynamic scale used as standard temperatures in the standardisation of thermometers and pyrometers :—

Mercury	−39° C.	Aluminium	660° C.	Copper free from oxide	1083° C.
Ice	0° C.	Silver-copper eutectic	779° C.	Nickel	1455° C.
Diphenylamine	54° C.	Sodium chloride	801° C.	Iron	1539° C.
Naphthalene	79° C.	Silver	961° C.	Palladium	1554° C.
Tin	232° C.	Gold	1063° C.	Platinum	1773° C.
Lead	327° C.	Copper-copper oxide eutectic	1063° C.	Tungsten	3400° C.
Zinc	419·4° C.				
Antimony	631° C.				

At temperatures of 1000° C. the error is of the order of 0·1° C., at the melting point of platinum 5° C., at that of tungsten 25° C.

In addition to the above the boiling points of water (100° C.) and sulphur (444·7° C.) are employed as standard temperatures.

Platinum-platinum rhodium (13 per cent.) 'immersion' thermo-couples are now extensively employed in the measurement of molten steel temperatures up to 1600° C., and even higher. The thermo-couple 'end' is dipped into the liquid steel for a period of about 30 seconds. Measurement of temperature is then indicated by direct reading on a sensitive potentiometer or a dial-amplifier. Amplification is also used to give permanent pen-records on charts.

Electric Resistance Pyrometers.

In these instruments advantage is taken of the change of resistance of a wire with temperature. Platinum wire is generally employed. The essential features of the commercial apparatus are a coil of fine platinum wire, enclosed in a suitably designed protecting tube, a cell or accumulator for the supply of current to the wire and an indicator or recorder on which the temperature may be read off directly.

In the original type of this instrument errors due to the residual strain and impurities in the lead wires entered, but later instruments are designed with 'compensating wires,' whose component parts are equal in resistance to and are so arranged that they are heated to the same temperature as corresponding parts of the lead wires. Changes in resistance in the lead wires are compensated by equal changes in the 'compensating wires.'

Resistance pyrometers are accurate at temperatures up to 1000° C., and are employed industrially wherever reliable measurement is essential They suffer owing to their high first cost and their fragile character.

METALS AND ALLOYS.

ALLOYS.

There are three well-recognised types of alloys:—

(i) Aggregates of metals, or of metals and non-metals.

(ii) Homogeneous solid solutions of metals, or of metals and non-metals.

(iii) Intermetallic compounds, or compounds of metals with metals or with non-metals.

(i.)

In this type of alloy the component elements crystallise out from the molten alloy in a practically pure state, or in the form of solid solutions of different concentrations, and the addition of the one element to the other results—within certain limits—in a lowering of the freezing point of the resulting alloy. The alloy having the lowest initial solidifying point in such a series is called the 'eutectic.' Lead-tin, lead-antimony, and bismuth-tin alloys are of this type.

(ii.)

In this type of alloy the component metals do not crystallise out from the molten alloy in the pure state, but the resulting alloys contain one kind of solid solution only. Copper-nickel alloys are of this type.

(iii.)

Intermetallic compounds behave similarly to pure metals, crystallising in like manner, *i.e.* their freezing and melting points are identical, so that freezing and melting occurs at constant temperature; with solid solutions and other alloys there is a melting or freezing range.

Intermetallic compounds are usually extremely hard and brittle.

IRON.

IRON (FERRITE).

Chemically pure iron is not an article of commerce. Iron has been obtained by electrolytic deposition of a degree of purity greater than 99·9 per cent. iron, while the American Rolling Mills Co. manufacture a commercial product which they claim to contain 99·84 per cent. of iron, and spectrographically pure iron is now available. Pure iron powder is also being produced by hydrogen reduction in increasing quantities.

Stead, by extrapolation, arrived at the following figures for the approximate mechanical properties of pure iron:—

Yield point	9 tons per square inch.	Elongation .	51 per cent. (length/area = 8).
Ultimate tensile strength	17 ditto.	Reduction in area	84 per cent.

Actual tests on high purity iron gave ultimate tensile strength of 9–12 tons per sq. inch with 100 per cent. reduction of area. (*Adcock and Bristow.*)

The term 'Ferrite' is applied to pure iron when it is considered as a microscopical constituent of iron or steel. Pure iron is composed of polyhedral crystal grains of ferrite.

Carbon, manganese, silicon, sulphur, phosphorus, nickel, molybdenum, etc., all have effect on the iron in which they occur and are frequently added with a view to obtain alloys having more or less definite physical and mechanical properties.

INGOT IRON.

Ingot Iron, unlike Puddled Wrought Iron, is produced in the basic open-hearth furnace, and in view of a certain amount of difficulty in finding an acceptable definition of 'iron' and in differentiating between 'iron' and 'steel' it is well to quote the following definitions advanced by Dr. Albert Sauveur:—

'COMMERCIAL IRON is the element iron as pure as it can be commercially produced.'

'INGOT IRON is commercial iron which has been produced in a fluid condition and cast.'

'WROUGHT IRON is a ferrous metal which is malleable and which has been produced from a pasty condition.'

'**Steel** is an alloy of iron and carbon which is malleable, usually containing substantial quantities of manganese.'

Properties of 'Armco' Ingot Iron.

Chemical Purity.—Contains less than one-sixth of 1 per cent. of impurities considering silicon, sulphur, phosphorus, manganese, carbon, copper, and the gases oxygen, hydrogen, and nitrogen.

Uniformity.—The method of manufacture and its freedom from impurities ensure uniformity of analysis; this, coupled with the fact that the metal is free from slag inclusions, such as exist in even the best puddled irons to a greater or lesser degree, ensures mechanical uniformity.

Corrosion Resistance.—The conclusion that pure iron resists rust is not merely derived from theoretical inference, but represents the definite finding of many observers of the problems of corrosion, after fully investigating generations of service evidence in the form of old hand-wrought irons, such as, for example, the Iron Pillar of Delhi. These old irons were not rustless, but they resisted rust and corrosion to a remarkable degree, by reason of their freedom from corrosion-promoting impurities.

Workability.—Ingot Iron being a dense, ductile, and uniform metal, possesses exceptional working qualities.

Welding Properties.—The superior welding qualities of Ingot Iron are due to its purity and homogeneity. Being of even texture and density throughout, it welds evenly and easily.

Paint-Holding Surface.—It has been found in service that Ingot Iron sheets require painting less often than other iron and steel sheets. This is partly explained by its velvety surface, which grips and holds paint tightly, and partly by the degasification which the metal undergoes during manufacture, thereby reducing to a minimum occluded gases which in course of time tend to become liberated, and causing flaking of the paint film. It is also suitable as a base for vitreous enamelling.

Electrical Properties.—Ingot Iron possesses a high magnetic permeability and low retentivity. It is further a better conductor of electricity than steel; in this connection the following is the result of tests undertaken by the National Physical Laboratory:—

Resistivity

By relative mass = 5·43. By relative volume = 6·17.
(Compared with the British standards for soft copper.)

Ingot Iron can be supplied in the form of black and galvanised sheets, plates, billets, bars, shapes, wires, rods, welding rods, cold rolled strip, and welded tubes.

WROUGHT IRON.

Wrought iron is iron low in carbon content, which in course of production is in a pasty condition, owing to the fact of its being worked at a temperature too low to render it fluid. It contains a relatively large percentage of non-metallic impurities in the form of slag, which may be readily seen when the iron is examined microscopically, and which, therefore, serves, among other features, to distinguish this product from steel. The percentage of carbon in wrought iron rarely exceeds 0·15 per cent., and the total of other elements present in the metal is generally less than 0·25 per cent. Wrought iron is not appreciably hardened by quenching.

The fracture of wrought iron is fibrous in appearance and is occasionally characterised by the presence of somewhat larger proportions of non-metallic inclusions.

PIG IRON.

Typical Analyses of Pig Iron, etc.

The following list of analyses is taken from Dr. W. H. Hatfield's book, 'Cast Iron in the Light of Modern Research.' It is published with the kind permission of the author and of the publishers, C. Griffin & Co. Ltd.

West Coast Hematites.

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.	Cu.
No. 1 .	3·750	·300	2·60	·02	·045	·50	·04
„ 2 .	3·500	·460	2·40	·03	·045	·50	·04
„ 3 .	3·250	·540	2·10	·04	·045	·50	·04
„ 4 .	2·800	1·000	1·65	·10	·045	·50	·04
„ 5 .	2·400	1·600	1·20	·20	·045	·50	·04
Mottled	1·600	1·950	·90	·25	·045	·20	·04
White .	trace	3·250	·65	·30	·045	·10	·04
From Native Ores only.							
Special .	3·75	·300	2·60	·02	·02	0·600	—

East Coast Hematites. ('Seaton.')

Brand.	G.C.	C.C.	Si.	S.	P.	Mn.	Cu.
No. 1	3·725	·300	2·50	·02	·05	1·00	·05
" 2	3·525	·450	2·25	·03	·05	1·00	·05
" 3	3·150	·560	2·00	·04	·05	1·00	·05
" 4	2·750	1·000	1·50	·10	·05	1·00	·05
" 5	2·450	1·550	1·00	·20	·05	·75	·05
Mottled	1·500	2·050	·75	·25	·05	·50	·05
White	trace	3·150	·65	·30	·05	·50	·05
Special No. 1	3·525	·300	2·50	·015	·03	1·00	·05
" " 2	3·350	·400	2·25	·020	·03	1·00	·05
" " 3	3·00	·500	2·00	·025	·03	1·00	·05

Scotch Hematites. Welsh Hematites.

These are practically of the same composition as the East Coast Hematites.

No. 1 Foundry Pig-Iron.

Approximate Analyses (Makers') of Diverse Brands.

Brands.	Silicon.	Sulphur.	Phosphorus.	Manganese.	Com. Carbon.	Graphite.
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
Gartsherrie	3·50	0·020	0·70	1·35	0·25	3·20
Carron	2·80	0·035	0·70	1·00	0·14	3·50
Clarence	2·90	0·03	1·52	0·60	0·15	3·30
Cleveland	3·00	0·03	1·21	0·70	trace	3·50
Stanton	3·19	0·02	1·07	0·48	—	2·6
Staveley	3·20	0·02	1·40	0·80	0·15	3·45
Butlin	2·80	0·006	1·66	0·36	trace	3·79
Lilleshall H.B.	2·50	0·035	0·70	1·00	0·25	3·15
Fenton	2·70	0·02	1·34	2·16	0·77	2·80
Heath	2·75	0·03	1·13	2·75	0·32	3·05
Doncaster	2·25	0·03	1·2	2·2	0·5	3·4
N.L.B.	2·87	0·03	1·27	1·37	—	—
K.H.	3·00	0·05	0·90	1·20	0·20	3·0

No. 3 Foundry Pig-Iron.

Approximate Analyses (Makers') of Diverse Brands.

Brands.	Silicon.	Sulphur.	Phosphorus.	Manganese.	Com. Carbon.	Graphite.
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
Gartsherrie	2·89	0·03	0·70	1·42	0·35	2·82
Carron	2·95	0·04	0·70	1·00	0·20	3·35
Clarence	2·80	0·03	1·50	0·58	0·27	3·01
Cleveland	2·88	0·05	1·20	0·68	0·12	3·24
Stanton	3·29	0·06	1·17	0·5	0·32	2·9
Staveley	2·90	0·04	1·40	0·75	0·25	3·20
Butlin	1·78	0·02	1·60	0·40	0·13	3·22
Lilleshall H.B.	2·00	0·045	0·70	0·85	0·40	3·00
Fenton	2·52	0·04	1·35	1·65	0·65	2·95
Heath	2·25	0·04	1·12	2·25	0·40	2·56
Doncaster	2·0	0·03	1·2	1·9	0·5	3·4
N.L.B.	2·16	0·03	1·22	1·33	—	—
K.H.	2·50	0·06	0·90	1·0	0·4	2·8

'Economics of Iron and Steel,' by J. H. Skelton, 2nd ed.)

'Ferro' Alloys.

Brand.	C.	Si.	S.	P.	Mn.
Ferro-Manganese	6·64	·61	·028	·180	80·10
Ferro-Silicon	1·30	11·00	·040	·060	2·50
„ (Electro)	·27	28·52	·040	·040	·40
Silico-Spiegel	1·70	11·00	·040	·049	18·32
„ (Special)	·69	20·90	·018	·097	55·76
Spiegel	4·50	·90	·030	·070	18·00

NOTE.—Ferro-Manganese is usually 80% Mn. and 6–7% C.
Ferro-Silicon varies from 8% to 80% Si.
Silico-Spiegel varies from 10% to 14% Si and 18% to 20% Mn.
Spiegel varies from 10% to 25% Mn.

CAST IRON.

(By W. J. Driscoll, B.Sc. (Eng.), A.M.I.Mech.E., British Cast Iron Research Association.)

DEFINITION.

Cast irons may best be defined and classified by reference to their structure as seen under the microscope. Norbury's definition as slightly modified by Pearce defines cast irons as 'alloys of iron and carbon with or without other elements, which contain eutectic carbide (white cast iron) or eutectic graphite (grey cast iron), or both (mottled cast iron) in the microstructure.'

The U.S. Bureau of Standards gives the following definition: 'Cast iron is a cast alloy of iron and carbon, with or without other elements, in which the carbon content exceeds the maximum limit of solid solubility, as determined at any temperature (which in plain cast iron is 1·7 per cent.), and hence contains eutectic carbide or graphite as a structural feature. It is not usefully forgeable at any temperature.'

The following descriptions, while not complete definitions, may be more readily understood.

Cast irons are alloys of iron and carbon, with or without other elements, usually containing between 1·7 and 4·5 per cent. carbon. They are not usefully wrought, in their as-cast state, at any temperature.

Grey cast irons contain flake graphite and no free iron carbide.

White cast irons contain free iron carbide and no graphite.

Mottled cast irons contain both free iron carbide and flake graphite. Chilled cast iron, so called because one part of the casting is caused to cool much more quickly than the remainder, generally falls in this category.

Pig irons are usually the product of the blast furnace, and are cast irons which may be either grey, white or mottled.

Refined pig irons are pigs of cast iron, also grey, white or mottled, which have had their composition and structure modified either before solidification from the blast furnace or by a subsequent remelting process.

Malleable cast iron is the product obtained by the annealing of a white cast iron in an oxidising or neutral atmosphere so that part or the whole of the carbon previously in the combined state is deposited as graphite in the nodular or spheroidised state known as temper carbon, as distinct from the flake graphite present in grey and mottled cast irons.

CLASSIFICATION.

The method of classification adopted by B.S.S. 991 based on reports of the Institution of Mechanical Engineers, to which students may be referred, is to make a first division into three categories according to the type of graphite, if any, present:

Class 1	Grey cast iron	Flake graphite.
Class 2	Malleable cast iron	Temper carbon graphite.
Class 3	White cast iron	No graphite.

Subdivisions are then made according to the microstructure of the metallic matrix. The two most common constituents of the matrix of grey and malleable cast irons are (*a*) *ferrite* (pure iron in the case of pure iron-carbon alloys), and (*b*) the intimate mixture of lamellae of iron carbide and ferrite known as *pearlite*. The matrix of grey cast irons may also be modified by heat-treatment or alloying to produce other structures, of which the most important are *martensite* (corresponding to the hard constituent of tool steel) and *austenite* (corresponding to the corrosion-resisting matrix of the austenitic stainless steels). White cast irons, in addition to the free carbide, normally contain pearlite but, again, this can be changed wholly or in part to martensite by means of alloy additions.

Thus the three main categories may be subdivided as follows :

		Prime Constituent.	Other Constituents.
Class 1 Grey cast iron	Class 10	Flake graphite	Ferrite
	,, 10A	,, ,,	Ferrite and pearlite
	,, 11	,, ,,	Pearlite
	,, 12	,, ,,	Martensite
	,, 13	,, ,,	Austenite
Class 2 Malleable cast iron	Class 20	Temper carbon	Ferrite
	,, 20A	,, ,,	Ferrite and pearlite
	,, 21	,, ,,	Pearlite
Class 3 White cast iron	Class 30	Free carbide	Pearlite
	,, 31	,, ,,	Martensite

It will be noted that the two classes to which the suffix A has been added, and which are not specifically mentioned in B.S.S. 991, are really of an intermediate nature. There are, of course, irons with other intermediate structures, but these have not been given as their commercial use is insignificant compared with those given in the table.

One micro-constituent which has not been mentioned as it does not affect the classification scheme is *phosphide eutectic*. This is present in cast irons in amounts approximately proportional to the phosphorus content of the iron. It is present in the majority of grey cast irons, it may be present in white cast irons, but malleable cast irons do not contain phosphide owing to their low phosphorus content.

GRAPHITISATION.

The microstructure of cast iron in the as-cast state depends almost entirely on two factors : (*a*) the composition of the iron, and (*b*) the rate of cooling of the iron in the mould. This second factor is in turn dependent primarily on the thickness of the casting being produced. Thus, as a certain type of microstructure is aimed at in any one casting, cast iron is perhaps unique in that its chemical composition needs to be suited to the physical dimensions of the article being produced.

As already indicated, the carbon in unalloyed, un-heat-treated cast irons may exist either as flake graphite or as carbide, which in turn may be either lamellar (pearlite) or massive ('free' carbide). Ironfoundry practice thus consists, to a large extent, of suitably controlling the iron composition to give, in a given casting, the required relative amounts of graphite and carbide, *i.e.* controlling the degree of 'graphitisation,' or breakdown from carbide to graphite which takes place.

Pure iron-carbon cast irons, which are not commercially used, contain all their carbon in the combined state, *i.e.* as free carbide and pearlite. Silicon, which is present in all commercial irons, has, however, a graphitising effect and increasing amounts of this element therefore increase the amount of graphite and decrease the amount of combined carbon. The increase in graphite content takes place first at the expense of the free carbide and then of the pearlite. Other elements which are or may be present in cast iron also have either a graphitising or carbide stabilising effect, but this will be dealt with more fully in the section dealing with the elements themselves. The following diagram illustrates the mechanism of graphitisation, with reference to microstructure.

Increasing graphite content / Decreasing combined carbon content } Graphitisation and increasing softness. →

Free carbide and pearlite ⇄ Free carbide, pearlite and graphite ⇄ Pearlite and graphite ⇄ Pearlite, ferrite and graphite ⇄ Ferrite and graphite

← Decreasing graphite content / Increasing combined carbon content } Carbon stabilisation and increasing hardness.

Properties of Micro-Constituents.

Graphite.—Graphite present in cast iron has, as may be expected, very little strength in tension or shear, but is relatively strong in compression. Thus grey cast iron has a high compression/tension strength ratio compared with other metals and alloys. The size, shape and distribution of the graphite flakes can have a marked effect on the properties of an iron, and these factors can be controlled to a large extent by suitable control over composition, melting conditions or by treatment of the iron in the molten state.

Ferrite.—This is the only constituent of pure iron, represented commercially by electrolytic or wrought iron, and as such is of fairly low strength, soft and ductile. In normal cast irons, however, ferrite contains other elements, principally silicon and manganese, in solid solution, with the result that it becomes harder and stronger.

Pearlite.—This constituent, which under the microscope has the well-known 'thumb-print' appearance, consists of an intimate mixture of lamellae of ferrite and carbide and has properties intermediate between those of its two components, *i.e.* it is harder and stronger than ferrite and tougher than carbide. In pure iron-carbon alloys pearlite (the 'eutectoid') has a carbon content of 0·9 per cent., but the presence of other elements in cast iron may vary its carbon content within the range 0·7 to 1·1 per cent. Thus the combined carbon content of a cast iron containing no free carbide cannot exceed about 1·1 per cent., while that of an iron containing no ferrite cannot be less than about 0·7 per cent.

Cementite.—Cementite or carbide corresponds to the true chemical compound of iron and carbon Fe_3C, although the presence of other elements may result in the formation of a complex carbide. It can be present in cast iron as one of the constituents of pearlite and/or in its free or massive state. Cementite is hard and brittle.

Austenite.—This constituent cannot be obtained in unalloyed cast iron, and is only found in irons alloyed with elements which appreciably reduce the pearlite change point temperature, notably nickel alone or nickel and manganese combined. Austenite is a solid solution of iron carbide in iron. Its most important properties are that it is non-magnetic, it is stable at high temperatures and the presence of nickel renders it corrosion-resistant to a high degree. It is relatively strong and ductile.

Martensite.—Martensite may be regarded as an intermediate constituent between austenite and pearlite. It is not found in as-cast unalloyed irons but may be obtained by alloying and/or heat-treatment. It has a typical needle-like structure under the microscope. Martensite is very hard, relatively strong and less brittle than cementite.

Phosphide.—A small amount of phosphorus in cast iron may be dissolved or combined in other constituents but the greater part of it generally exists as a eutectic of iron, iron phosphide Fe_3P, and iron carbide Fe_3C, briefly known as phosphide eutectic. This eutectic has a relatively low melting-point and is therefore generally found at the grain boundaries of the previously solidified constituents. It contains only about 7 per cent. of phosphorus and the volume of eutectic present therefore increases rapidly with irons of increasing phosphorus content. Phosphide eutectic is brittle and relatively hard.

The following mechanical properties of micro-constituents have been quoted:

Constituent.	Tensile Strength tons/sq. in.	Elongation per cent. on 2 in.	Brinell Hardness. Number.
Pure ferrite [1]	17·0	51	70
Pure ferrite [2]	17·6–18·7	61	75
Ferrite containing 0·82 per cent. silicon [3]	20·2	50	88
Ferrite containing 2·0 per cent. silicon [1]	38·0	43	130
Ferrite containing 2·28 per cent. silicon [3]	28·4	50	124
Ferrite containing 3·4 per cent. silicon [3]	34·6	21	150
Pearlite [1]	53·6	15	225
Pearlite [4]	53·6	15	240
Cementite [4]	—	—	550 +
Cementite [5]	—	—	600 +

[1] Jeffries and Archer.
[2] Fuller.
[3] Yensen.
[4] American Society for Testing Materials, Symposium on Cast Iron.
[5] Sauveur.

CHEMICAL COMPOSITION.

The composition of the majority of cast irons lies within the following limits:

Total carbon	2·2–4·0 per cent.
Silicon	0·4–3·0 „ „
Manganese	0·05–1·5 „ „
Sulphur	0·03–0·35 „ „
Phosphorus	0·02–1·5 „ „
Iron	Balance.

The 'total' carbon content may be wholly 'combined" (as in white cast iron), wholly 'graphitic' (as in completely ferritic grey cast iron, which is, however, seldom encountered in the as-cast state with silicon contents of the order of those given above), or partly combined and partly graphitic (as in mottled cast iron or grey cast iron containing pearlite). As already mentioned, the combined carbon content of a grey cast iron cannot exceed about 1·1 per cent., while that of a mottled cast iron cannot be less than about 0·7 per cent.

In addition to the cast irons falling within the range given above there are the irons which are given special properties by the addition of other elements or by increasing the normal percentages of existing elements, principally silicon and manganese. Thus, what may be regarded as *alloy cast irons* may contain the following elements, singly or in combination, up to the amounts given:

Silicon	up to 17 per cent.
Manganese	„ „ 9 „ „
Nickel	„ „ 22 „ „
Copper	„ „ 7 „ „
Chromium	„ „ 35 „ „
Molybdenum	„ „ 1 „ „
Aluminium	„ „ 8 „ „

Other elements are also occasionally used as alloying elements in cast iron, but their use is not so common as that of those just mentioned.

THE EFFECT OF THE VARIOUS ELEMENTS.

Carbon.—The carbon in cast iron may exist as graphite or as combined carbon. The amounts of other elements present in the iron will decide the carbon content of the pearlite. A combined carbon content in excess of this amount will indicate the presence of free cementite. From the properties of graphite, pearlite and cementite which have been given, it follows that for an iron with a given total carbon content an increase in combined carbon content up to the amount contained in the pearlite will result in an increase in strength and hardness due (*a*) to the improved strength and hardness of pearlite compared with ferrite, and (*b*) to the reduced graphitic carbon content. An increase in combined carbon content over the amount in the pearlite (eutectoid) will result in a rapid increase in hardness due to the presence of free cementite, with consequent difficulty in machining. The ultimate product, is, of course, white iron in which all the carbon is in the combined form and there is no graphite. As normal white iron consists of pearlite and free cementite it follows that an increase in total carbon content will result in increased hardness due to the formation of an increased quantity of cementite. Thus high carbon white irons are often used where a high degree of hardness and abrasion-resistance is required.

In common with other alloys which are mutually soluble in the liquid state but not in the solid, iron-carbon alloys have a *eutectic* composition at which the temperature where solidification begins during freezing from the liquid is at its lowest. The eutectic iron-carbon alloy contains 4·3 per cent. of carbon, but the eutectic composition in most cast irons is modified by the presence of other elements, notably silicon and phosphorus, both of which reduce the carbon content of the eutectic. Irons containing less or more of carbon than the eutectic are known as hypo- or hyper-eutectic irons, respectively, the majority of commercial irons falling in the former class.

Silicon.—The principal effect of silicon in cast iron is that it is a graphitiser. That is, increasing silicon additions to a white cast iron would result in it becoming mottled, then grey and pearlitic, then passing through the pearlite-ferrite stage until at a relatively high silicon content (in excess of, say, 4 per cent.) the iron is completely ferritic. Thus, considering grey irons only, the effect of increasing silicon is to reduce the strength and hardness of the iron by reducing the pearlite content, and for maximum strength the silicon content is therefore adjusted to give the fully pearlitic (eutectoid) structure. Silicon in cast iron is dissolved in the ferrite portions, on which it has a hardening effect, but this effect on the ferrite is generally masked on the iron as a whole by the graphitising (hence softening) effect.

It has been found that in certain cases, if some of the silicon to be present is added to molten iron before pouring, the castings possess superior properties, particularly from the point of view of strength, to those which would be obtained with iron of the same final composition obtained straight from the furnace. This fact is the basis of many 'inoculation' processes.

Silicon contents higher than about 4 per cent. generally result in irons consisting only of graphite and ferrite (or, more correctly, silico-ferrite owing to the silicon in solution) and such irons are relatively hard and brittle, although still quite machinable.

Each 1 per cent. of silicon in cast iron reduces the carbon content of the eutectoid by about 0·07 per cent. and the carbon content of the eutectoid by about 0·3 per cent.

Silicon increases the temperature of the 'critical point,' *i.e.* the temperature above which the carbide in pearlite is re-dissolved to form austenite, and at which the iron undergoes a sudden volume change.

Both the thermal and electrical conductivities of grey iron are reduced by increasing silicon contents.

Manganese.—Part or all of the manganese present in cast iron exists in combination with the sulphur as inclusions of the chemical compound manganese sulphide, MnS. To form this compound 1·7 parts of manganese are required for each 1 part of sulphur, but it has been found that in order to ensure that no sulphur remains uncombined with manganese, an excess of manganese is necessary. Norbury gives this excess at 0·3 per cent. Most grey irons contain sulphur to the extent of about 0·1 per cent., and as the presence of excess sulphur in such irons is generally undesirable the manganese content is generally kept above 0·5 per cent. The manganese sulphide inclusions have very little effect on the properties of cast iron.

Manganese present in excess of the amount combined with the sulphur has a carbide-stabilising effect, *i.e.* it tends to increase the percentage of pearlite in a grey cast iron and so increase the strength and hardness. It also both reduces the carbon content of the eutectoid and the temperature at which the eutectoid is formed, with the result that when about 2 per cent. of manganese is reached martensite tends to be formed instead of pearlite, while at manganese contents of the order of 10 per cent. fully austenitic structures are obtained. High manganese irons containing martensite or austenite are virtually unmachinable.

Sulphur.—In most cast irons all the sulphur exists as inclusions of manganese sulphide, to which reference has already been made, and therefore has little effect on the properties of the iron. Sulphur present in excess of the amount which can combine with the manganese acts, however, as a powerful carbide stabiliser, *i.e.* high sulphur and low manganese irons tend to solidify in the mottled or white condition. White irons for the production of malleable cast iron, particularly of the 'whiteheart' type, frequently have high sulphur and low manganese contents.

Phosphorus.—Almost all the phosphorus present in cast iron exists as phosphide eutectic which, having the relatively low freezing point of about 960° C., exists at the grain boundaries of the other constituents. Although phosphide eutectic is brittle it has no very marked effect on the static strength of cast iron in amounts up to about 0·7 per cent. phosphorus, but higher phosphorus contents tend to reduce strength by the formation of continuous phosphide networks. Impact strength progressively falls and hardness increases with increasing phosphorus content, the increase in hardness having, however, a relatively slight effect on machinability, except in the case of some of the stronger irons where machinability is impaired somewhat.

It has been indicated that in many cases the presence of phosphide eutectic has a beneficial effect on wear resistance.

Phosphorus reduces the freezing point of the iron-carbon eutectic, and this fact, combined with the low freezing point of the phosphide eutectic results in molten high phosphorus irons being very fluid for a given temperature. For this reason such irons are often used where intricate castings of thin section need to be produced. The appreciable temperature range over which high phosphorus irons solidify sometimes tends to produce shrinkage defects in castings, although such defects are often due to unsuitable contents of other elements.

Graphitisation is not noticeably affected by phosphorus but the carbon content of the eutectic is reduced by 0·3 per cent. for each 1 per cent. of phosphorus.

L. W. Bolton showed that phosphorus increases the rigidity of cast irons at temperatures below the melting point of the phosphide eutectic.

The physical properties of cast iron, as distinct from the mechanical properties, are but slightly affected by phosphorus.

Nickel.—The action of nickel in cast iron is, perhaps, rather complicated. It acts as a graphitiser in respect of irons containing free carbide, but its graphitising effect on grey irons is less noticeable. Thus, considering an originally pearlitic iron, while the hardening effect of increasing silicon is masked by the softening effect of the graphitisation of pearlite, the reverse is true of nickel, *i.e.* the nickel in solution hardens and strengthens the iron more rapidly than the increasing graphite content can soften and weaken it. Thus castings containing sections of varying thickness tend to be free from mottle in the lightest sections and to have a more uniform hardness between the thinnest and thickest sections.

Nickel, similarly to manganese, reduces the eutectoid temperature, and when a nickel content of about 3 per cent. is reached martensite begins to appear in the microstructure until at about 5 per cent. nickel the iron is completely martensitic. Further increase in nickel content results in the formation of austenite with the martensite until at about 18 per cent. nickel the structure

is fully austenitic. The actual nickel contents at which the above changes take place will, of course, vary with the cooling rates (section thicknesses) and the amounts of other elements present.

Pearlitic nickel cast iron is, of course, readily machinable but machinability rapidly falls off as increasing nickel content causes increasing amounts of martensite, until fully martensitic grey iron is only machinable with great difficulty. Machinability improves again as the iron becomes austenitic until fully austenitic grey iron is quite readily machinable. Martensitic white iron may only be machined with carbide tools at low speeds.

Austenite is non-magnetic and austenitic nickel cast irons are often used when this property is necessary.

Copper.—Copper as an addition to cast iron differs from most others in that only a limited amount is soluble in the iron. This limit may be taken as about 3·5 per cent. when no other alloying elements are present. Above this limit free copper begins to appear in the microstructure as small microscopic particles (secondary copper), while above about 5 per cent. the limit of liquid solubility is exceeded and visible globules or layers of free copper (primary copper) appear in the castings. Irons containing primary or secondary copper are at present little used.

The effect of copper on cast iron in amounts up to about 3 per cent. is similar to that of nickel, *i.e.* the graphitisation of cementite is favoured and the pearlite is hardened and strengthened, the hardness between thin and thick sections thus tending to be equalised.

Copper is unlike nickel in that the temperature of the pearlite point is not reduced and martensite or austenite are therefore not produced at higher copper contents.

Combined additions of nickel and copper to cast iron increase the solid solubility of the copper itself above the normal figure of 3·5 per cent. Each 1 per ecnt. of nickel increases the solubility of copper in cast iron by 0·5 per cent.

Chromium.—Chromium is a carbide stabiliser, being rather more powerful in this respect than silicon is as a graphitiser. Thus chromium additions to cast iron result in rapidly increasing hardness and combined carbon content, other factors being unchanged, while additions of equal amounts of chromium and silicon would result in less rapid carbide stabilisation. Chromium in cast iron is present in the cementite as a complex carbide of iron and chromium, this carbide being harder and more stable at high temperatures than the normal iron-carbon cementite. An increased degree of corrosion-resistance is also imparted by the presence of chromium. Thus chromium is frequently used in castings which need to have an improved degree of hardness, wear-resistance, heat-resistance or corrosion-resistance.

Amounts of chromium in excess of 1 per cent. are seldom used in normal pearlitic irons on account of its powerful carbide-forming action, but about 1·5 per cent. may be present in martensitic white irons to give a high hardness value, while 2 per cent. or more may be added to austenitic grey irons to increase hardness or supplement corrosion-resistance. Irons of much higher chromium content (30 to 35 per cent.) have interesting properties in that they are very resistant to heat, corrosion and erosion. The microstructure of such irons consists entirely of carbides in a ferritic matrix, their hardness (250–350 B.H.N.) varying directly as the carbon content. These irons are not brittle like most white irons and, although relatively hard, can be machined fairly readily.

Molybdenum.—Molybdenum is an element which has a very marked effect on the properties of cast iron even when present in relatively small quantities. It has a mildly carbide-stabilising effect and markedly improves the hardness and strength of grey cast irons when present in amounts up to about 1 per cent., the improvement being less rapid above this content. Molybdenum-bearing irons have been found to possess greatly improved impact and fatigue characteristics, particularly when nickel is present also, and the strength of such irons can often be increased even more by a low temperature heat-treatment process. Grey irons containing molybdenum are normally quite machinable, but difficulty may be experienced with irons of very high strength.

Aluminium.—An addition of aluminium of the order of 0·1 per cent. may be made to molten cast iron to act as a deoxidiser, but such a small amount may not really be regarded as an alloying addition. When present in larger amounts, aluminium has a graphitising influence equal to about one-half that of silicon. This holds true up to about 4 per cent. of aluminium, above which it begins to have a carbide stabilising effect until at about 8 per cent. completely white iron is formed. Aluminium tends to reduce the strength and toughness of grey iron, firstly owing to the difficulty of ensuring that inclusions of alumina are not present in serious proportions, and secondly, owing to the tendency for coarse graphite flakes to be formed.

The presence of aluminium in cast iron greatly improves its resistance to scaling and growth at high temperatures. The electrical conductivity of cast iron is appreciably reduced by the addition of aluminium.

Aluminium forms a very hard compound with nitrogen, and it is therefore included, generally together with chromium, in cast irons which are to be hardened by nitriding. An iron containing, say, 1·5 per cent. aluminium and 1·7 per cent. chromium may be nitrided to give a hard case with a diamond-pyramid hardness number of about 1,000.

The following table summarises some of the principal effects of various elements on cast iron.

Element.	Carbon Content of Eutectoid (Pearlite).	Temperature of Critical Point (Pearlite Point).	Carbon Content of Eutectic.	Approximate Graphitising Value (Silicon Basis).
Silicon	Decreased	Increased	Decreased	1·0
Manganese	Decreased	Decreased	—	−0·25
Phosphorus	—	—	Decreased	—
Nickel	—	Decreased	—	0·35
Copper	—	—	—	0·35
Chromium	-	Increased	—	- 1·2
Aluminium	—	—	—	0·5

The negative signs in the last column of the table indicate that elements concerned have a carbide stabilising effect and not a graphitising effect. The numerical values themselves are an attempt to give quantitatively the graphitising or carbide stabilising effects of the various elements but the figures should be used with caution as the exact effect of an element will often vary with the amount of it to be present. as has already been mentioned, and with the amounts of other elements present.

A blank (—) in the table does not necessarily mean that the element has no appreciable effect on the given property—it may mean that the effect, if any, has little practical significance.

Malleable Cast Iron.

There are two main types of malleable cast iron known as *blackheart* and *whiteheart*, the terms having reference to the appearance of the fractures of the irons. In the as-cast state all irons for the production of malleable cast iron are in the white form, *i.e.* all the carbon present is combined, and the malleable properties are obtained by annealing the castings at a high temperature for a suitable period of time.

Whiteheart castings are made from an iron which has been rendered white by the use of a low silicon content and, sometimes, a relatively high sulphur content. The castings are then packed in iron ore, heated to a temperature of 950°–1,000° C. for 60–120 hours and then slowly cooled in the furnace. While the castings are at the high temperature some of the carbon they contain diffuses outwards from the centres of the castings and is oxidised by the ferric oxide of the ore, which itself is reduced to magnetic oxide. Some of the carbon in the castings also becomes deposited throughout the castings as graphite of the nodular or spherical form known as temper carbon. On the outside surface of whiteheart castings there is usually a layer from which all the carbon has been removed and which therefore consists only of ferrite. The remainder of the castings normally contains temper carbon, pearlite and ferrite, although very thin castings tend to become more completely ferritic. The total carbon content of annealed whiteheart castings is, of course, lower than that of the original iron.

Blackheart castings are made from white irons with lower total carbon contents and rather higher silicon contents than for whiteheart. The castings are then packed in an inert material such as sand, heated to 850°–900° C. for 60–100 hours and then slowly cooled. Blackheart castings are rendered malleable almost entirely by the deposition of graphite in the temper carbon form, little carbon being lost by decarburisation; hence the need for starting with an iron of low carbon content. The structure of blackheart malleable, consisting of temper carbon and ferrite, is substantially uniform throughout the casting, although there may be a slight tendency for a carbon-free skin to be present at the outside surface.

The chemical compositions of white cast irons for the production of malleable iron generally lie within the following ranges:

	Whiteheart. Per cent.	*Blackheart.* Per cent.
Total carbon	3·0 —3·7	2·2 —2·8
Silicon	0·4 —0·9	0·7 —1·1
Manganese	0·05—0·4	0·25—0·35
Sulphur	0·10—0·35	0·03—0·09
Phosphorus	below 0·10	below 0·15

The properties of malleable cast irons will be dealt with in subsequent sections.

The Properties of Cast Irons.

Each of the following sections will, as far as possible, include information on the property under consideration on the whole range of cast irons, but where, for instance, a particular type of cast iron is important only from the point of view of one special property, *e.g.* heat-resistance, a note of its other properties may be included in that particular section.

Tensile Strength.

Normal Grey Cast Irons.—It has previously been pointed out that the structure of cast iron in the as-cast state is primarily dependent, firstly, on the composition of the iron and, secondly, on the rate of cooling of the iron in the mould. The second factor renders it necessary that any bar on which a mechanical or physical test is to be carried out should have a cooling rate similar to that of the iron in the casting concerned. British Standard Specifications Nos. 321—1938 (General grey iron castings) and 786—1938 (High duty iron castings) therefore include test bars of five standard diameters, as follows:—

Nominal Diameter of Test Bar.	Limits on Diameter.	Main Cross-sectional Thickness of Casting Represented.
0·6 in.	± 0·045 in.	Not exceeding ⅜ in.
0·875 „	± 0·065 „	Over ⅜ in. and not exceeding ¾ in.
1·2 „	± 0·090 „	Over ¾ in. and not exceeding 1⅛ in.
1·6 „	± 0·10 „	Over 1⅛ in. and not exceeding 1⅝ in.
2·1 „	± 0·10 „	Over 1⅝ in.

The minimum tensile strengths of the six grades of iron covered by the two specifications are given in the following table.

Nominal Diameter of Test Bar—In.		0·6	0·875	1·2	1·6	2·1
B.S.S.	Grade	Ultimate Tensile Stress—Tons per Sq. In.				
321	C	10	10	9	9	9
„	A	12·5	12	11	10·5	10
786	1	16	15	14	13	12·5
„	2	19	18	17	16	15
„	3	23	22	20	19	18
„	4	26	25	23	22	21

The strength of grey cast iron depends on the strength of the metallic matrix and on the quantity and form of the free graphite. Thus an iron becomes stronger as the amount of ferrite decreases (the amount of pearlite increases), the pearlite is refined and strengthened, the amount of graphite decreases (hence the total carbon content decreases for a given combined carbon content) and as the arrangement of the graphite flakes improves.

The actual ways in which irons of improved strength may be produced can therefore be considered:

1. The composition of the iron, particularly with respect to silicon, may be so adjusted and controlled that a minimum amount of ferrite is present in the castings.

2. The phosphorus content of the iron may be reduced below the amount (approximately 0·7 per cent.) at which continuous phosphide networks tend to be present.

3. Suitable amounts of steel may be melted together with the pig iron, etc. The silicon content of the resulting iron is thereby reduced, this tending to increase the pearlite content, while the reduction in total carbon content which also takes place causes the amount of graphite to be reduced. This is the manner in which so-called *semi-steel* is produced, but the use of this term is strongly to be deprecated as the product is essentially a true cast iron. *Emmel* iron is a low carbon cast iron produced by incorporating steel in the charge, although the silicon content is normally kept at a relatively high figure. A typical Emmel iron may have the following composition: total carbon 2·7 per cent., silicon 2·4 per cent., manganese 1 per cent., phosphorus 0·1 per cent. This type of iron is now seldom used.

4. An iron which would normally solidify white because of a low silicon content may be cast into a heated mould. The reduced rate of cooling of the iron in the mould results in its solidifying grey and pearlitic. This is the process by which *Lanz* or *Perlit* iron is produced. The composition of this type of iron may be as follows: total carbon 3 per cent., silicon 0·9 per cent. manganese 0·7 per cent., phosphorus 0·3 per cent.

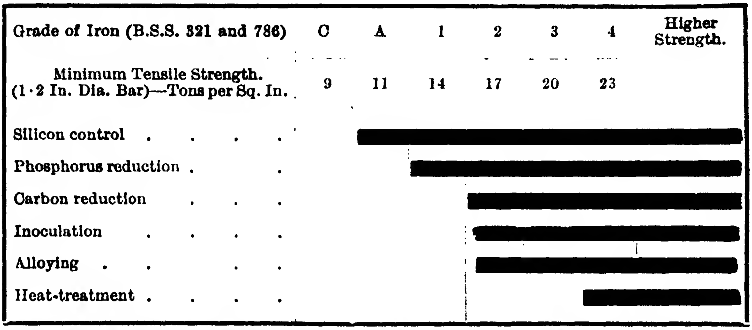
5. An iron which would normally solidify white may be 'inoculated' while in the molten condition with a silicon-containing material. This causes the iron to solidify grey and pearlitic, a favourable effect on the graphite structure being obtained also. The inoculation by calcium silicide was the original basis for the production of *Meehanite* cast iron, while *Ni-Tensyl* is a cast iron alloyed with nickel and inoculated with a silicon-bearing material. A typical analysis of

Ni-Tenysl iron is as follows: total carbon 2·9 per cent., silicon 1·5 per cent., manganese 0·8 per cent., sulphur 0·1 per cent., phosphorus 0·3 per cent., nickel 1·5 per cent. Many manufacturers now make inoculated irons, with or without brand names, and a number of inoculants are used in practice.

6. Alloying elements such as nickel, copper and molybdenum, which strengthen the matrix of the iron, may be used. Additions of nickel or copper, which have a graphitising action, are generally accompanied by a reduction in the silicon content or by the use of a chromium addition to compensate for any graphitising effect. Molybdenum, which is a carbide stabiliser, is frequently used in conjunction with nickel or copper.

7. Heat-treatment such as quenching and tempering may improve the strength of cast irons considerably, particularly some alloyed irons.

The following diagram gives some indication of the means by which irons of increasing strength are obtained commercially.



The above table is intended as a guide only and should therefore be used with discretion. For instance, it is not suggested that a Grade I cast iron cannot be made without a low phosphorus content, that both inoculation and alloying are always necessary for the production of a Grade 2 iron, or that Grade 3 irons are not sometimes produced by heat-treatment.

It should be pointed out that while British Standard Specifications at present only cover irons with a tensile strength of up to 26 tons per sq. in., strengths in excess of this—up to about 40 tons per sq. in.—are being produced regularly in commercial practice.

Martensitic and Austenitic Grey Cast Irons.—The tensile strength of martensitic grey cast irons may vary between 15 and 30 tons per sq. in. according to composition and heat-treatment, if any.

Austenitic grey irons such as Nomag, Nicrosilal and Ni-Resist have tensile strengths between about 10 and 20 tons per sq. in. Such irons are remarkable by virtue of their appreciable ductility and toughness compared with normal grey irons. While the elongation of the latter in the tensile test is negligible, austenitic irons have elongations of between 1 and 4 per cent.

White Cast Irons.—These irons are seldom used in applications where tensile strength is of primary importance and relatively few tests have been carried out because of the difficulty in machining the test-pieces and of carrying out the tests accurately. The results available show, however, that tensile strength increases as total carbon content decreases, the range being approximately 10 to 25 tons per sq. in. Alloyed martensitic white irons (*e.g.* Ni-Hard) are up to 50 per cent. stronger than unalloyed irons.

Malleable Cast Irons.—British Standard Specifications Nos. 309—1927 (Whiteheart) and 310—1927 (Blackheart) require the following tensile test results on unmachined, cast-to-shape malleable iron test bars:

	Whiteheart.	*Blackheart.*
Minimum ultimate tensile stress, tons per sq. in.	20	20
Minimum elongation, per cent. on 2 in.	5	7½

Malleable cast irons in commercial production normally have superior properties to the above, the approximate ranges being as follows:

	Whiteheart.	*Blackheart.*
Ultimate tensile stress, tons per sq. in.	22–28	20–25
Elongation, per cent. on 2 in.	5–18	12–18

Transverse Strength.

Normal Grey Cast Irons.—The transverse test, in which a bar supported at the ends is centrally loaded until fracture takes place, is commonly used for grey cast irons in view of its economy and simplicity. The two readings which are taken are the maximum load sustained by the bar before fracture and the total deflection of the bar at the loading point when fracture takes place. The deflection of the bar may be considered as some indication of the toughness of the iron although, more recently, it is considered that the value has little mechanical or metallurgical significance but serves as a valuable guide to the founder as to regularity and uniformity of product.

From the transverse breaking load is calculated the *transverse rupture stress* by means of the beam formula $f = Wl/4Z$ where :

f = transverse rupture stress in tons per sq. in.
W = breaking load in tons.
l = distance between supports, in ins.
Z = modulus of section, in in.^{3}

For a beam of circular cross-section $Z = \pi d^3/32$, hence transverse rupture stress equals $8Wl/\pi d^3$ where d is the diameter of the test bar in in.

The following table gives the minimum values for transverse rupture stress and deflection required in irons to British Standard Specifications Nos. 321 and 786.

Nominal Diameter of Test Bar—In.		0·6	0·875	1·2	1·6	2·1
Distance between Supports—In.		9	12	18	18	24
B.S.S.	Grade.	Transverse Rupture Stress—Tons per Sq. In.				
321	C	19·9	19·6	18·9	18·3	17·7
,,	A	25·1	24·1	23·1	21·4	19·6
786	1	27	25·9	25	24	23·6
,,	2	30	28·9	28	27	26·1
,,	3	34	33	31	30	29·1
,,	4	41	39	37	35	33
B.S.S.	Grade.	Deflection in Ins.				
321	C	0·06	0·09	0·13	0·10	0·14
,,	A	0·07	0·10	0·15	0·12	0·15
786	1	0·07	0·11	0·16	0·13	0·17
,,	2	0·08	0·12	0·17	0·14	0·19
,,	3	0·09	0·13	0·18	0·16	0·21
,,	4	0·10	0·14	0·23	0·18	0·24

The beam formula which is used for transverse test calculations actually only holds true for a material behaving elastically. Almost all metals and alloys, however, do not deform elastically up to their breaking point, and this is true of grey cast iron, with the result that the 'transverse rupture stress' is greater than the true maximum tensile stress present in the test bar, which must approximate to the ultimate tensile stress. The ratio of transverse rupture stress to ultimate tensile stress varies in practice between about 1·6 : 1 and 2 : 1 for cylindrical test bars, the value of the ratio tending to decrease as the strength of the iron increases. J. G. Pearce, following an analysis of a large number of transverse and tensile test results, gives the formula $R = 1 \cdot 1T + 9 \cdot 5$ as applicable over the range of irons from 8 to 20 tons per sq. in. tensile strength, where R = transverse rupture stress and T = ultimate tensile stress.

Martensitic and Austenitic Grey Cast Irons.—Martensitic irons have a similar transverse/tensile relationship to ordinary irons. Austenitic irons have a transverse rupture stress of between 20 and 35 tons per sq. in., the transverse/tensile ratio thus being rather higher than for pearlitic irons. Their ductility is also strikingly demonstrated by the high total deflections obtained in the transverse test. Thus a 1·2 in. diameter bar tested on 18 in. centres will have a deflection at breaking load of about 1 in. compared with the maximum of 0·23 in. given in the B.S. Specification for high duty grey cast iron.

White Cast Irons.—These irons behave relatively elastically and their transverse rupture stress approaches more nearly to their tensile strength than in the case of the grey irons. Deflections at fracture are also lower than in comparable grey iron bars as the plastic portion of the total deflection is extremely small.

Malleable Cast Irons.—The normal transverse test is obviously unsuitable for malleable cast iron which is therefore subjected, according to B.S. Specifications Nos. 309 and 310, to a cold bending test. The test bars have a 1 in. by ½ in. rectangular cross-section and are bent round a former with a 1 in. radius. The specifications call for angles of bend, without signs of cracks or flaws, as follows:

Whiteheart	45°
Blackheart	90°

Many commercial malleable irons, both whiteheart and blackheart, are now normally produced with angles of bend of the order of 180°.

Compression Strength.

The compression strength of grey cast iron is generally between three and four times its tensile strength, the actual ratio decreasing as the strength of the iron increases. Thus an iron of 12 tons per sq. in. tensile strength will have a compression strength of 50 tons per sq. in., while a 30-ton tensile iron will have a compression strength of about 90 tons per sq. in.

Malleable cast iron, when tested in compression, has an ultimate strength similar to its tensile strength, although exact results are difficult to obtain owing to the appreciable amount of plastic deformation which takes place.

Shear Strength.

Direct Shear.—A number of investigators have compared shear and tensile strengths for a range of grey cast irons, and it is apparent that the shear/tensile ratio decreases as the strength of the iron increases. A mean of the results which have been obtained indicates that the shear/tensile ratio is of the order of 1·5 for irons of tensile strength 10 tons per sq. in., 1·3 at 20 tons per sq. in. and approaches unity at about 25 tons per sq. in. and above.

Torsional Shear.—Few results are at present available on the strength of cast iron in torsion. The ultimate torsional stress will, of course, vary with the shape of the cross-section of the test bar, as in the case of the transverse test, it being probable that the torsional/tensile strength ratio will also decrease with irons of increasing tensile strength. Draffin and Collins obtained, for a 13-ton tensile iron, a torsional/tensile strength ratio of 1·3 for a solid cylindrical test bar, and of unity for a hollow thin-walled, cylindrical bar.

Elastic Moduli.

Grey Cast Irons.—The elastic moduli of grey cast iron increase with irons of increasing strength and decreasing graphitic carbon content. Thus *Young's Modulus* for an iron of tensile strength 10 tons per sq. in. will be about 12×10^6 lb. per sq. in., while for a 22-ton iron it will be about 21×10^6 lb. per sq. in.

Few estimations of the *Modulus of Rigidity* of grey cast iron have been made but, assuming a value for *Poisson's Ratio* of 0·25, the modulus of rigidity is 40 per cent. of the value for Young's Modulus. This relationship is mostly confirmed by the results available.

White Cast Irons.—Young's Modulus for white cast irons is in the region of 30×10^6 lb. per sq. in.

Malleable Cast Irons.—Blackheart malleable cast iron has a Young's Modulus of about 25×10^6 lb. per sq. in., while that for whiteheart tends to be somewhat higher, although the exact figure will depend on the section size (hence degree of decarburisation) of the casting concerned.

Impact Strength.

Normal Grey Cast Irons.—The normal single-blow impact tests, such as the Izod and Charpy, on standard notched test bars, are unsuitable for most grey cast irons owing to the low values which are obtained. In order to show the very appreciable difference between the impact strength of various types of grey cast iron, therefore, a modified Izod test is used. It is carried out in a standard 120 ft.-lb. capacity Izod machine, the test-piece, however, being a cylindrical un-notched bar of 0·798 in. diameter. The striking height—distance from the top face of the anvil to the moving knife-edge—is 22 mm. Using this test impact strengths vary from about 5 ft.-lbs. for ordinary high phosphorus irons to about 30 ft.-lbs. for high duty alloyed irons. In general, impact strength increases as tensile strength increases, and is reduced by increasing phosphorus content. High impact strength is common to most molybdenum-bearing high duty irons.

Austenitic Grey Cast Irons.—The relatively high ductility of these irons is reflected in their impact strength, which is normally in excess of the capacity of the 120 ft.-lb. machine in the above test.

Malleable Cast Irons.—The Izod impact value for blackheart malleable iron is of the order of 15 ft.-lbs. The value for whiteheart will vary with the amount of decarburisation which has taken place, *i.e.* thin sections may give rather higher and thick sections rather lower results than the above figure. Impact tests on whiteheart malleable irons are complicated by the fact that the ductile decarburised skin may be pierced by the notch, and in such cases will give low impact values. This must be borne in mind where malleable castings have to be machined or threaded.

Fatigue Strength and Notch Sensitivity.

Grey Cast Irons.—The endurance limit for grey cast irons is generally 0·45 to 0·5 times the ultimate tensile stress, the higher values being applicable to most high strength irons. The fatigue strength is little affected by temperatures up to about 400° C. Of importance to designers is the fact that the sensitivity of grey irons to any notch effects and stress-raisers is negligible.

Malleable Cast Irons.—The endurance limit for malleable irons varies from 0·43 to 0·54 of the tensile strength. The notch sensitivity is greater than that of grey iron but less than that of steel.

Vibration Damping Capacity.

This property is normally measured by stressing a machined bar in torsion to a pre-determined stress, making an autographic record of the vibration of the bar after the stress is suddenly removed, and calculating from this record the percentage loss of energy per cycle within the material. Hatfield and others give results which show that normal engineering cast irons have a damping capacity generally within the range 8–15 per cent. loss of energy per cycle at an initial stress of 2 tons per sq. in. compared with values of the order of 1 per cent. for most steels. High duty cast irons of about 20 tons per sq. in. tensile strength gave about 5 per cent. loss of energy per cycle.

In addition to the value of grey cast iron in the structural members of high speed machinery, it is, because of its high damping capacity, being increasingly used for tool-shanks, milling cutter bodies, crankshafts, etc.

White cast iron, because of its highly elastic nature, may be expected to have a low damping capacity, while that of malleable cast iron will probably be intermediate between those of grey iron and steel.

Hardness and Machinability.

It is convenient to consider the two properties of hardness and machinability together because they are, of course, closely related, although that is not to say that they are simply related. The difficulty in relating hardness and machinability lies partly in the difficulty of giving a numerical value to the latter property owing to the number of factors which need to be considered, *e.g.* tool material, cutting speed, feed, tool life between sharpenings, power consumption, etc.

The hardness and machinability of a cast iron depend, of course, on the quantities, arrangement and composition of its micro-constituents, the properties of which, individually, have already been described. The properties of the various types of cast irons may, however, be summarised as follows:—

Normal Grey Cast Irons.—The Brinell hardness of unalloyed irons containing ferrite and pearlite or pearlite only generally lies between about 130 and 230, the hardness increasing as the pearlite content increases. Phosphorus, when present in amounts up to about 0·5 per cent., does not materially affect the hardness, but higher percentages do increase the hardness on account of the rapidly increasing amount of the hard phosphide eutectic present, this also tending to have an adverse effect on tool wear. Unalloyed grey irons up to about 230 Brinell are readily machinable but a hardness figure in excess of this usually means that a small amount of 'mottle' or free carbide is present with the consequent deleterious effect on machinability.

The maximum ease of machining of cast iron is obtainable by annealing the castings at 800°–900° C. and cooling slowly through the critical temperature. The pearlite then breaks down into graphite and ferrite, the matrix, on cooling, thus being entirely ferritic. The Brinell hardness is then of the order of 120 but the strength of the iron is, of course, lower than when it contained pearlite.

With alloyed, pearlitic, grey irons a Brinell hardness in excess of about 230 does not necessarily indicate the presence of free carbide owing to the hardening and strengthening effect of the alloys on the pearlite itself. Thus high strength grey irons may have a Brinell hardness in excess of 300 and still remain readily machinable although, of course, the cutting loads may be somewhat higher than with unalloyed irons.

Martensitic and Austenitic Grey Cast Irons.—Martensitic grey irons may have a Brinell hardness between 380 and 500, and are machinable only with great difficulty.

Austenite is ductile and relatively soft, the Brinell hardness of austenitic irons generally being of the order of 170. Such irons are, however, susceptible to work-hardening, and slower cutting speeds than with normal grey irons are therefore generally necessary. Deep cuts are preferable to light cuts. Austenitic irons are generally fine-grained and after machining take a polish which, in the case of the nickel-austenitic irons, is highly corrosion-resistant.

White Cast Irons.—Unalloyed white irons consist solely of carbide and pearlite. Increasing the carbon content therefore increases the amount of carbide present and hence increases the hardness. Any addition of chromium to a white iron also slightly increases the hardness of the carbide itself. The Brinell hardness of an unalloyed white iron varies from about 400 for a low carbon iron to about 500 for a high carbon iron.

Martensitic white irons, which consist entirely of two hard constituents—carbide and martensite—have Brinell hardnesses varying between about 575 and 700, the hardness again increasing with the carbon content.

All white cast irons, including the martensitic type, may be machined with carbide tools at low speeds, although grinding is frequently used for both roughing and finishing.

Malleable Cast Irons.—The Brinell hardness of blackheart malleable iron castings generally lies within the range 110 to 120. Whiteheart castings vary in hardness between the skin and the core, the ferritic skin having a Brinell hardness of 120 to 140 and the ferrite-pearlite core having a hardness of 140 to 180 in castings of normal thickness. In thin castings a hardness of 140 will probably not be exceeded throughout.

Blackheart castings, consisting entirely of ferrite and graphite, have extremely good machining properties. The graphite-free, ferritic skin of whiteheart castings is soft and easily cut, but as the lubricating effect of graphite is absent an external coolant is advisable in most cases, as for steel. The core of whiteheart castings, containing temper carbon, is freely machinable.

Wear-Resistance.

Wear-resistance is, of course, difficult of definition and evaluation because results always need to be related to service conditions, *e.g.* the material by which the iron under consideration is being worn, degree of external lubrication, if any, extent to which wear is caused by corrosion, etc. For this reason the following remarks should be taken as general indications only and not necessarily applicable to all conditions.

Normal Grey Cast Irons.—Practical experience has shown that, in general, grey cast iron has good wear-resisting properties, possibly due to the fact that either the graphite at the surface of the iron acts as a lubricant or the graphite cavities act as pockets for applied lubricants.

The wear-resistance of grey cast iron normally increases as the pearlite content is increased, fully pearlitic irons giving the best results. Phosphide eutectic has a beneficial effect on wear as long as it is firmly embedded in the rest of the matrix. If particles of phosphide eutectic tend to become detached in service they will, of course, cause an increase in wear. Alloy additions, such as chromium, nickel, copper, molybdenum, which harden or strengthen pearlite, improve wear-resistance in numbers of applications.

Surface treatments such as flame-hardening, cyanide hardening, nitriding and chromium plating may be applied to cast iron surfaces to give excellent wear-resisting properties. A typical composition of an iron which would be particularly susceptible to nitrogen hardening is: total carbon 2·6 per cent., silicon 2·6 per cent., manganese 0·6 per cent., phosphorus 0·1 per cent., aluminium 1·5 per cent., chromium 1·7 per cent. The hardness of the nitrided case on this iron would be of the order of 1,000 diamond pyramid number.

Martensitic Grey Cast Irons.—The wear-resistance of these hard irons is very high and they are used where a strong iron is required under severe erosive or abrasive conditions, and where the difficulty of machining the irons is not of primary importance. A typical composition is as follows: total carbon 3·3 per cent., silicon 1·2 per cent., manganese 0·8 per cent., nickel 5 per cent., chromium 0·75 per cent.

Austenitic Grey Cast Irons.—These irons are corrosion-resistant, and in applications where corrosion is contributory to wear, *e.g.* cylinder liners for internal combustion engines, austenitic irons have given improved life.

White and Chilled Cast Irons.—Chilled cast irons are so named because the mould for the casting is so arranged that the part of the casting which is required to solidify white and hard is caused—generally by means of a metal insert—to cool much more quickly than the remainder of the casting which solidifies grey because of the lower cooling rate. Examples of this type of casting are chilled rolls and plough points, in which the hard, wear-resistant working face is supported by strong, grey iron at the back of it.

White cast iron, by virtue of its high hardness, is very resistant to abrasive wear, while martensitic white iron is even more resistant. A typical analysis of *Ni-Hard*—a martensitic white iron—is as follows: total carbon 3 per cent., silicon 0·7 per cent., manganese 0·8 per cent., nickel 4·5 per cent., chromium 1·5 per cent. This iron would have a Brinell hardness of about 650.

Malleable Cast Iron.—Malleable castings are not at present often used in applications where wear-resistance is of primary importance. Provided that the carbon-free skin is removed by machining, whiteheart castings can, however, be hardened by heating to about 800° C. and quenching. A wear-resistant structure results, and castings so treated have given very satisfactory results in certain applications.

Heat-Resistance.

Iron castings may fail in high temperature service due to one or more of various causes, *e.g.* (*a*) fracture or distortion due to stresses, which may be externally applied or internal, which the material cannot withstand at the temperature at which failure takes place, (*b*) 'growth' which shows itself as a large permanent expansion, usually accompanied by distortion, due to decomposition of carbide into graphite and to the formation of oxides around the graphite flakes, (*c*) surface oxidation or scaling, and (*d*) corrosion by the materials in contact with the castings. These factors have led to the use of certain classes of iron for high temperature applications.

White Cast Irons.—Such irons contain, of course, no graphite, and provided that the composition is such that the combined carbon is stable at the temperature concerned no growth can therefore take place. In spite of this the applications of these irons are limited, firstly because of their relative brittleness, and secondly, because of their low resistance to scaling even under mildly oxidising conditions.

Low Silicon Grey Cast Irons.—Close-grained grey irons, such as may be obtained by using a low silicon content, are moderately resistant to growth because of their small graphite size. Growth resistance may be further increased by incorporating, say, 0·5 per cent. of chromium which reduces the tendency for the pearlite to break down at high temperatures. The critical temperatures, at which cast iron undergoes a marked and rapid contraction on heating and expansion on cooling, remain, however, between 700° C. and 750° C., and operating temperatures are therefore normally kept below this to minimise risk of growth and cracking. Phosphide eutectic increases the rigidity of cast iron at temperatures up to 900° C., but as it increases the risk of cracking as a result of uneven heating the phosphorus content is generally kept below

0·1 per cent. A typical analysis of an iron in this class is as follows: total carbon 3·5 per cent., silicon 1·5 per cent., manganese 0·8 per cent., phosphorus 0·05 per cent., chromium 0·5 per cent. Such irons still possess a relatively low resistance to oxidation.

High Silicon Grey Cast Irons.—The structure of these irons should consist entirely of fine flake graphite in a matrix of silico-ferrite. Their heat-resisting properties depend on (*a*) the low carbon content and fine graphite structure which minimise growth, (*b*) the relatively high oxidation-resistance of the silico-ferrite, and (*c*) the increase in the critical temperature caused by the high silicon content. The application of these irons is sometimes limited by their rather brittle nature. A typical composition for *Silal* cast iron, which falls within this class, is as follows: total carbon 2·5 per cent., silicon 5 per cent., manganese 0·5 per cent., phosphorus 0·1 per cent.

High Aluminium Grey Cast Irons.—These irons also possess good resistance to growth and scaling at high temperatures, their oxidation-resistance being good even in atmospheres containing sulphur—a property not common to most other heat-resisting irons. In common with the high silicon irons, these irons are rather brittle. Standard *Cralfer* cast iron has the following composition, which may be modified, however, to suit special circumstances: total carbon 3 per cent., silicon 1 per cent., manganese 0·6 per cent., chromium 0·75 per cent., aluminium 7·25 per cent. Its tensile strength is 15–19 tons per sq. in., Brinell hardness 290–340, specific gravity 6·77, this latter being lower than for most cast irons because of the high aluminium content.

High Silicon Austenitic Grey Cast Irons.—The essential micro-constituents of this type of iron are fine flake graphite in a matrix of austenite, which is produced by the use of nickel as an alloying element. Chromium carbide may also be present in the matrix according to the amount of chromium in the iron. The good heat-resisting properties of this type of iron are due to (*a*) the low total carbon content and fine graphite structure which minimise growth, (*b*) the excellent oxidation resistance of the silicon-nickel-austenite and chromium carbide of the matrix, and (*c*) the absence, on heating, of critical temperatures at which sudden volume changes take place. Austenitic irons are ductile and retain a relatively high proportion of their strength up to moderately high temperatures. The composition of *Nicrosilal*, an iron of this type, may be as follows: total carbon 1·8 per cent., silicon 5 per cent., manganese 0·8 per cent., nickel 18 per cent., chromium 2 to 5 per cent. High chromium improves both the graphite structure and the oxidation-resistance of the matrix but impairs machinability, although even the 5 per cent. chromium iron is still machinable with high speed steel tools. *Ni-Resist*, which will be referred to under corrosion-resisting irons, is also an austenitic iron with good heat-resisting properties, the silicon content of this iron being, however, lower.

High Chromium White Cast Irons.—Cast irons containing 30 to 35 per cent. of chromium consist entirely of chromium carbide and ferrite and are therefore immune from growth. The oxidation resistance is very high, they are not attacked by sulphur-bearing atmospheres, their tensile strength is good—of the order of 25 tons per sq. in.—and they are not brittle like normal white irons. Their Brinell hardness of 250 to 350 varies directly as the carbon content and they may be machined fairly readily with carbide tools. Their resistance to corrosion and erosion is also good. A typical analysis is as follows: total carbon 1·7 per cent., silicon 2 per cent., manganese 0·8 per cent., chromium 33 per cent.

Corrosion-Resistance.

It is obviously impossible to assess the corrosion-resistance of a material without considering such factors as the nature and concentration of the corroding agent, the temperature and pressure of operation, etc. For this reason, without considering all sets of circumstances, it is only possible to state the position in fairly general terms.

Cast iron normally offers excellent resistance to corrosion compared with any other ferrous materials not specifically manufactured to resist it. The first requirements of any casting which is needed to withstand chemical corrosion are that it should be close-grained and sound. Apart from this there is, generally speaking, little that can be done to improve the corrosion-resistance of ordinary cast irons by modifying the amounts of the normal elements present. Low alloy additions of chromium, nickel or copper do under some circumstances, have a beneficial effect due partly to increased corrosion-resistance of the matrix, and partly to increased soundness and homogeneity of the casting. It is by means of high alloy additions, however, that irons of more universal corrosion-resistance are obtained, the three more commonly used types being given below.

High Silicon Grey Cast Irons.—These irons are produced under various trade names and for chemical use normally contains 14 to 17 per cent. of silicon. They are very resistant to all acids excepting hydrofluoric and hot concentrated hydrochloric. The high silicon irons are, however, rather brittle and difficult to machine.

High Chromium White Cast Irons.—The 30 to 35 per cent. chromium irons which have already been mentioned as heat-resisting irons are also very resistant to attack by a wide range of chemicals.

High Nickel Austenitic Grey Cast Irons.—The austenitic cast iron *Ni-Resist* is highly resistant to attack by a variety of corrosive media. The normal composition of Ni-Resist is: total carbon 3 per cent., silicon 1·5 per cent., manganese 1 per cent., nickel 14 per cent., copper 7 per cent.

Where freedom from the possibility of copper contamination is essential as in food processing plant and in the case of certain alkalis, the copper is omitted and the nickel content increased to 22 per cent. The austenitic iron *Nicrosilal*, previously mentioned as a heat-resisting iron, is also highly corrosion-resistant.

There are a number of protective finishes which can be applied to ordinary cast iron to give an increased degree of resistance against atmospheric and other mild forms of corrosion. These include *Parkerising*, by which treatment the surface of the casting is converted into a continuous coating of a mixture of iron and manganese phosphates, *Bonderising*, by which iron and copper phosphates are formed on the surface, and *Bower-Barffing*, by which the surface is converted to the black magnetic iron oxide. *Sherardising* gives an alloyed zinc coating to the casting, by heating the casting in contact with zinc powder, while *Calorising* is a similar process involving the use of aluminium powder. Cast iron may be protected with a continuous coating of other metals by hot dipping (particularly zinc and tin), electro-plating (chromium, nickel, copper, cadmium, tin, silver, etc.) or metal-spraying (copper, nickel, aluminium, tin, lead, zinc, etc.). It may also be a vitreous enamelled, painted, varnished or coated with bituminous compounds.

Physical Properties.

Specific Gravity.—The specific gravity of a cast iron obviously depends on the amounts and specific gravities of its individual constituents. The approximate specific gravities of the main constituents are as follows :—

Graphite	2·5
Ferrite	7·9
Pearlite	7·8
Cementite	7·65
Phosphide eutectic	7·2

It will be apparent that the specific gravity of an iron will decrease considerably with increasing graphitic carbon content, while increasing phosphorus content will also effect a slight reduction. Thus castings in grey iron to the lower British Standard grades will have specific gravities between 6·8 and 7·2, while the value for low-carbon high-duty irons will lie between about 7·3 and 7·5.

The specific gravity of white irons is about 7·7 to 7·75, increasing carbon content tending to decrease the density slightly.

Malleable cast irons also, of course, increase in density with decreasing graphitic carbon contents. The specific gravity of blackheart castings is normally in the range 7·25 to 7·4, while that of whiteheart castings is 7·3 to 7·7 according to thickness, *i.e.* according to the degree of decarburisation. The carbon-free skin, consisting solely of ferrite, will, of course, have a correspondingly high density.

Specific Heat.—For most cast irons the approximate specific heats are as follows: from atmospheric temperatures to (*a*) 100° C. 0·13, (*b*) 400° C. 0·14, (*c*) 700° C. 0·155, (*d*) 800° C. 0·17, (*e*) 1,000° C. 0·175, (*f*) 1,200° C. 0·22.

Coefficient of Thermal Expansion.—The expansion coefficient for cast iron increases with increase in temperature. For normal grey irons and malleable irons the value of the expansion coefficient varies from 10 to 15 $\times$ 10^{-6} per ° C. over the temperature range 0–700° C.

Austenitic cast irons of the Nicrosilal and Ni-Resist types have expansion coefficients between 16 and 20 $\times$ 10^{-6} per ° C. according to temperature. Austenitic irons with very high nickel contents have, however, low expansion coefficients. For example, *low expansion Ni-Resist* with the following composition has an expansion coefficient of 6 $\times$ 10^{-6} per ° C.: total carbon 2·2 per cent., silicon 1·5 per cent., manganese 0·8 per cent., nickel 34 per cent., chromium 2 per cent.

Thermal Conductivity.—The thermal conductivity of grey cast iron at normal temperatures is of the order of 0·10 to 0·135 c.g.s. units, decreasing to approximately 0·09 to 0·11 c.g.s. units at higher temperatures. Ferrite is a better conductor than pearlite. Silicon and phosphorus both reduce conductivity.

The thermal conductivity of blackheart malleable castings is about 0·15 c.g.s. units at atmospheric temperature. For whiteheart castings it is slightly higher in thin sections and lower in thick sections.

Austenitic grey irons have conductivities of 0·07 to 0·08 c.g.s. units.

Electrical Conductivity.—The electrical resistivity of grey cast iron is increased by increasing carbon and silicon contents, and variations between 60 and 200 microhms per cm. cube are obtainable by adjusting these two elements alone. The majority of normal grey irons fall, however, within the range 75 to 100 microhms per cm. cube. The resistivity of austenitic grey irons is normally about 150 microhms per cm. cube, this being associated with a low temperature coefficient of resistance. Blackheart malleable castings have an electrical resistivity of approximately 30 microhms per cm. cube, this increasing rapidly with temperature.

Magnetic Properties.—The permeability of ferrite is greater than that of pearlite. Silicon increases the permeability of cast iron both by increasing the ferrite content and by increasing the permeability of the ferrite itself. Permeability is reduced by manganese, nickel and chromium. The remanence, coercive force and hysteresis loss is reduced by silicon and increased by manganese, chromium and aluminium. Irons containing temper carbon have a higher permeability than comparable irons containing flake graphite.

It follows that for high permeability, grey irons should be ferritic, this structure being obtained by annealing, if necessary. The permeability of grey cast irons may be between 200 and 1,000 gauss, low silicon irons in the as-cast state being at the lower end of the range, the upper end representing the higher silicon ferritic irons.

Blackheart malleable cast iron, consisting of temper carbon and ferrite, has a high permeability of the order of 2,000 gauss, while that for whiteheart iron, which contains pearlite (except in the thinnest sections), is about 1,300 to 1,400 gauss.

Austenitic cast irons such as Nicrosilal and Ni-Resist are virtually non-magnetic. *No-Mag*, an austenitic iron specially developed for its non-magnetic properties, has the following approximate comparison: total carbon 3 per cent., silicon 1·5 per cent., manganese 7 per cent., nickel 11 per cent.

Another austenitic iron which is practically non-magnetic (permeability 1·1), but which is slightly more difficult to machine than No-Mag, has the following composition: total carbon 3·5 per cent., silicon 3 per cent., manganese 9 per cent., aluminium 3 per cent.

Special Note.

It should be noted the names of most of the special cast irons which have been mentioned are registered trade marks, and the production of these irons may be covered by patents.

STEEL.

The term 'steel' is applied in a general sense to those alloys of iron and carbon whose carbon is present entirely, or almost entirely, in the combined condition and whose total carbon content does not exceed 2·00 per cent. In a special sense the term is applied to alloys of iron and carbon with elements which are generally in excess of the carbon and whose presence considerably modifies the resulting alloys; such alloys are generally referred to as 'alloy or special steels.'

Steel always contains some or all of the elements—manganese, silicon, sulphur and phosphorus, and the allowable percentages are given in the following specifications. Sulphur and phosphorus should be kept as low as possible owing to the detrimental effect of these elements, although in some instances high sulphur contents may be desirable (see pp. 1162–1163).

Carbon renders the steel susceptible to hardening, although below 0·1 per cent. steel shows no appreciable hardening effect when rapidly cooled. As the carbon percentage increases, so does the tenacity and hardness, with a corresponding reduction in the ductility and toughness.

Steel is generally graded according to the percentage of carbon present. A general classification is as follows:—

Carbon under 0·10 per cent.	Dead soft steel, very mild or extra mild steel, very low-carbon steel.
Carbon 0·10 to 0·25 per cent.	Mild steel, soft steel, low-carbon steel.
Carbon 0·25 to 0·60 per cent.	Medium-carbon steel, medium high-carbon stee half-hard steel.
Carbon 0·60 to 1·20 per cent.	High-carbon steel, tool steel, hard steel.
Carbon over 1·20 per cent.	Extra-hard steel, very high-carbon steel.

The classification is not rigid, as no attempt has been made to standardise the various grades. A further classification is made with carbon tool steels, the steels being graded according to their 'temper.' (This term has no connection with the term 'tempering,' used in thermal treatment.) The Temper numbers are arbitrarily chosen by the makers, and the following is an example:—

Temper No.	1	2	3	4	5	6
Carbon Content (per cent.)	1½	1¼	1⅛	1	⅞	¾

The critical points in a straight carbon steel are dependant on the amount of carbon present, and these critical points control the temperatures for forging, annealing, hardening, and tempering. The higher the carbon content the lower the critical temperatures and the more sensitive is the steel. For the before-mentioned steels the recommended working temperatures are as follows:—

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F. 1159

Carbon Content. Per cent.	Maximum Forging Temp. ° C.*	Annealing Temp. ° C.	Hardening Temp. ° C.
1½	1000	720	760–780
1¼	1025	720	760–780
1⅛	1050	720	770–800
1	1100	720	780–800
⅞	1125	750	785–810
¾	1150	770	800–820

The tempering temperature depends on requirements. The higher the reheating (tempering) temperature, the greater is the loss in hardness, with a corresponding increase in the ductility.

Carbon Content. Per cent.	Suitable for:
1½	Special turning and planing tools.
1¼	The most suitable for turning and planing tools, twist drills and small cutters.
1⅛	For large turning tools, circular cutters, taps, small punches, reamers, screwing dies, and small shear blades.
1	For hot setts, large punches, large taps, cold chisels, etc.
⅞	For chisels, setts, dies, smiths' tools, large shear blades, masons' tools, etc.
¾	For hammers, general dies, and miners' drills.

Having selected the correct quality steel for the particular purpose, it is essential to give the correct heat treatment in order to obtain the best results. The hardening temperatures are given in the previous table. The following table gives the necessary tempering temperature:—

Tempering Temperature ° C.	Suitable for:
300	Springs, wood saws.
295	Circular saws for metal.
290	Cold chisels for wrought iron.
285	Moulding and planing cutters for soft wood, needles, cold chisels for cast iron.
280	Firmer chisels, saws for bone and ivory, cold chisels and setts for steel, gimlets.
275	Hot setts, axes, and adzes.
270	Augers, pressing cutters, flat drills for brass.
265	Coopers' tools, wood borers, twist drills, stone-cutting tools.
260	Plane irons, gouges, planing and moulding cutters.
255	Moulding and planing cutters for hard wood, punches and dies, cups, snaps, and shear blades.
250	Penknives, chasers, mill chisels and picks, taps, rock drills, screw-cutting dies.
245	Leather-cutting dies, boring cutters, reamers.
240	Bone-cutting tools, milling cutters, drills, wood engraving tools.
235	Paper cutters, planers for iron.
230	Ivory-cutting tools, planes for steel, screwing dies for brass, hammer faces.
225	Light turning tools, steel engraving tools.
220	Scrapers and lathe tools for brass.

* Forging should be finished about 700° C.

High-Speed Tool Steels.

Typical Compositions.

Type.	C.	Si.	Mn.	S. & P.	Cr.	W.	V.	Mo.	Co.
A	0·65/0·70	0·25	0·20	as low as possible	3·0/3·5	14·0/16·0	nil to 0·5	—	—
B	0·70/0·75	0·20	0·20	,,	3·5/4·5	16·0/18·0	0·5/1·0	—	—
C	0·75/0·80	0·30	0·20	,,	4·5/5·0	18·0/20·0	0·5	—	6·0
D	0·80	0·30	0·20	,,	5·0	20·0/22·0	1·5	0·5	12·0

Heat-Treatment.

Type.	Forge at	Anneal at	Preheat for hardening at*	Harden at	Temper at	Quench in
A	1100° to	800° C.	800° C.	1280–1300° C.	560–580° C.	Oil
B	1200° C.	,,	,,	1300–1320° C.	,,	Oil or Air
C	Reheat and	,,	,,	1300–1340° C.	,,	Air
D	cool off	,,	,,	1320–1350° C.	,,	Air

When hardened high-speed steel is tempered its hardness increases, a maximum value being obtained at some temperature 550° and 600° C. This phenomenon is known as *secondary hardness*. Beyond 600° C. the hardness falls off so that it is advisable to regard 580° C. as the safe maximum temperature. Typical hardness values are as follows:—

Hardening temp. °C.	C 0·70, Si 0·20, Mn 0·25, Cr 4·0, W 14·0, V 0·65.				C 0·75, Si 0·20, Mn 0·20, Cr 4·2, W 18·00, V 1·00.			
	Air-cooled.		Secondary hardened (575° C.)		Air-cooled.		Secondary hardened (575° C.)	
	Vickers.	Rockwell.	Vickers.	Rockwell.	Vickers.	Rockwell.	Vickers.	Rockwell.
1320	764	63·5	836	64·5	793	64·1	852	65·0
1328	760	62·9	831	64·6	780	63·4	864	65·2

Cold-Treatment.

Recently, it has been claimed that cold or 'deep-freeze' treatment after hardening, prior to secondary hardening, improves both hardness and cutting efficiency. Treatment consists in cooling tools, after hardening, to about − 70° C. (− 100° F.) for about 2 hours, followed by a usual secondary hardening treatment.

Atmosphere Control of Hardening Furnaces.

Some control over the nature of the furnace atmosphere is imperative in order to prevent decarburisation and scaling. It has been shown that the scaling is purely superficial in the absence of water-vapour. For the hardening of high-speed steel a muffle furnace, into which are lead the cooled products of combustion, yields most satisfactory results. By cooling the burnt gases, water-vapour is condensed and removed, and an inert atmosphere obtained in the heating chamber. In this way the risks of overheating and burning are minimised since the flame does not impinge on the work and the furnace walls are always hotter than the articles placed within them.

* The thermal conductivity of high speed is low at low temperatures and the steel must be heated slowly to a red heat, otherwise it tends to crack.

Super Hard Cutting Alloys.

Name.	C.	W.	Co.	Cr.	Va.	Mo.	Ta.	Ti.	Ce.	Fe.
Widia and Carboloy	6	88	6	—	—	—	—	—	—	—
Volomit	4	94	—	—	—	—	—	—	—	2
Lohmanit	?	93	—	—	—	7	—	—	—	—
Miramant	2	55	—	—	—	20	15	—	—	—
Tisit	3	60	—	—	—	—	—	5	2	30
Celsit	2·8	25	31	26	0·6	—	—	—	—	4·6
Stellite	2	22	45	25	—	—	—	—	—	1·3 to 2

Chromium-plated Tools.*

Chromium-plated tools are capable of giving greatly increased service over unplated tools in certain applications. The efficiency-ratio is the ratio of the number of articles machined with chromium-plated tools to the number similarly machined with unplated tools.

Tool.	Material.	Average Efficiency Ratio.
Reamer	Brass	5
Reamer	Bronze	3
Broach	Brass	5
Drill	Various mild and alloy steels	2
Slotting saw	Copper	2
Slotting saw	Mild steel	4
Heading die	Mild steel	2†
Wire-drawing die	Copper	1·5†
Plug gauge	Carbon steel	6†

CARBON AND ALLOY STEEL SPECIFICATIONS.

The following data are based upon B.S.I. Report 5005–1924‡ and upon the very complete specifications prepared by the S.A.E. (Society of Automotive Engineers, U.S.A.).

The range of steels given covers practically all general requirements. The selection of a given steel for a given part must depend upon an intimate knowledge of a number of important factors, such as the detailed design of the part and the severity of the service to be imposed, whether the part is to be machined or forged, machinability, and the method of manufacture. From the data given, a steel can be selected from a consideration of physical properties, after which the final choice depends upon such considerations as machinability, heat-treatment, etc.

Carbon Steels.

Reference No.	Carbon Range.	Manganese Range.	Phosphorus. Max.	Sulphur. Max.
1	0·05–0·15	0·30–0·60	0·045	0·05
2	0·10–0·20	0·30–0·60	0·045	0·05
3	0·15–0·25	0·30–0·60	0·045	0·05
4	0·20–0·30	0·50–0·80	0·045	0·05
5	0·25–0·35	0·50–0·80	0·045	0·05
6	0·30–0·40	0·50–0·80	0·045	0·05
7	0·35–0·45	0·50–0·80	0·045	0·05
8	0·40–0·50	0·50–0·80	0·045	0·05
9	0·45–0·55	0·50–0·80	0·045	0·05
10	0·90–1·05	0·25–0·50	0·045	0·05
11	0·45–0·55	0·90–1·20	0·045	0·05
12	0·55–0·70	0·90–1·20	0·045	0·05

* *Machinery.*
† Comparison refers to one plating only. Replating and further service is practicable.
‡ Superseded by B.S.S. 970—1947.

Steels Nos. 1 and 2.—Used for tubing and pressings; they are soft and ductile and will stand much deformation without cracking. Low tensile strength in annealed condition, but the yield point can be raised considerably by cold-working. They do not machine freely and will tear badly in turning, threading, or broaching. Machining properties can be improved by quenching in water from 925° C. Suitable for case-hardening.

Steel No. 3.—Forges well and machines better than Nos. 1 and 2. Can be used for machined forged, and case-hardened parts where no great strength is required. Can be drawn into tubes and cold rolled, and for cold-worked shapes that do not require deep drawing. When cold worked, yield point 18–35 tons per sq. in. in rounds not over ½ in. diam. or in flats ½ in. thick. Heat treatment will refine the grain after rolling or forging. Particularly suitable for case-hardening.

Steels Nos. 4 and 5.—Suitable for forged, machined, or cold-worked parts requiring higher physical properties than No. 3, also more responsive to heat-treatment and machines better. These steels should not be used for case-hardened parts.

Steels Nos. 6 and 7.—Medium carbon steels possessing good machining properties, and are suitable for small and medium-size forgings. After heat-treatment Brinell range 174–212.

Steels Nos. 8 and 9.—Medium carbon steels suitable for large forgings. Can also be used for a wide range of parts from bar stock. After heat-treatment Brinell range 192–235.

Steel No. 10.—Suitable for leaf springs and certain coiled springs. Heat-treatment: heat to 760–780° C., quench in oil, water, or brine, depending on size and shape of part. Temper to required hardness.

Steels Nos. 11 and 12.—Suitable for helical springs made of round cold drawn wire up to $\frac{3}{16}$ in. diam. No. 11 steel is recommended for wire from 0·025 to 0·100 in. diam., and steel No. 12 for wire 0·100 in. diam. and larger.

Free-cutting or Rapid-machining Steels.—Mild carbon steels containing high proportions of sulphur are used in automatic machines where machineability and good surface finish are the essential requirements, and where the steel is used for such purposes as low duty bolts, nuts, studs, sparking-plug bodies, etc. These steels are sometimes case-hardened and used for certain parts in cycles, automobiles, etc.

Contrary to earlier opinion, their high sulphur contents does not seriously influence the mechanical properties, provided that the sulphur is balanced by a sufficient proportion of manganese. Curiously enough, these steels only exhibit the best free-cutting properties when in the bright-drawn or cold-worked condition.

Typical specifications are given in the following table:—

	Ordinary Free-cutting Steel. No. 13.		Free-cutting Case-hardening Steel. No. 14.	
Composition.				
C	0·08/0·12		0·13/0·18	
Si	0·04 max.		0·10 max.	
S	0·2/0·3		0·10/0·15	
P	0·06 max.		0·05 max.	
Mn	0·60/0·80		1·1/1·5	
	As rolled.	Bright drawn.	As rolled.	Bright drawn.
Mechanical Properties.				
Y.P.	—	30 min.	—	30 min.
M.S.	25/30	32/40	28/33	34/42
Elong.	30 min.	14 min.	30 min.	10 min.
R. of A.	50 min.	35 min.	50 min.	35 min.
Izod	50	—	50	—

Steel No 13 fulfils the requirements of the well-known British Standards Institution Specification No. 32—Grade 4, the requirements for which are laid down as follows:—

Composition.				
C	Si	S	P	Mn
0·20 max.	0·10 max.	0·2/0·3	0·10 max.	0·6/1·2

Mechanical Properties.

Condition.	M.S.	Elong. %.
Cold-finished	28 min.	14 min.
Other-finishers . . .	25 „	26 min.

Actual results on high-sulphur free-cutting steels are given in the following table:—

Composition.				
C	0·12		0·10	
Si	0·005		trace	
S	0·234		0·424	
P	0·030		0·038	
Mn	0·85		1·03	
	As rolled.	Bright drawn.	As rolled.	Bright drawn.
Mechanical Properties.				
Y.P.	16·8	26·0	15·2	31·0
M.S.	24·8	26·8	25·0	31·6
Elong.	34	22	33	15
R. of A.	57	54	50	39
Izod	55, 60, 57	37, 36, 36	50, 50, 50	27, 27, 26

(*Gregory.*)

Lead-bearing Steels —During recent years lead has been introduced into steel, with the object of improving its machineability, both to the usual high-sulphur free-cutting steels and to steels of more normal sulphur contents, even including certain alloy steels. In regard to the lower sulphur steels, the introduction of lead considerably improves their machineability and also results in improved tool-life. The amount of lead added for this purpose is between 0·15 and 0·30 per cent.

Lead is insoluble in steel, but in the leaded steels exists in the form of emulsified sub-microscopic particles. This distribution can only be obtained by proper steel-making technique, which is protected by patent, and the steel, if used for ordnance purposes, is required to pass a standard 'sweating' test. This consists in reheating the leaded-steel to a specified temperature for a given time, after which no exudation of lead should occur. It is claimed that the introduction of lead does not influence the mechanical properties of the steel.

MOLYBDENUM STEELS.

Reference Letter.	Carbon Range.	Manganese Range.	Chromium Range.	Nickel Range.	Molybdenum Range.
A	0·25–0·35	0·40–0·70	0·50–0·80	—	0·15–0·25
B	0·35–0·45	0·40–0·70	0·80–1·10	—	0·15–0·25
C	0·45–0·55	0·40–0·70	0·80–1·10	—	0·15–0·25
D	0·10–0·20	0·30–0·60	—	1·5–2·0	0·20–0·30

In all cases sulphur max. = 0·045 ; phosphorus max. = 0·040.

In general these steels have similar physical characteristics to the corresponding nickel and nickel-chromium steels. Steel D is suitable for case-hardening.

NICKEL AND NICKEL-CHROMIUM STEELS.

	Type of Steel.	Specification.	Mechanical Tests. Y.P. Tons/sq. inch. Min.	MS. Tons/sq. inch. Min.	MS. Tons/sq. inch. Max.	E.% Min.	R.A.% Min.	B.H. No. Min.	B.H. No. Max.	Izod. ft. lb. Min.	Heat-Treatment. Carburise at ° C.	Refine at ° C.	Quench in	Harden at ° C.	Quench in	Temper at ° C.	Quench in
Case-Hardening Steels.	3 % nickel .	BS5005/103	—	42	—	18	45	—	—	40	900/950	860	—	760	W or O	—	—
	3 % nickel .	3 S15	—	45	60*	18	45	—	—	40	880/930	860	—	770	W or O	max. 200†	—
	5 % nickel	BS5005/104	—	55	—	15	40	—	—	20	900/950	(a) 830 (b) omit	— —	760 770	W O	—	—
	5 % nickel .	S90	—	65	—	13	—	—	—	30	850/900	850	O or A	760	O	max. 200†	—
	5 % nickel	S67	—	40	60	20	45	—	—	50	880/930	830	—	740	O	max. 200†	—
	3½ % nickel-chromium	BS682—1936	—	55	70	15	40	255	321	35	—	—	—	780 to 800	O	—	—
	4¼ % nickel-chromium	S82	—	85	—	12	35	—	—	25	850/900	830	O or A	760	O	max. 200†	—
Structural Steels.	1 % nickel normalised	BS5005/203	18	35	45	20	40	146	201	—	Normalise: Heat to 830/860° C. and cool in air.						
	1 % nickel normalised	3 S6	**	35	45	20	—	146	201	20‡	Normalise: Heat to 850° C. and cool in air.						
	1 % nickel (heat-treated) .	BS5005/204	25	40	50	20	45	174	241	30	—	—	—	850	O	max. 680	—
	1 % nickel (heat-treated) .	S76	—	40	50	22	40	174	223	35§	—	—	—	850	O	500 to 650	W or O

Structural Steels.	3 % nickel	BS5005/401	32	45	—	22	50	192	—	40	—	—	—	850	O	max. 660	—
	3½ % nickel	BS5005/402	40	55	65	18	50	241	311	35	—	—	—	850	O	max. 660	—
	3½ % nickel	S69	—	55	65	18	50	241	293	35	—	—	—	830	O	530 to 630	W or O
	55 ton nickel-chromium	BS5005/501	45	55	65	18	50	241	311	40	—	—	—	830	O	580 to 660	—
	55 ton nickel-chromium	3 S11	—	55	65	18	50	241	293	40	—	—	—	830	O	560 to 660	W or O
	65 ton nickel-chromium	S65	—	65	70	17	40	293	321	35	—	—	—	830	O	500 to 600	W or O
	65/75 ton nickel-chromium	2 S81	— ††	65	75	16	—	293	341	35 ‡‡	—	—	—	830	O	500 to 650	W, O or A
	Air-hardening nickel-chromium	BS5005/502	—	100	—	8 to 6	18 to 10	444¶	—	9 to 5	—	—	—	820	A	max. 250	—
	Air-hardening nickel-chromium	2 S28	—	100	—	12	25	444‖	—	15	—	—	—	820	A	max. 250	—

[*Bureau of Information on Nickel.*

* This limit only applies to material of 1½ in. diameter (or width across flats) and over.
† Optional.
‡ When bars, etc., are over 6 ins. diameter, a core test shall give a minimum value of 15 ft.-lbs.
§ For bars 4 ins. diameter (or width across flats) and under.
¶ In the softened for machining condition maximum B.H. No. = 269.
‖ In the softened for machining condition maximum B.H. No. = 302.
** 0·1 % proof stress (optional) 20 tons per square inch minimum.
†† 0·1 % proof stress (optional) 55 tons per square inch minimum.
‡‡ When bars, etc., are over 4 ins. diameter, a core test shall give a minimum value of 25 ft.-lbs.

W = water. O = oil.

NICKEL AND NICKEL-CHROMIUM STEELS.

	Type of Steel.	Specification.	Chemical Composition. C. Min.	C. Max.	Si. Max.	Mn. Min.	Mn. Max.	S. Max.	P. Max.	Ni. Min.	Ni. Max.	Cr. Min.	Cr. Max.	V. Max.	Mo. Max.	W. Max.
Case-Hardening Steels.	3 % nickel	BS5005/103	—	0·15	0·30	0·20	0·60	0·05	0·05	2·50	3·50	—	0·30	—	—	—
	3 % nickel	3 S15	0·10	0·15	0·30	0·20	0·60	0·05	0·05	2·75	3·50	—	0·30	—	—	—
	5 % nickel	BS5005/104	—	0·15	0·30	—	0·40	0·05	0·05	4·50	6·00	—	0·30	—	—	—
	5 % nickel	S90	—	0·16	0·30	—	0·60	0·05	0·05	4·50	5·50	—	0·30	0·25†	0·50†	1·00†
	5 % nickel	S67	0·08	0·14	0·30	—	0·35	0·05	0·05	4·60	5·20	—	0·10	—	—	—
	3½ % nickel-chromium	BS682—1936	0·10	0·15	0·30*	0·30	0·60	0·05	0·05	3·00	3·50	0·80	1·10	—	—	—
	4½ % nickel-chromium	S82	—	0·18	0·30	—	0·50	0·05	0·05	4·00	4·50	1·00	1·60	0·25†	0·50†	1·00†
Structural Steels.	1 % nickel normalised	BS5005/203	0·35	0·45	0·30	0·40	0·80	0·06	0·06	—	1·00	—	—	—	—	—
	1 % nickel normalised	3 S6	0·35	0·45	0·30	—	1·20	0·05	0·05	—	1·00	—	—	—	—	—
	1 % nickel (heat-treated)	BS5005/204	0·35	0·45	0·30	0·40	0·80	0·06	0·06	0·30	1·00	—	—	—	—	—
	1 % nickel (heat-treated)	S76	0·35	0·45	0·30	—	1·20	0·05	0·05	—	1·00	—	—	—	—	—
	3 % nickel	BS5005/401	0·25	0·35	0·30	0·35	0·75	0·05	0·05	2·75	3·50	—	0·30	—	—	—
	3½ % nickel 3½ % nickel	BS5005/402 S69	0·35	0·45	0·30	0·50	0·80	0·05	0·05	3·25	3·75	—	0·30	—	—	—
	55 ton nickel-chromium	BS5005/501	0·28	0·34	0·30	0·45	0·70	0·05	0·05	3·00	3·75	0·50	1·00	—	—	—
	55 ton nickel-chromium	3 S11	0·25	0·35	0·30	0·45	0·70	0·05	0·05	3·00	3·75	0·50	1·00	0·25†	0·65†	1·00†
	65 ton nickel-chromium	S65	0·22	0·28	0·30	0·35	0·65	0·05	0·05	2·75	3·50	1·00	1·40	0·25†	0·65†	1·00†
	65/75 ton nickel-chromium	2 S81	0·28	0·35	0·30	—	0·70	0·05	0·05	3·00	3·75	0·50	1·30	0·25†	0·50†	1·00†
	Air-hardening nickel-chromium	BS5005/502	0·25	0·35	0·30	0·35	0·60	0·05	0·05	3·75	4·75	1·00	1·50	—	—	—
	Air-hardening nickel-chromium	2 S28	0·25	0·32	0·30	0·35	0·60	0·05	0·05	3·75	4·50	1·00	1·50	0·25†	0·65†	1·00†

* Minimum 0·1 %

† Optional.

[*Bureau of Information on Nickel.*]

CHROMIUM STEELS.

Reference Letter.	Carbon Range.	Manganese Range.	Chromium Range.	Sulphur Max.	Phosphorus Max.
E	0·15–0·25	0·30–0·60	0·60–0·90	0·045	0·04
F	0·35–0·45	0·50–0·80	0·80–1·10	0·045	0·04
G	0·45–0·55	0·50–0·80	0·80–1·10	0·045	0·04
H	0·95–1·10	0·20–0·50	1·20–1·50	0·030	0·03

Steel E. This steel is intended primarily for case-hardening. For structural purposes the chromium content of this steel gives a deeper penetration of the effect of heat treatment than can be obtained in a straight carbon steel having a similar carbon content.

Steel G is an oil-hardening type of steel intended for heat-treated forgings that require greater strength and toughness than are obtainable with straight carbon steels, or with some of the simple alloy steels having a lower carbon content. It also has a wider heat-treating range than the other simple alloy steels.

Steel H is used chiefly for the races and balls or rollers of anti-friction bearings. The chromium and carbon contents cause a maximum penetration of the effect of heat treatment and give a high degree of hardness and resistance to wear.

CHROMIUM VANADIUM STEELS.

Reference Letter.	Carbon Range.	Manganese Range.	Chromium Range.	Vanadium.	
				Min.	Desired.
I	0·15–0·25	0·50–0·80	0·80–1·10	0·15	0·18
J	0·25–0·35	1·0 –1·5	0·80–1·10	0·15	0·18
K	0·35–0·45	1·0 –1·5	0·80–1·10	0·15	0·18
L	0·45–0·55	0·50–0·80	0·80–1·10	0·15	0·18
M	0·90–1·05	0·20–0·45	0·80–1·10	0·15	0·18

Steel I is intended primarily for case-hardening and can be used for structural parts after suitable heat treatment.

Steels J and K. These steels can be used in place of the nickel and nickel-chrome steels of similar carbon content for heat-treated forgings requiring greater strength, toughness and resistance to fatigue than it is possible to obtain with straight carbon steels.

Steel M can be used for anti-friction bearings and is used extensively for machine-tool parts, being primarily a tool steel.

SILICO-MANGANESE STEELS.

Reference No.	Carbon Range.	Manganese Range.	Silicon Range.
N	0·45–0·55	0·60–0·90	1·80–2·20
O	0·55–0·65	0·60–0·90	1·80–2·20

Sulphur and phosphorus = 0·045 max.

These steels are essentially spring steels, but may be used for gears. In either case the steel must be suitably heat-treated.

HEAT TREATMENT OF STEELS.

In order to obtain the desired physical properties from steels it is generally necessary to subject them to a thermal treatment. The subject of heat treatment embraces the operations of (*a*) normalising, (*b*) annealing, (*c*) hardening, (*d*) cementing, carburising or case-hardening, (*e*) the tempering or drawing of hardened steels.

The nomenclature and definitions relating to heat treatment as given by B.S.I. Report No. 5005—1924 are as follows:

(*a*) **Normalising.** Normalising means heating a steel (however previously treated) to a temperature exceeding its upper critical range, and allowing it to cool freely in air. It is desirable that the temperature of the steel shall be maintained for not less than 15 minutes, and shall not exceed the upper limit of the critical range by more than 50° Centigrade.

(*b*) Annealing. Annealing means reheating followed by slow cooling. Its purposes may be

(1) To remove internal stresses or to induce softness, in which case the maximum temperature may be arbitrarily chosen.

(2) To refine the crystalline structure in addition to the above, in which case the temperature used must exceed the upper critical range as in normalising.

(*c*) Hardening. Hardening means heating a steel to its normalising temperature, and cooling more or less rapidly in a suitable medium, *e.g.* water, oil or air.

(*d*) Cementing. Cementing means heating a steel above its normalising temperature in a medium which will increase its carbon content. The core of a case-hardened bar is the interior portion of the bar which is substantially unaffected by the cementing process.

(*e*) Tempering. Tempering means heating a steel (however previously hardened) to a temperature below its lower change-point with the object of reducing the hardness or increasing the toughness to a greater or less degree.*

Effect of Tempering.

3½ per cent. nickel steel, oil-hardened at 860° C. and tempered at the temperatures indicated.

Tempered at. ° C.	Yield Point. Tons per sq. in.	Ultimate Stress. Tons per sq. in.	Elongation. %.	Reduction of Area. %.	Impact. Test. Ft.-lbs.	Brinall No.
Not tempered	102·0	116·0	10·0	23·0	6	514
200	92·5	101·0	10·5	32·5	5	461
300	83·0	92·5	12·0	40·0	5	415
400	66·4	76·3	15·1	50·0	11	352
500	54·5	63·5	20·0	59·5	50	285
600	44·0	53·0	25·0	64·5	73	241
650	40·0	49·0	27·0	66·0	82	235

(*f*) Refining (cemented parts). Refining means reheating a steel to its normalising temperature, and is usually followed by quenching.

See also p. 1188.

(*g*) Sub-critical Annealing. As its name implies, this means reheating the steel to a temperature below the lower critical point and in some ways may thus be regarded as high-temperature tempering. It differs from the latter, however, in that a previous hardening is not a preliminary operation, the reheating temperature is generally somewhat higher and conducted for much longer periods of time and, moreover, the objects of sub-critical annealing are entirely different from those of tempering. During sub-critical annealing the carbide is ' balled up ' and the steel softened, a necessary preliminary for such purposes as the cold-rolling or deep-pressing of strip, and the cutting and subsequent hardening of certain high-carbon tool steels.

Grain Growth and Grain Size Control.

When a steel is above its critical range *grain-growth* sets in, *i.e.* some of the austenite-crystals of which it is composed grow at the expense of others. The higder the temperature the larger is the grain size. If reheated to very high temperatures the steel exhibits a coarse fracture even when slowly cooled or rapidly cooled by quenching in oil or water. The steel is then described as *over-heated*. Even when fully tempered the influence of the quenching of steel from too high a temperature is revealed by mechanical tests; the reduction of area per cent. and impact values are then lower than expected.

American workers have shown that the grain size of steel can be controlled by the addition of aluminium to the ingot moulds during the time that the molten steel is being cast into them. From 0·02 to 0·03 per cent. of aluminium is added, some of which is oxidised to alumina, Al_2O_3, the remainder existing in solid solution. Swinden and Bolsover, in this country, have thoroughly investigated this question of grain-size control, and express the opinion that it is essentially a question of the ultimate degree of deoxidation prior to the addition of aluminium. With proper furnace technique it is possible to produce steels of relatively fine or relatively coarse grain at will, as determined by the *McQuaid-Ehn carburising test*. This consists in packing the specimen of the steel in a good case-hardening compound, contained in a sealed box, heating for 2 hours and soaking for 6 hours at 927° C., followed by a slow cooling to 400° C. or less before emptying. A section is then cut, polished and etched with boiling alkaline sodium picrate solution. The carburised case then has a clearly defined grain size as indicated by the hyper-eutectoid carbide and the structure is examined at a magnification of 100 diameters on a projection microscope.

* Tempering generally implies subsequent cooling in air from the tempering temperature but sometimes the steel is quenched in oil or water after tempering, mainly with the object of avoiding temper-brittleness. (See also p. 1184.)

The grain-size number is then indicated by the number of grains per sq. in. at this magnification, as indicated as follows:—

Grain Size. Index *N*.	Number of Grains per sq. in. × 100.		
	Mean.	Maximum.	Minimum.
1	1	1·5	—
2	2	3	1·5
3	4	6	3
4	8	12	6
5	16	24	12
6	32	48	24
7	64	96	48
8	128	—	96

Swinden and Bolsover have shown that considerable latitude is permissible in regard to carburising temperature, time at carburising temperature and temperature to which slowly cooled. Some idea of the influence of grain size as indicated by the McQuaid-Ehn test is given by the following table, due to Swinden and Bolsover (*Journ. Iron and Steel Inst.*, 1936, vol. ii.).

***Steel* A. C, 0·39 %. Mn, 0·79 %. Si, 0·140 %. S, 0·019 %. P, 0·038 %.**

Size.	Treatment.	Grain Size (N).	Max. Stress. Tons per sq. in.	Yield Point. Tons per sq. in.	Elongation on 2 ins. %.	Reduction of Area. %.	Izod Impact Figures. Ft.-lbs.			
										Average.
1½ in. dia.	O.Q. 850°, T. 650° C.	2–3	45·7	30·9	27·0	65·2	21	29	15	21·7
	,, ,,	7	45·4	34·8	28·5	63·6	70	81	88	79·7
	W.Q. 850°, T. 500° C.	2–3	60·9	45·8	16·5	52·4	33	34	32	33·0
	,, ,,	7	59·2	43·5	20·5	54·3	54	60	60	58·0
3½ × 1½ in.	O.Q. 850°, T. 650° C.	2–3	46·2	31·7	28·0	63·6	39	42	45	42·0
	,, ,,	7	45·0	30·4	29·5	62·6	74	72	96	80·7
	W.Q. 850°, T. 500° C.	2–3	53·9	39·0	20·0	49·6	30	30	26	28·7
	,, ,,	7	52·6	37·4	22·5	52·4	60	55	69	61·3

***Steel* B. C, 0·50 %. Mn, 1·02 %. Si, 0·180 %. S, 0·027 %. P, 0·022 %.**

Size.	Treatment.	Grain Size (N).	Max. Stress. Tons per sq. in.	Yield Point. Tons per sq. in.	Elongation on 2 ins. %.	Reduction of Area. %.	Izod Impact Figures. Ft.-lbs.			
										Average.
1½ in. dia.	O.Q. 840°, T. 600° C.	3	56·5	39·2	23·0	54·8	24	19	22	21·7
	,, ,,	7	53·4	37·7	26·0	57·2	57	56	51	54·7
	,, T. 625° C.	3	55·0	37·4	24·0	54·8	15	11	25	17·0
	,, ,,	7	51·6	35·0	24·5	59·2	56	48	42	48·7
	,, T. 560° C.	3	53·1	37·0	25·5	59·2	24	13	13	16·7
	,, ,,	7	50·0	35·1	27·0	61·6	66	86	76	76·0
	,, T. 675° C.	3	50·8	36·0	23·5	59·2	19	22	29	23·3
	,, ,,	7	47·6	34·6	27·5	63·6	72	64	78	71·3
3 × 1½ in.	O.Q. 840°, T. 620° C.	3	53·5	34·2	22·0	49·6	11	25	15	17·0
	,, ,,	7	52·0	34·8	24·0	49·6	30	42	40	37·3
	O.Q. 840°, T. 650° C.	3	52·7	34·8	22·5	52·4	19	22	19	20·0
	,, ,,	7	51·2	34·4	22·5	52·4	37	43	43	41·0

MASS EFFECT.

Summarising the effects of grain size on mechanical properties, the maximum stress and yield-point values of fine-grain steels are slightly lower and the elongation and reduction of area values slightly greater than those of coarse-grain steels. The greatest difference is found in the Izod impact values: if these are accepted as a criterion of toughness, then fine-grained steels are infinitely superior to coarse-grained steels of otherwise similar composition and properties. Swinden and Bolsover obtained similar results on alloy steels of low carbon contents; with steels of higher carbon contents the differences between the properties of fine and coarse-grained steels becomes less marked.

The McQuaid-Ehn test was really the outcome of experiments on case-hardening. It was found that the fine-grained steels exhibited *abnormality, i.e.* carburisation was difficult and only a very thin carburised case obtained. For carburising, a relatively large grain size is desirable. A further important point is the fact that fine-grained steels do not harden to the same depth as steels of coarser grain. Shallow hardening is a characteristic of fine-grained steel.

It should be realised that it is possible to overheat fine-grain steels, just as coarse-grain steels; the essential difference lies in the fact that for a given degree of overheating the grain size is considerably less in the former instance than in the latter.

Effect of Mass.

The effect of mass must receive serious consideration when dealing with the physical properties of any heat-treated steel.

The hardening operation consists of heating the steel through and just above its critical range in order to obtain the condition of solid solution, and quenching with sufficient rapidity to retain this condition. With large masses it is practically impossible to obtain a uniform temperature throughout the mass on heating, the interior portion always being at a lower temperature than the outside. On quenching the heat is absorbed from the outside, and in many cases it is impossible, even with the most drastic quenching, to remove the heat sufficiently rapidly to retain the desired structure in the core. Again, with very thin sections, the rate of cooling may be sufficiently rapid if carried out in air. Only by experience can the effect of mass be correctly estimated. The effect of mass on heat-treatment is greatest with carbon steels and least with steels containing large proportions of alloying elements.

The following table shows the effect of mass on the physical properties of a 5 per cent. nickel case-hardening steel. Bars of varying diameters were given exactly the same treatment and then tested:

Size of Bar. In.	Heat Treatment.	Yield Point. Tons per sq. in.	Ultimate Stress. Tons per sq. in.	Elongation. %.	Reduction of Area. %.	Impact. Test. Ft.-lbs.	Brinell No.
½	Carburised at 860° C. Reheated to 860° C. Quenched in water Reheated to 750° C. Quenched in water	40·0	51·0	27·0	66·1	55	235
¾		34·5	44·6	31·0	68·3	90	212
1		31·0	40·0	35·0	70·0	105	197
1¼		27·8	37·5	36·7	72·2	111	179
1½		26·7	36·2	36·8	73·5	111	174
1¾		26·5	36·1	37·0	73·5	110	174
2		26·2	36·0	37·0	72·5	104	174
2½		26·0	36·0	37·0	72·0	90	170

Due cognisance of the mass effect is taken in the British Standards Institution's 'War Emergency' Schedule 970 (1942). Some details are given in the following pages, as it is fully anticipated that this schedule will persist for many years.

British Standard Schedule 970 (En Series).

Wrought Steels for General Engineering Purposes up to 6 in. Ruling Section.

En No.	Type of Steel.	Composition. C.	Mn.	Ni.	Cr.	Mo.	Others.	Properties. M.S.	E.	Imp'ct.	Limiting Size.
1	Free cutting steel	<0·2	0·6/1·2	—	—	—	S. 0·2/0·3 P. <0·1	28	14	—	CR or CD
1A	Do. . .	0·07/0·15	0·8/1·2	—	—	—	S. 0·2/0·3 P. <0·07	25	26	—	Not CR or CD
2	Mild steel . .	0·15/0·25	0·4/0·8	—	—	—	—	28/33	25	—	R or A.C.
		<0·25 0·25/0·30 0·30/0·35	<1·25	—	—	—	—	35/45	15	—	CD up to 1¼" CD 1¼"/1¾" CD 1¾"/3"
2A	Mild steel for cold processing	<0·2	<0·8	—	—	—	—	26 20	28 28	— —	R or A.C. Annealed
3	'20' carbon steel	<0·25	<1·0	—	—	—	—	25/35 28	25 17	— —	R If CR or CD
4	'25' carbon steel, A.C.	<0·3	<1·2	—	—	—	—	28/38	25	20	—
5	'30' carbon steel, H.T.	0·25/0·35	0·5/1·2	—	—	—	—	N 30 P 35	25 25	25 20	Impacts up to 2½" only
6	35/45 ton bright carbon steel	0·15/0·4	0·5/0·9	—	—	—	—	35/45	15	20 15 10	Up to ¾" ¾" to 1⅛" Above 1⅛"
7	Bright free cutting steel	0·1/0·3	0·5/1·75	—	—	—	S. <01/0·3 P <0·06	35/45	15	20 15 10	Up to ¾" ¾" to 1⅛" Above 1⅛"
7A	Do. (Suitable for C.H.)†	0·12/0·18	1·0/1·5	—	—	—	S. 0·08/0·15 P <0·06				
8	'40' carbon steel, R. or A.C.	0·3/0·45	>1·5	1·0 opt.	—	—	—	35/45	20	10	—

BRITISH STANDARD SCHEDULE 970 (EN SERIES)—*contd.*

En No.	Type of Steel.	Composition.						Properties.			
		C.	Mn.	Ni.	Cr.	Mo.	Others.	M.S.	E.	Imp'ct.	Limiting Size.
9	'55' carbon steel, A.C.	0·5/0·6	0·5/0·8	—	—	—	—	45 50/65	18 12·5	— —	A.C. CR or CD
10	'55' carbon steel, H.T.	0·5/0·6 0·5/0·7	0·4/0·75 Above 1⅛″ diam.	—	—	—	—	S 50 T 55	18 15	— —	Tests up to 1⅛″ only
11	'60' carbon chorme steel	0·5/0·7	0·5/0·8	—	0·5/0·8	—	—	T 55 V 65	15 12	25 12	Impacts up to 2½″ diam. only
12	'40' carbon steel, H.T.	0·3/0·45	<1·5	<1·0 opt.	—	—	—	40	22	35 25	Up to 4″ 4″ to 6″
13	Mn.-Ni.-Mo. steel	0·15/0·25	1·4/1·8	0·4/0·7	—	0·15/0·35	—	40	22	40	6″
14	C.-Mn. steel, A.C. or H.T.	0·2/0·3	1·3/1·8	—	—	—	—	Q 40 R 45 — 38	20 20 20	35 30 —	H.T. 4″ H.T. 2½″ A.C. 6″
15	C.-Mn. steel (higher tensile)	0·3/0·4	1·3/1·7	—	—	—	—	Q 40 Q 40 R 45 S 50	22 22 20 20	25 35 30 30	4″ to 6″ 4″ 2½″ ⅞″
16	Nn.-Mo. steel .	0·25/0·4	1·3/1·8	—	—	0·2/0·35	—	R 45 S 50 T 55 U 60 V 65	22 20 18 17 16	40 40 40 36 35	6″ 4″ 2½″ 1½″ ⅞″
17	Mn.-Mo. steel .	0·3/0·4	1·3/1·8	—	—	0·25/0·55	—	R 45 S 50 T 55 U 60 V 65	22 20 18 17 16	40 40 40 35 35	6″ 6″ 4″ 2½″ 1½″

BRITISH STANDARD SCHEDULE 970 (EN SERIES)—*contd.*

En No.	Type of Steel.	Composition.						Properties.			
		C.	Mn.	Ni.	Cr.	Mo.	Others.	M.S.	E.	Imp'ct.	Limiting Size.
18	1 % Cr. steel	0·35/0·45	0·6/1·2	—	0·8/1·2	—	—	R 45	22	40	4"
								S 50	20	40	2½"
								T 55	18	40	1⅛"
19	1 % Cr.-Mo. steel	0·35/0·45	0·5/0·8	—	0·9/1·5	0·2/0·4	—	R 45	22	40	6"
								S 50	20	40	4"
								T 55	18	40	4"
								U 60	17	35	2½"
								V 65	16	35	1⅛"
								W 70	15	30	1⅛"
20	1 % Cr.-Mo. steel	0·22/0·50	0·4/0·8	<0·3	0·5/1·5	0·4/1·0	—	T 55	17	40	2½"
								V 65	16	35	1⅛"
21	3 % Ni. steel	0·25/0·35	0·35/0·75	2·75/3·5	<0·3	—	—	R 45	22	40	4"
								S 50	20	40	2½"
22	3½ % Ni. steel	0·35/0·45	0·5/0·8	3·25/3·75	<0·3	—	—	S 50	20	40	4"
								T 55	18	35	2½"
23	3 % Ni.-Cr. steel	0·25/0·35	0·45/0·7	2·75/3·5	0·5/1·0	<0·65 opt.	—	S 50	20	40	6"
								T 55	18	40	6"
								U 60	17	35	6"
								V 65	16	35	2½"
24	1½ % Ni.-Cr.-Mo. steel	0·35/0·45	0·45/0·7	1·3/1·8	0·9/1·4	0·2/0·35	—	S 50	20	40	6"
								T 55	18	40	6"
								U 60	17	35	4"
								V 65	16	35	2½"
								W 70	15	30	1⅛"
								X 75	14	25	1⅛"
								Y 80	14	25	1⅛"
								Z 100	8	8	1⅛"

BRITISH STANDARD SCHEDULE 970 (EN SERIES)—*contd.*

En No.	Type of Steel.	Composition. C.	Mn.	Ni.	Cr.	Mo.	Others.	Properties. M.S.	E.	Imp'ct.	Limiting Size.
25	2½ % Ni.-Cr.-Mo. steel (medium carbon)	0·27/0·35	0·5/0·7	2·3/2·8	0·5/0·8	0·4/0·7	—	T 55	18	40	6″
								U 60	17	35	6″
								V 65	16a	35	6″
								W 70	15a	30	4″
								X 75	14	25	2½″
								Y 80	14	25	2½″
								Z 100	10	10	2½″
26	2½ % Ni.-Cr.-Mo. steel (high carbon)	0·36/0·44	0·5/0·7	2·3/2·8	0·5/0·8	0·4/0·7	—	U 60	17	35	6″
								V 65	16a	35	6″
								W 70	15a	30	6″
								X 75	14a	25	6″
								Y 80	14a	25	6″
								Z 100	10a	10	4″
27	3 % Ni.-Cr.-Mo. steel	0·25/0·35	<0·7	3·0/3·75	0·5/1·3	0·2/0·65	—	T 55	18	40	6″
								U 60	17	35	6″
								V 65	16a	35	6″
								W 70	16a	30	4″
28	3½ % Ni-Cr.-Mo. steel	0·25/0·40	<0·7	3·0/4·5	0·75/1·5	0·2/0·65	—	U 60	17	35	6″
								V 65	16a	35	6″
								W 70	15a	30	4″
								Y 80	14	25	2½″
29	3 % Cr.-Mo. steel	0·15/0·35	<0·65	<0·4	2·5/3·5	0·3/0·7	—	R 45	22	45	6″
								S 50	20	40	6″
								T 55	18	40	6″
								U 60	17	35	6″
								V 65	16a	35	4″
								W 70	15a	30	4″
								Z 100	10	10	2½″

BRITISH STANDARD SCHEDULE 970 (EN SERIES)—*contd.*

En No.	Type of Steel.	Composition. C.	Mn.	Ni.	Cr.	Mo.	Others.	Properties. M.S.	E.	Imp'ct.	Limiting Size.
30	4½% Ni.-Cr. steel	0·25/0·35	0·35/0·6	3·75/4·5	1·0/1·5	<0·65 Opt.	—	100	10*a*	10	Up to 6" if O.H. Up to 2½" if A.H.
31	1% carbon Cr. steel for parts of max. hardness	0·9/1·2	0·3/0·75	—	1·0/1·6	—	—	Brin. 229 Max. when softened.			
32	Carbon C.H. steel	<0·2	0·4/1·0	—	—	—	—	32	20	40	—
33	3% Ni. C.H. steel	<0·16	0·2/0·6	2·75/3·5	<0·3	—	—	45	18	40	—
34	2% Ni.-Mo. C.H. steel	0·12/0·20	0·3/0·6	1·5/2·0	—	0·2/0·3	—	45	18	40	—
35	2% Ni.-Mo. C.H. steel	0·20/0·28	0·3/0·6	1·5/2·0	—	0·2/0·3	—	55	15	25	—
36	3% Ni.-Cr. C.H. steel	<0·18	0·3/0·6	3·0/3·75	0·6/1·1	—	—	T55/75 V65/80	15 13	35 30	— —
37	5% Ni. C.H. steel	<0·16	<0·45	4·5/5·2	<0·3	—	—	40/60	20	50	
38	5% Ni. C.H. steel	<0·16	<0·6	4·5/5·5	<0·3	0·15/0·3 Opt.	—	65	13	30	—
39	4¼% Ni.-Cr. C.H. steel	0·12/0·18	<0·5	3·8/4·5	1·0/1·4	0·15/0·35 Opt.	—	85	12	25	—
40	3% Cr.-Mo. stee for nitriding	0·15/0·35	<0·65	<0·4	2·5/3·5	0·3/0·7	<0·25V Opt.	R 45 S 50 T 55 U 60	22 20 18 17	45 40 40 35	6" 6" 6" 4"

BRITISH STANDARD SCHEDULE 970 (EN SERIES)—*contd.*

En No.	Type of Steel.	Composition.						Properties.			
		C.	Mn.	Ni.	Cr.	Mo.	Others.	M.S.	E.	Imp'ct.	Limiting Size.
41	Cr.-Al.-Mo. steel for nitriding	0·18/0·45	<0·65	<0·4	1·4/1·8	0·1/0·25	Al. 0·9/1·3	P35/45 R45/55 T55/65	24 20 17	5 40 35	— — —
42	Spring steel (O.H.)	0·75/0·9	0·35/0·7	—	—	—	—		Brin. 341/429 when H.T.		
43	Spring steel (W.H.)	0·45/0·65	0·5/1·0	—	—	—	—		Brin. 341/429 when H.T.		
44	Spring steel (O.H.)	0·9/1·2	0·45/0·75	—	—	—	—		Brin. 341/429 when H.T.		
45	Si.-Mn. spring steel (O.H.)	0·5/0·6	0·7/1·0	—	—	—	Si. 1·5/2·0		Brin. 341/429 when H.T.		
46	Si.-Mn. spring steel (W.H.)	0·33/0·5	0·6/1·0	—	—	—	Si. 1·5/2·0		Brin. 341/429 when H.T.		
47	Cr.-V. spring steel (O.H.)	0·45/0·55	0·5/0·8	—	0·8/1·2	—	V. 0·15		Brin. 341/429 when H.T.		
48	Cr. spring steel (O.H.)	0·45/0·55	0·5/0·8	—	1·0/1·4	—	—		Brin. 341/429 when H.T.		
49	Valve spg. wire (C.D.)	0·7/0·8	<1·0	—	—	—	—	95/120	—	—	—
50	Valve spg. wire (O.H.)	0·4/0·5	0·5/0·7	—	1·0/1·5	—	V. 0·15	90/110	—	—	—
51	Ni. valve steel .	0·25/0·35	0·35/0·75	2·75/3·5	<0·3	—	—	Brin. <229		40	—
52	Si.-Cr. valve steel	0·4/0·5	0·3/0·6	<0·5	7·5/9·5	—	Si. 3·0/3·75	Brin. 255/293		—	—

BRITISH STANDARD SCHEDULE 970 (EN SERIES)—*contd.*

En No.	Type of Steel.	Composition. C.	Mn.	Ni.	Cr.	Mo.	Others.	Properties. M.S.	E.	Imp'ct.	Limiting Size.
53	Si.-Cr. valve steel	0·55/0·65	0·3/0·6	<0·5	5·75/6·75	—	Si. 1·4/1·7	Brin. 235/285		12	—
54	Ni.-Cr.-W. valve steel	0·35/0·5	<1·5	10	12/16	—	Si. 1·0/2·5 W. 2·0/4·0	Brin. <302		15	—
55	Cr.-Ni.-W. valve steel	0·18/0·45	<1·0	6/12	17	—	Si. 1·0/2·5 W. 2·0/4·0	Brin. <302		20	—
56	Cr. rust resisting steel	0·15/0·35	<1·0	<1·0	12	—	Si. 0·1/1·0				
56M	Do. (free machining)	0·15/0·35	<1·0	<1·0	12	<0·6b	Si. 0·1/1·0 S. <0·75 Se. <0·6b Pb. <0·45b Zr. <0·6b	45	20	25 20	Up to 2" Above 2"
56A	Cr. rust resisting steel (low carbon)	<0·15	<1·0	<1·0	12	—	Si. 0·1/1·0				
56AI	Do. (for turbine blading)	<0·12	<1·0	<1·0	12	—	Si. 0·1/1·0	35/45	25	45 25	Up to 2" Above 2"
56AM	Do. (free (machining)	<0·15	<1·0	<1·0	12	<0·6b	Si. 0·1/1·0 S. <0·75 Se. <0·6b Pb. <0·35b Zr. <0·6b				
57	Cr.-Ni. rust resisting steel	<0·25	<1·0	1·0	15·5/20	—	Si. 0·1/1·0	53	15	20	—
58	Aust. Cr.-Ni. rust resisting steel	<0·2	<1·0	6/20	12·0	Opt.	W. Ti. Cu. V. Cb. Ta. Opt.	35	30	50	—
58A	Do.	<0·2	<1·0	8/10	16/20		Si. 0·2 min.				

BRITISH STANDARD SCHEDULE 970 (EN SERIES)—*contd.*

En No.	Type of Steel.	Composition.						Properties.			
		C.	Mn.	Ni.	Cr.	Mo.	Others.	M.S.	E.	Imp'ct.	Limiting Size.
100	Heat-treated low alloy steel bars for machining	0·35/0·45 (0·25/0·45 for MS of 45 and 50 tons)	1·2/1·5	0·5/1·0	0·3/0·6	0·15/0·25	Si. <0·6	R 45 S 50 T 55 U 60 V 65	22 20 18 17 16	40 40 40 35 35	6" 4" 2½" 1⅛" 1⅛"
101	Carbon manganese C.H. steel	0·12/0·18	1·25/1·75	—	—	—	—	40		20	40
102	Do. (free machining)	0·12/0·18	1·25/1·75	—	—	—	S.0·08/0·15 P. <0·06	40	18	30	
110	Low Ni.-Cr.-Mo. steel for bars and forgings	0·35/0·45	0·4/0·8	1·2/1·6	0·9/1·4	0·1/0·2	—	R 45 S 50 T 55 U 60 V 65 W 70	22 20 18 17 16 15	40 40 40 35 35 30	6" 4" 4" 2½" 1⅛" 1⅛"

SIGNS AND ABBREVIATIONS APPLYING TO THE ABOVE TABLES.

Figures represent minima unless otherwise indicated.

<	Not more than.
A.C.	Normalised (air cooled)
H.T.	Hardened and tempered.
R	Hot rolled.
C.H.	Case hardening.
CR	Cold rolled.
CD	Cold drawn.
OH	Oil hardened.
WH	Water hardened.
SRS	Stainless and rustless (non-corrodible) stee
Opt.	At the option of the manufacturer.
a	Reduce by 2 for 4 ins. and over.
b	Total of Mo. Se. Zr. and Pb. not to exceed 1%.
c	Depending on size.
d	12 SWG and thicker only.
e	If required.
f	Titanium content not less than five times carbon content.
g	On tubes not suitable for proof bend test.
h	Columbium content not less than ten times carbon content.
k	Square tubes only.
l	Welded.
†	Up to 2 ins. only.
‡	Over 2 ins.
§	Up to 2½ ins. only.

Age-Hardening of Steel.

Age-hardening or *Precipitation-hardening* is the gradual increase in hardness of certain heat-treated steels with time when allowed to stand at ordinary temperatures or, in some cases, at temperatures above atmospheric. Although increased hardness may result from this source in high carbon steels it is a phenomenon usually associated with steels of low carbon content, and has been attributed to the presence of oxygen, nitrogen, carbon and other elements such as phosphorus. It is assumed that supersaturated solid solutions are formed which undergo gradual breakdown with time with the precipitation of sub-microscopic particles of carbides, oxides, nitrides, etc. It has been shown that ageing effects can be minimised by drastic de-oxidation of molten steel. The term *quench-ageing* is applied to the increased hardness which results when quenched mild steels are allowed to stand. The increased hardness is accompanied by decreased ductility and lower resistance to impact and shock. In certain cases a disastrous lowering of the impact value may result. The effects of quench-ageing are least in thoroughly killed steels, particularly when the final deoxidation is carried out by means of aluminium. Even in normalised mild steels super-saturated solid solutions may be obtained, which undergo breakdown with time, and there is little doubt that the *blue-brittleness* of mild steel is associated with the precipitation of compounds previously held in a state of super-saturation.

Strain-ageing is the term applied to the gradual increase in hardness when steel is allowed to stand after being strained by cold deformation. Generally the effects of strain-ageing are less than those of quench-ageing, although the degree of ultimate embrittlement may be greater in the former than in the latter case. It has been shown that steels susceptible to quench-ageing are also susceptible to strain-ageing. The effects of strain-ageing are more marked in semi-deoxidised steel of the *rimming* steel type than in the fully deoxidised steel. The ultimate degree of deoxidation is an important factor in connection with the formation of *stretcher-strains* (rough surfaces) of cold-deformed sheets. Boiler-plate failures have been attributed, in part, to the effects of strain-ageing in areas around rivet holes.

The cold-deformation of structural steel, followed by reheating as in such operations as galvanising, may lead to a decided embrittlement. In such cases the most satisfactory procedure is a reheating to about 650° C. before galvanising, an operation often described as *stabilising*.

Fully-killed steels are sometimes described as *non-ageing*. In these steels the freedom from ageing effects are partly attributed to (*a*) smaller grain size and (*b*) greater freedom from oxygen.

In fine-grained alloy steels containing special elements, quench- and strain-ageing effects are much less than in carbon steels, although certain alloy steels, *e.g.* copper steels, may exhibit age-hardening properties to a very marked degree.

Quenching Liquids.

Liquid quenching mediums include water, brine, oils, and special liquids. Soft water, distilled if possible, is used for hardening ordinary carbon steels. Impurities in the water, such as grease or certain acids, are objectionable, as the former are liable to cause uneven hardness by insulating the steel locally with an oil film, and the acids produce brittleness. Hard water is very unsatisfactory because of the scale thrown down when its temperature is raised. Water to which soap or salt has been added is sometimes used to secure quenching rates respectively lower or higher than that of pure water. Cold sea-water or brine produces extreme rapidity of quenching and consequently maximum hardness. Pieces of complicated design cannot be quenched safely in this medium because of the shock.

Oil-quenching mediums are extensively used where extreme hardness is not required, and where freedom from quenching shock is necessary. These include mineral oils, paraffin, fish oil, whale oil, cotton-seed oil, linseed oil, lard oil, and special animal hydrocarbon oils.

Mineral oils are satisfactory where the quenching bath is kept cool by artificial means. If the oil is not cooled there occurs a gradual breaking down and thickening of the oil with heat, the formation of residue, and wide variation in the quenching rate. Paraffin is dangerous unless kept at a temperature well below its flash point.

Fish and whale oils have the disadvantage of offensive odours and are open to the same objections as the seed oils. Cotton-seed and linseed oils become gummy from oxidation. This increases the viscosity and seriously affects the quenching speed. Lard oil is unsatisfactory because of the tendency to become rancid.

Special oils usually obtained by distillation of wool grease are used extensively with satisfactory results. Artificial cooling of these oils is very necessary when handling large quantities of metal continuously.

(Millett.)

TEST RESULTS OBTAINED FROM HEAT-TREATED STEELS.

Abbreviations used: N = normalised at; WQ = water-quenched from; OQ = oil-quenched from; YP = stress at yield point, tons per sq. in.; MS = ultimate stress, tons per sq. in.; E % = elongation per cent. on 2 ins.; R/A% = reduction of area, %; I = impact test, foot-pounds; B = Brinell No.

All tests made on bars 1¼ in. diameter.

Type of Steel.	Analysis.							Heat Treatment.	Physical Properties.					
	C.	Si.	Mn.	S.	P.	Ni.	Cr.		YP.	US.	E%.	R/A%.	I.	B.
Dead mild case-hardening	0·06	0·01	0·25	0·01	0 05	—	—	N 920° C.	19·0	26·0	41·0	71·0	36	121
								WQ 920° C.	—	35·0	30·0	67·0	50	167
								OQ 920° C.	—	30·0	35·0	69·0	90	137
								WQ 920° C.; reheated, WQ 760° C.	—	33·0	33·0	68·0	—	—
Mild case-hardening	0·17	0·24	0·72	0·041	0·047	—	—	N 920° C.	20·0	31·0	38·0	64·0	80	137
								WQ 920° C.	—	46·0	22·0	51·0	24	223
								OQ 920° C.	—	36·0	31·0	65·0	92	170
								WQ 920° C.; reheated, WQ 760° C.	—	40·0	32·0	64·0	43	183
Mild carbon	0·26	0·30	0·75	0·044	0·05	0·15	—	N 900° C.	24·5	35·5	32·0	59·0	46	159
								WQ 900° C.	—	57·0	11·0	30·0	13	—
								OQ 900° C.	—	44·0	25·0	55·0	31	207
Medium carbon	0·45	0·32	0·78	0·02	0·025	—	—	N 870° C.	27·0	44·0	27·0	54·5	31	192
								WQ 870° C.	47·0	67·0	12·0	28·0	14	321
								OQ 870° C.	36·0	56·0	20·0	49·0	27	—

The figures given after water- or oil-quenching represent the hardened condition. Intermediate values, ranging from the values given for the normalised condition to the hardened condition, are obtained by tempering, as shown in the table on p. 1168, which gives the values for a 3½ % nickel steel.

CHISEL STEELS.

Class of Steel.	Analysis.							Heat-treatment.
	C.	Si.	Mn.	Cr.	W.	Ni.	V.	
Carbon . . .	0·85	0·15	0·40	—	—	—	—	W.Q. 780° C. T. 280° C.
Carbon-vanadium .	0·80	0·15	0·40	—	—	—	0·25	W.Q. 785° C. or O.Q. 820° C. T. 280° C.
Tungsten-vanadium	0·75	0·15	0·50	—	1·0	—	0·10	O.Q. 810° C. — T. 280° C.
Chrome-tungsten .	0·50	0·20	0·50	1·5	2·0	—	—	W.Q. 810° C. or O.Q. 850° C. T. 300° C.
Chrome-nickel .	0·40	0·25	0·50	0·75	—	3·0	—	O.Q. 800° C. — T. 250° C.
High-speed . .	0·65	0·20	0·20	3·50	14·0	—	0·50	As for cutting steels

TYPICAL COMPOSITIONS OF VARIOUS STEELS.

Type of Steel.	Analysis.								
	C.	Si.	S.	P.	Mn.	Ni.	Cr.	V.	W.
Boiler plates . .	0·22	0·32	0·028	0·03	0·52	—	—	—	—
40-ton forging . .	0·36	0·41	0·035	0·032	0·80	—	—	—	—
Axle steel . . .	0·28	0·28	0·030	0·026	0·90	—	—	—	—
Rails (railway) . .	0·48	0·36	0·038	0·036	0·72	—	—	—	—
,, ,, . .	0·59	0·32	0·052	0·023	0·75	—	—	—	—
Tyres, loco. . .	0·62	0·28	0·02	0·04	0·55	—	—	—	—
,, waggon . .	0·38	0·19	0·05	0·04	1·20	—	—	—	—
Turbine disc . .	0·30	0·20	0·05	0·034	0·83	—	—	—	—
High-tensile carbon steel . .	0·45	0·26	0·03	0·038	0·56	—	—	—	—
30-ton carbon steel .	0·32	0·30	0·048	0·042	0·68	—	—	—	—
40-ton ,, ,, .	0·51	0·33	0·040	0·036	0·59	—	—	—	—
Steel tubes for welding	0·20	0·42	0·039	0·027	0·51	—	—	—	—
Piano wire . . .	0·60	0·10	0·022	0·026	0·45	—	—	—	—
Plough steel (for wire ropes) . . .	0·85	0·14	0·01	0·01	0·60	—	—	—	—
Plough steel (for wire ropes) . . .	0·58	0·16	0·032	0·033	0·41	—	—	—	—
Steel rolls . . .	1·08	0·27	0·02	0·025	0·39	—	1·88	—	—
File steel . . .	0·90	0·10	0·01	0·02	0·30	—	—	—	—
,, ,, . . .	1·40	0·25	0·04	0·03	0·80	—	—	—	—
Carbon spring steel .	0·89	0·22	0·022	0·024	0·43	—	—	—	—
Silico ,, ,, .	0·50	1·72	0·038	0·033	0·68	—	—	—	—
Chrome ,, ,, .	0·61	0·40	0·030	0·029	0·79	—	—	—	—
Silico-chrome spring steel . . .	0·53	1·06	0·031	0·022	0·35	—	0·78	—	—
Chrome - vanadium spring steel . .	0·55	0·36	0·035	0·035	0·60	—	1 25	0·16	—
Silico-manganese spring steel . . .	0·50	1·95	0·04	0·04	0·70	—	—	—	—
Non-corrodible nickel steel . . .	0·32	0·38	0·026	0·021	0·26	26·7	—	—	—
Nickel-vanadium steel	0·24	0·51	0·035	0·0[illegible]1	0·70	3·5	—	0·15	—
Chrome - vanadium forging . . .	0·26	0·44	0·04	0·038	0·50	—	1·0	0·18	—
Ball-bearing steel .	1·12	0·015	0·017	0·019	0·19	—	0·25	—	—
,, ,, .	0·95	0·014	0·018	0·019	0·25	—	1·25	—	—
Chrome-molybdenum steel . . .	0·40	0·35	0·04	0·032	0·41	—	1·0	Mo 0·35	—
Nickel-chrome tube .	0·30	0·16	0·03	0·03	0·45	1·10	4·0	V	—
Exhaust valve steel .	0·37	0·10	0·03	0·02	0·35	—	2·5	0·32	16·53
,, ,, ,, .	0·78	0·31	0·03	0·02	0·20	—	2·24	1·03	19·75
Silico-chromium valve steel . . .	0·45	3·00	0·03	0·03	0·40	—	9·00	0·30	—
Heat-resisting valve steel	0·30	1·50	low	low	0·30	12·0	20·00	—	3·00
Cobalt - chromium valve steel . .	1·25	0·40	0.04	0·04	0·40	—	13·00	Co 5·00	Mo 0·70

MAGNET STEELS.

Type of Steel.	Analysis. C.	Si.	Mn.	W.	Cr.	Co.	Ni.	Al.	Mo.	Ti.	Cu.	Heat-treatment.	Magnetic Properties. B_r (gauss).	H_c (oersted)	B-H max.
Carbon .	0·90	0·20	0·35	—	—	—	—	—	—	—	—	W.Q. 760° C.; T. 200° C.	9,000	55	200,000
Tungsten .	0·65	0·20	0·40	6·0	0·5	—	—	—	—	—	—	O.Q. 850° C.; T. 200° C.	10,000	70	260,000
Chrome-tungsten	0·95	0·15	1·10	4·0	5·0	—	—	—	—	—	—	O.Q. 800° C.; T. 250° C.	10,000	65	280,000
Cobalt-tungsten	0·90	0·20	0·40	9·0	3·0	15·0	—	—	—	—	—	O.Q. 900° C.; T. 300° C.	9,000	160	450,000
Chromium	1·00	0·15	0·40	—	6·0	—	—	—	—	—	—	O.Q. 480° C.; T. 250° C.	9,000	70	250,000
Cobalt-chromium	0·90	0·20	0·40	—	9·0	5·0	—	—	0·5	—	—	A.C. from 1150° C.; reheat to 700° C. and A.C.; reheat to 1000° C.; cool to 300° C. and O.Q.	7,500	145	450,000
	1·00	0·20	0·30	—	9·0	15·0	—	—	1·0	—	—		8,200	185	600,000
	0·95	0·20	0·30	5·0	—	35·0	—	—	0·5	—	—	O.Q. 930° C. . . .	9,500	235	900,000
Nickel-aluminium *	0·08	0·60	0·30	—	—	—	27·0	12·5	—	0·25	3·50	A.C. 1200° C.; T. 650° C.	8,000	450	1,750,000
Nickel-aluminium-cobalt * .	0·05	0·10	0·20	—	—	25·0	11·5	8·0	—	—	5·50	A.C. 1250° C.; T. 575° C.	12,400	570	4,300,000

* Alloys of these types are also placed into service as castings; to get the best magnetic tests the rate of cooling after casting is then modified to suit the cross-sectional area.

Effect of Nickel in Steel.

In the annealed condition the average effect of nickel in amounts up to 8 per cent. may be stated as follows:—

Each 1 per cent. of nickel:—

Increases the elastic limit by 1·8 tons per sq. in.
" " tensile strength by 1·9 tons per sq. in.
" " reduction of area by 0·5 per cent.
Decreases the elongation by 1·0 per cent.

For nickel steel with up to 5 per cent. nickel each 1 per cent. of nickel lowers the Ar_1 critical point from 10°–20° C. below those of a corresponding straight carbon steel, and the Ac_3 point from 8°–14° C. The lower the carbon content the greater is the depression of both Ac_3 and Ar_1. With a nickel content of 32–33 per cent. the $Ar_{3,1}$ point is apparently depressed to absolute zero. The addition of nickel also reduces the eutectoid carbon ratio from ·9 per cent. for a straight carbon steel to about ·75 per cent. for a 3½ per cent. nickel steel, and about ·7 per cent. for a 5 per cent. nickel steel.

In the heat-treated condition, the tensile strength, yield point, and hardness are raised with a corresponding reduction in the ductility as compared with a carbon steel. Nickel tends to retard grain growth, thus allowing greater latitude in treating; and a nickel steel may be held above the critical range for longer periods of time without serious injury. On slow cooling, the tendency is to maintain a fine-grained pearlitic or sorbitic structure instead of a full lamellar pearlite.

Nickel further improves the endurance ratio, and this is marked in the heat-treated condition. Steel with a high percentage of nickel (austenitic nickel steels) have excellent corrosion-resisting properties and resistance to oxidation at elevated temperatures, and are practically non-magnetic.

(Bureau of Information on Nickel.)

Effect of Chromium in Steel.

Chromium differs from nickel in that it forms a double carbide, resulting in greater strength and hardness in a heat-treated steel. By the use of nickel and chromium, the nickel strengthens the ferrite matrix and the chromium strengthens the carbide constituent, giving superior physical properties than when either element is used alone. The presence of nickel and chromium ensures considerable hardening, with a rate of cooling through the critical range slower than that permissible in the case of a steel with a similar content of either element. This makes nickel-chrome steels especially suited for large sections requiring heat-treatment; giving them deep and uniform hardening power.* In carbon steel chromium raises fairly uniformly both the Ac_1 and Ac_3 points. In nickel steel, however, the effect is somewhat different. The Ac_1 point is uniformly raised, but according to available data the Ac_3 point, when the carbon is below ·3 per cent., seems to be actually lower than in a steel of equivalent nickel content, while with higher carbon the Ac_3 point is higher than in an equivalent steel.

Invar.†

Invar is a nickel iron alloy containing about 36 per cent. nickel, together with about 0·5 per cent. each of carbon and manganese, with metallurgically negligible quantities of sulphur, phosphorus and other elements. It is made either in the open-hearth furnace or by the crucible method. It melts sharply at 1425° C. Above some 200° C. to its melting point, invar may be considered to consist of a homogeneous solid solution of iron, nickel and carbon. Below 200° C. and at a temperature dependent on its history and exact composition it undergoes a reversible transformation of such a nature that for any sample the transformation may be incomplete. This condition of thermo-chemical instability gives rise to both slowly changing and quickly changing values of its physical properties—changes which are particularly manifested in the expansion.

Invar can be forged, rolled, turned, filed and drawn into wires, and it takes a beautiful polish, giving an excellent surface on which fine lines may be ruled. In general it should be worked slowly. It will withstand without spotting the corrosive action of water. Its density is about 8·0 grm. per cm.^{3}, its electrical resistivity is of the order of 80 microhm per cm., or about eight times that of pure iron, and its temperature coefficient of electrical resistance about 0·0012 per degree centigrade. It is ferromagnetic, but becomes paramagnetic in the neighbourhood of 165° C.

The mean coefficient of linear expansion between 0 and 40° C. is in the order of one millionth for the ordinary invar, and samples have been prepared with even small negative coefficients; the amounts of carbon and manganese present appear to exercise considerable influence on the expansion. Above 200° C. the expansion of invar is approximately that of Bessemer steel.

Invar is subject to changes in length due to 'after effects' following cooling from a high temperature, and to changes in length following even slight alterations in temperature. Invar also gradually elongates with time, forged and drawn material behaving somewhat differently in this respect, so that there is a determinable, seasonal correction to be applied to its length when used as a length standard. It also shows marked magnetostriction phenomena or changes of length accompanying changes in strong magnetic fields.

Mechanical properties: Tensile strength, 22·5–36 tons per sq. in.; elastic limit, 3·1–13·5 tons per sq. in.; elongation, 40–50 per cent.; reduction of area, 40–65 per cent.; Brinell hardness, 160; scleroscope hardness, 19; modulus of elasticity, 22,500,000 lbs. per sq. in.

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* See, however, p. 1184.

† Extract from Circular No. 58, Bureau of Standards.

Expansion of Nickel Steels (Guillaume).

Mean coefficients of linear expansion between 0° and t° C. Applicable between 0° and 38° C.

Per Cent. Nickel.	Mean Coefficient of Linear Expansion × 10⁶.
30·4	4·570 + 0·01194 t
31·4	3·395 + 0·00885 t
34·6	1·373 + 0·00237 t
35·6	0·877 + 0·00127 t
37·3	3·457 − 0·00647 t
34·8 + 1·5 Cr.	3·580 − 0·00132 t
35·7 + 1·7 Cr.	3·373 + 0·00165 t
36·4 + 0·9 Cr.	4·433 − 0·00392 t

Coefficient of Expansion of Steels at Elevated Temperatures.

Average Coefficient of Expansion on Heating, × 10^{-6}.

C.	Si.	Mn.	Cr.	W.	Ni.	V.	Co.	20° — 500° C.
0·10	0·10	0·40	—	—	—	—	—	11·5 — 14·5
0·40	0·15	0·50	—	—	—	—	—	11·5 — 15·5
1·00	0·10	0·40	—	—	—	—	—	11·5 — 12·5
0·30	0·20	0·60	—	—	3·6	—	—	12·0 — 15·0
0·30	0·20	0·50	1·25	—	3·5	—	—	11·5 — 12·5
1·05	0·15	0·40	1·50	—	—	—	—	11·5 — 14·5
0·65	0·20	0·20	3·50	14·0	—	0·5	—	11·0 — 12·5
0·40	3·50	0·50	9·00	—	—	—	—	11·0 — 13·0
1·00	0·40	0·40	13·00	—	—	—	6·0	11·0 — 12·5
0·30	0·20	0·35	13·00	—	—	—	—	18·0 — 20·0
0·10	0·25	0·40	14·00	—	—	—	—	18·0 — 22·0
0·15	0·40	0·60	18·00	—	2·0	—	—	11·0 — 12·0
0·15	0·50	0·50	18·00	—	9·0	—	—	17·0 — 19·0
0·10	0·50	0·40	12·00	—	12·0	—	—	17·0 — 18·0
0·40	2·50	0·75	14·00	5·0	28·0	—	—	16·0 — 17·5
0·35	0·50	0·55	22·00	3·5	12·0	—	—	15·0 — 18·0
0·20	1·00	0·60	28·00	—	20·0	—	—	15·0 — 16·5
0·20	0·50	0·50	25·00	—	12·0	—	—	18·0 — 19·0
0·20	1·00	0·60	9·00	—	20·0	—	—	29·0 — 36·0
1·00	0·30	12·00	—	—	—	—	—	32·0 — 36·0
0·20	0·50	0·50	8·00	—	18·0	1·0	—	34·0 — 36·0

Effect of Molybdenum in Steel.

As little as 0·25 per cent. molybdenum is an effective hardening agent when incorporated in alloy steels, and heat-treated steels containing molybdenum are more ductile than many similar alloy steels. It acts much like tungsten but is more potent, and is similar to chromium in the properties it imparts to steel, but much smaller quantities are required to produce a given result.

It is used in nitriding steels, and may be substituted for tungsten in high-speed steel. It is a valuable addition to rustless, heat-resisting and acid-resisting steels, and to grey iron castings and steel castings. It enters into alloy steel guns and armour plate, saw steels, die steel, razor blades.

Mo reduces mass effect.

Temper-brittleness.—Certain steels, notably Ni-Cr steels, when hardened and tempered exhibit abnormally low resistance to impact or shock. When hardened nickel-chromium steels are tempered the impact test—tempering temperature curves show a minimum at some temperature between 250° and 500° C. For this reason these steels are never tempered within this range. Even when tempered at higher temperatures and *when slowly cooled from the tempering temperature* low impact figures may be obtained, whereas when quenched after tempering the resistance to impact is enormously increased. This phenomenon, known as temper-brittleness, although by no means confined to nickel-chromium steels is illustrated by the following test results:—

Composition: C, 0·33; Cr, 0·70; Ni, 3·60.

Heat-treatment: O.Q. 830° C. — T. 625° C.

Method of cooling after tempering.	Y.P.	M.S.	Elong.%	R.A.%	Impact.
Oil-quenched	55·0	60·5	22·0	61·0	67
Furnace	54·0	59·5	23·5	61·0	15

Molybdenum reduces the susceptibility of nickel-chromium steels to temper-brittleness to a minimum and it is chiefly on this account that from 0·30 to 0·70 per cent. of the element is introduced. Slow cooling from the tempering temperature then has little influence on the impact values.

Manganese-Molybdenum Alloy Steels.

Properties intermediate between the carbon steels and the high-class alloy steels. They are suitable for constructional steels, but their use is confined to small sections, as complete hardening in larger sections is difficult. These steels are almost free from temper brittleness and consistent impact figures can be obtained if slow cooled from the tempering temperature.

Produced in two grades, (*a*) carbon 0·15–0·20, (*b*) carbon 0·25–0·40; manganese in each case 1·3–1·7, and molybdenum 0·2–0·4.

Molybdenum additions to the normal carbon-manganese steel show a progressive increase in physical properties in direct ratio to the molybdenum content. With C 0·3, Mn 1·75, Mo 0·18, oil hardened at 850° and tempered between 550–650° C., a U.T.S. of 60 tons per sq. in. may be obtained. With C 0·15, Mn 1·6, Mo 0·25, oil hardened and tempered at 650° C., the tensile strength is 45–50 tons per sq. in. Parts that cannot be quenched may be normalised at 850° C., and tempered at 600° C.

Forging and stamping practice follows that of carbon steel, except that after these operations the parts can be thrown on the floor without any precautions being taken against cracking. Machining can be carried out at speeds approaching that of carbon steel. Comparative test values are indicated in the following table :—

Composition.						Heat-treatment.	Mechanical Properties on Bars 1⅛-in. Diameter				
C.	Si.	Mn.	S.	P.	Mo.		M.S.	Y.P.	Elong. %	R.A. %	Impact.
							tons per sq. in.	tons per sq. in.			ft. lbs.
0·30	0·25	1·35	0·035	0·030	—	N. 840° C. .	42·8	26·8	29·0	57	44
						O.Q. 850° C.; T^{d} 550° C.	48·4	34·4	26·0	60	41
						O.Q. 850° C.; T^{d} 600° C.	46·6	33·0	27·0	64	52
						O.Q. 850° C.; T^{d} 650° C.	44·6	31·1	27·5	66	78
0·35	0·25	1·35	0·035	0·035	0·30	N. 830/850° C.	42/50	30/40	26/20	60/45	35/20
						O.Q. 840° C.; T^{d} 500° C.	64·3	57·6	16·5	52	36
						O.Q. 840° C.; T^{d} 550° C.	59·9	54·7	18·0	52	51
						O.Q. 840° C.; T^{d} 650° C.	53·2	45·4	23·5	62	64

Non-Shrinking Die Steels.

Most steels after hardening have a larger volume than when in the annealed or unhardened state. The difference in volume between hardened and unhardened high-carbon steels containing between 1·0 and 2·0 per cent. of manganese is almost negligible. These steels, albeit erroneously, are described as 'non-shrinking' (better described as 'non-deforming') steels, and are used extensively for the manufacture of precision tools, taps, dies, milling cutters, gauges, etc., where extreme accuracy to size and shape after hardening and a minimum grinding or polishing allowance is required. A certain amount of the manganese is often replaced by chromium, tungsten or

STRENGTH OF VARIOUS ALLOYS AT HIGH TEMPERATURES.

Material.	Tensile Strength (Tons per sq. in.). Temperature °C. 15	100	200	300	400	500	600	700	800	900	1000	Heat Treatment.	Composition. C.	Si.	Mn.	Cr.	Ni.	V.	W.
Carbon steel ·14C.	27·2	28·1	30·0	32·5	25·2	20·0	12·5	6·8	4·0	2·0	—	N	0·14	·03	·56	—	—	—	—
,, ,, ·30C.	36·6	33·7	40·7	43·6	40·7	28·6	15·5	8·1	5·5	3·8	2·7	N	0·32	·19	·53	—	—	—	—
,, ,, ·42C.	41·0	40·1	43·7	49·7	42·0	33·5	21·0	10·1	6·0	4·2	2·4	N	0·42	·20	·60	—	—	—	—
3 % nickel steel	50·7	48·5	48·4	47·7	47·0	29·0	18·7	11·5	6·5	4·0	2·5	OQ & T	0·32	·17	·62	—	3·07	—	—
Chrome-vanadium steel	61·0	62·6	59·6	55·6	56·3	32·6	24·8	11·6	6·7	4·7	3·1	OQ & T	0·29	·22	·59	1·14	·32	·18	—
Chrome-nickel steel	57·8	56·4	55·0	55·3	55·4	44·9	23·6	12·3	5·6	4·5	3·0	OQ & T	0·29	·27	·60	·89	3·41	—	—
14 % chromium steel	48·5	45·9	43·5	41·3	37·8	32·7	18·8	12·1	5·0	—	—	OQ & T	0·27	·26	·30	13·3	—	—	—
Silicon-chromium steel	68·0	68·5	63·0	58·2	57·0	45·3	24·5	10·1	4·9	2·8	—	OQ & T	0·60	3·98	·52	8·75	—	—	—
Nickel-chromium-tungsten steel	63·0	53·5	49·4	43·4	42·7	39·5	34·1	28·2	17·2	12·0	6·0	AC	0·40	·97	·41	13·65	10·31	—	3·53
(Hatfield.)																			
Naval brass	26·0	26·5	23·4	18·0	6·5	2·5	—	—	—	—	—	—	Cu 61; Zn 38; Sn 1*						
Manganese bronze	29·0	27·5	25·5	13·0	4·5	—	—	—	—	—	—	—	Cu 60; Zn 39; Mn 1*						
Monel metal	24·0	23·5	22·0	20·0	18·0	16·0	12·0	8·0	—	—	—	Cast	Cu 29; Ni 69*						
,, ,,	42·5	42·0	40·0	38·0	36·0	30·0	20·0	12·0	6·5	—	—	Hot-rolled							

(G. & J. Weir, Ltd.)

* Approximate compositions (see pp. 1216 and 1220).

vanadium. For softening or hardening, temperatures between 780° and 800° C. are employed and the hardening is carried out by quenching in oil. Typical compositions are given in the following table :—

C.	Si.	Mn.	Cr.	W.	V.
0·85—0·95	0·2—0·4	1·5—1·75	—	—	0·1—0·25
0·90—1·00	0·2—0·4	0·9—1·15	—	—	—
1·00	0·25	0·85	0·75	0·45	—
0·90	0·25	1·00	0·50	—	0·50
1·80—2·00	0·2—0·4	0·2 0·4	14·00	—	0·2—0·3

Extrusion Die Steels and Mandrels.

Steels for these purposes have to withstand particularly onerous duty at elevated temperatures. Typical co mpositions are given in the following table :—

	C.	Mn.	Cr.	Ni.	W.	Mo.	V.
Dies	0·35	0·50	1·50	—	7·00	0·35	0·50
Mandrels	0·30	0·25	1·75	4·50	6·50	—	0·30

Heat-treatment consists in quenching in oil or cooling in air from about 1,000° C. followed by tempering between 550° and 650° C.

High-Carbon High-Chromium Steel.

Typical analysis: carbon 2·1–2·6, manganese 0·2–0·35, silicon 0·1–0·25, sulphur 0·25 max., phosphorus 0·25 max., chromium 12·0–15·0.

Heat-treatment: forge at 900–1050° C., anneal, heat to 750° C., cool in oil or water, Brinell 240. Harden, preheat to 720–750°, then raise to 920–950°, quench in oil or still air; Brinell 600. Temper at 300° C. This steel is suitable for rolls for cold-rolling and for drawing dies for wire.

CASE-HARDENING.

The case-hardening of steel is dependent upon the fact that above certain temperatures the affinity of iron for carbon is so great that when iron or steel is heated at these temperatures in contact with suitably chosen carbonaceous materials ready absorption of carbon supervenes. The quantity of carbon that can be absorbed by iron or steel under the above conditions varies according to the following factors—

1. The composition of the iron or steel.
2. The temperature at which the process is conducted.
3. The length of time during which the process is continued.
4. The nature of the carburising material.

The Composition of the Iron or Steel.

Case-hardening, in the accepted sense, is rarely practised on steel containing more than 0·20 per cent. of carbon or 0·50 per cent. of manganese. The process is stated by Guillet to be hindered by the presence of nickel, silicon and aluminium, and hastened in the presence of manganese, tungsten, chromium, and molybdenum.

The Temperature at which the Process is Conducted.

The process is best conducted at temperatures above the critical range of the iron or steel treated. The usual temperatures for carburisation vary from 900° C. to 1000° C. The higher the temperature of carburisation the greater the rate of absorption of carbon and the deeper the effect of absorption.

The Length of Time during which the Process is Continued.

This factor can only be determined by experiment. It is generally considered that the best results are obtained when the carbon content of the surface layers of the case-hardened material are of eutectoid composition (about 0·90 per cent. C.).

The Nature of the Carburising Material.

The carburising material may be either solid, liquid, or gaseous. Solid materials are employed more generally than are liquid or gaseous materials.

Solid Case-hardening Materials.

For the purpose of obtaining deep 'cases' the articles to be hardened are packed in a suitably designed box with solid case-hardening substances or mixtures of substances, of which the following are examples :—

Charcoal (animal or wood).
Charred leather.
Crushed bone or horn.
Barium carbonate 40 per cent. and charcoal 60 per cent. (Caron's mixture).
Charcoal (wood) 95 per cent. and soda ash 5 per cent.
Charcoal (wood) 90 per cent. and common salt 10 per cent. (Guillet).

As case-hardening composition is bought by weight, and is required to fill up the case-hardening boxes, the specific gravity is an important consideration. The speed of conduction of heat through various depths of carburising material requires determining by experiment. A good case-hardening material should not diminish in volume appreciably during the carburising operation, and should be free from sulphur and phosphorus.

The box should be of plate iron not less than ⅛ to ¼ inch thick. The lid of the box has two holes pierced in it fo drawing testing-pieces out if required. For small articles a piece of wrought-iron tube, plugged or capped at one end, may be used; the other end is closed by a second plug fastened with an iron pin passing through it and the pipe, the whole being luted with fine clay. The articles being previously finished, except polishing, are put into the box in alternate layers with the carburising mixture, commencing at the bottom of the box with carburising mixture to the thickness of about ¾ inch; upon this a layer of the articles is placed, then another of carburising mixture about one-third the thickness of the first, and so on until the box is nearly full, finishing with carburising mixture about the thickness of the first layer, leaving room every way for the expansion of the articles by the heat, otherwise they will bend each other in the box. The packing completed, the lid is put on, the box luted and placed in a suitable furnace. The contents of the box require to be very gradually and uniformly heated to redness, and retained at this heat for the period required for the depth of carburisation desired. The depth of carburisation depends on the length of time the temperature is maintained. For testing plain pieces of the same kind of steel as the articles may be used. They require to be brightened, and are placed, at the time of the packing of the box, in the central part, in such a manner that they may be readily pulled out through the holes in the lid, either by a piece of iron wire attached or by being made long enough to project through the holes, so that they may be gripped with pliers; the holes are luted the same as are the other parts.

When the case-hardening is required to terminate at any particular part of the article, the part needed to be soft may be coated with copper, either electrolytically deposited or by wiping the surface with an acid solution of copper sulphate. This prevents the iron from absorbing carbon at that part. A frequent procedure is to carburise an object and allow it to cool in the box, finally machining off the carburised portion before quenching wherever it is desired to keep the surface soft. It is of course necessary to leave sufficient metal on the object before carburising in order to ensure complete removal of all carburised metal before quenching. With long objects, one end of which requires hardening while the other is to remain soft, it is possible to carburise the entire length, then reheat and quench one end only, allowing the other end to cool in air.

For the purpose of superficial case-hardening it is necessary only to heat the iron or steel to be treated to redness, to sprinkle one or other of the following finely powdered compounds or mixtures on the surfaces to be hardened, and to plunge the iron so treated into pure cold water. This method is simple and speedy, but the case obtained is insufficiently thick to resist much wear.

Potassium cyanide.
Potassium ferrocyanide.
Sodium ferrocyanide 90 per cent. and anhydrous sodium carbonate 10 per cent.
Sodium ferrocyanide 80 per cent., anhydrous sodium carbonate 10 per cent., and potassium ferrocyanide 10 per cent.

Liquid Case-hardening Materials.

The immersion of steel articles in a bath of molten potassium cyanide, with sodium carbonate, heated to 900° C. results in the formation of very hard cases.

Gaseous Case-hardening Materials.

The following gases have been employed for case-hardening iron and steel—illuminating gas, carbon monoxide, acetylene and petroleum vapour. All, according to Brearley, are supposed to be more effective if they are led through a solution containing ammonia, or otherwise are partly saturated with ammonia gas before entering the furnace. By increasing the pressure of the gaseous mixture it is possible to increase the rate of carburisation.

Thermal Treatment of Case-hardened Materials.

Quenching.—If the quenching takes place directly after the carburising operation, the resulting product is liable to possess a coarse structure in both the core and the case. In order to improve the structure it is better to allow the work to cool in the box after cementation, and then reheat it in a separate operation. The reheat may be made bare in the furnace, but with large objects which take long periods to obtain the desired temperature it is advisable to protect them from decarburisation by covering them with charcoal or old case-hardening mixture. This second reheat is most satisfactory at temperatures of about 780° to 810° C. It is claimed that very good results may be obtained by reheating twice, first with a view to refining the core and secondly to refine the case. A sample treatment of this description is:—

1. Carburise at 920° C.
2. Reheat to 880° C. and quench in water.
3. Reheat to 770° C. and quench in water.

In order to refine the cores of case-hardened articles it is advisable to heat the same to at least 875° C. (unless the carbon content of the core is above 0·20 per cent.) and quench. This treatment refines the core of the article but coarsens the case, hence it is necessary to reheat to a temperature somewhat above the critical range corresponding to the carbon content of the case (760° to 820° C.), subsequently quenching the article in water, or in oil.

Quenching in water results in a much more rapid cooling than oil quenching; the former is necessary for ball races, cams, and all parts which have to be subjected to hard wear and require extreme hardness (80 to 90 scleroscope). Oil quenching of correctly carburised mild steel results in a hardness of 50 to 70 scleroscope. Small objects cool more rapidly, and therefore become harder than large objects undergoing the same treatment.

Tempering of the quenched articles still further improves the properties of the same.

Rapid Case-Hardening Methods.

One of the best is to instal a container for equal parts of sodium cyanide and sodium carbonate. These salts used molten are very efficacious, and many small parts can be thus cased without going to the trouble and expense of casing in boxes. Sodium cyanide needs careful handling, but it is a simple matter to arrange for it to be placed into the equivalent of a fume chamber so that an adequate draught will take away any poisonous fumes.

The immersion of clean machined steel parts in the mixture will produce a reasonably deep case in the course of ½ to 1 hour, and there is the additional advantage that the parts emerge from the bath quite clean and bright and can be quenched for hardening at once. Some trouble may be experienced in obtaining a suitable container for these molten salts, but the use of one of the nickel-chromium alloys will give excellent results.

A good method of obtaining a hardened case of reasonable depth on the surfaces of such parts as gear teeth, cams, etc., is the '*Shorter' flame-hardening process.* This process is mechanically operated, the apparatus consisting essentially of an oxy-acetylene blowpipe flame with a water jet for the rapid quenching of the surface of the steel when it has been heated to a temperature somewhat above the lower critical, *i.e.* the pearlite-austenite, change-point. One development of the system is the Shorter-Double-Duro Process for the hardening of the pins and journals of crankshafts and other shafts in which the heating and quenching are consecutive operations. A more recent development of the Shorter Process is the hardening of long shafts, rollers, etc., carried out during rotation of the work piece and the progressive movement of heating and quenching units. For the best results from carbon steels, a carbon content of about 0·5 per cent. is needed, but it is claimed that the process is equally applicable to the usual nickel, nickel-chromium and nickel-chromium-molybdenum steels of the oil hardening types. (The Shorter Process may be applied to the local hardening of grey cast irons, provided that the combined carbon content is of the order of 0·6 per cent.)

The *Tocco* process is the latest method for case-hardening. In this method there is no carburisation, but the surface layers to be hardened are heated by a high-frequency electric current and quenched by flushing with water supplied through holes in the inductor blocks. The process takes only a few seconds and almost any desired depth of hardening can be obtained.

In both the Shorter and Tocco methods the steel is previously in the heat-treated condition.

'Nitriding,' or Nitrogen Case-Hardening.

The process is based on the fact that certain steels will absorb nitrogen, and such steels undergo a eutectoidal transformation at 580° C., which enables an extremely hard non-brittle surface to be produced. Steels containing 0·5–2·0 per cent. aluminium and 0·5–4·0 per cent. of an element, or combination of elements, having the properties of chromium are used. Ordinary carbon and alloy steels are not suitable.

Typical Analyses of Steel 'Nitralloy' (Central Alloy Steel Corporation, U.S.A.).

	Nitralloy 'G.'	Nitralloy 'H.'
Carbon	0·36	0·23
Manganese	0·51	0·51
Silicon	0·27	0·20
Aluminium	1·23	1·24
Chromium	1·49	1·58
Sulphur	0·01	0·011
Phosphorus	0·013	0·011
Molybdenum	0·18	0·20
Average Brinell No., annealed	186	157
Average Brinell No., heat-treated core	300	255

Physical Properties, Alloy 'G.' Oil-Quenched from 1650° F. (900° C.).

Tempered at °F.	Tempered at °C.	Yield Point. Tons per sq. in.	Ultimate Strength. Tons per sq. in.	Elong. %	R./A. %	Impact. Ft.-lbs.	Brinell No.
800	427	80·5	100·0	11·0	36·0	12	445
900	482	73·6	92·4	11·5	37·5	15	415
1,000	537	70·6	81·5	15·0	50·0	22	363
1,100	593	61·5	69·6	16·5	57·0	35	330
1,200	649	53·7	61·6	20·0	60·0	44	285
1,300	704	46·2	54·1	23·0	62·5	55	226
1,400	760	36·0	46·6	28·0	59·0	54	200
Annealed 1,450	788	30·8	42·4	30·0	67·5	32	186

The nitriding process is carried out in a furnace (preferably electric), the temperature of which must be closely controlled around 950° F. (510° C.). The parts are placed in a gas-tight box provided with inlet and outlet tubes for the circulation of ammonia gas. No packing material is used, individual layers being separated by nickel wire netting. The rate of flow of gas is regulated by a needle valve on the ammonia cylinder, and this must be kept fairly constant during the entire cycle. The process takes from 2 to 90 hours, depending upon depth of case required.

Parts that require to be kept soft can be protected by tinning, nickel or copper plating, or other means.

Hardness of Case.—Readings taken with Herbert pendulum and converted into Brinell numbers.

Depth in Inches.	0·000	0·005	0·010	0·015	0·020	0·030	0·035	0·040
Brinell No.	1002	999	810	516	434	378	351	351

Before the nitriding operation the parts should be heat-treated to impart the desired physical properties to the core. All strains set up by forging, etc., prior to nitriding must be thoroughly relieved by annealing at 1,000° F. (537° C.). This process is described as '*stabilising*.'

Relation between Hardness Scales.

Scleroscope No.	Brinell No. (3,000 kg.)	Rockwell or Avery 'C' scale Cone 150 kg.	Vickers or Firth Pyramid Diamond.
20	105	—	110
30	200	18	215
40	235	23	250
50	300	32	320
60	370	39	397
70	440	46	475
80	525	53	585
90	610	59	705
100	650	63	780
103	700	66	850

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Ratio of Tensile Strength to Brinell Number.

Material.	Ratio = Tensile strength. Tons per sq. in. ÷ Brinell No.
Heat-treated alloy steels. Brinell range, 250–400	0·21
Heat-treated carbon and alloy steels. Brinell below 250	0·215
Medium carbon steels, as rolled, annealed, or normalised	0·22
Mild steels, rolled, annealed, or normalised	0·23

Tensile strength, tons per sq. in. = Brinell No. × ratio.
The above values do not apply to severely cold-worked or to austenitic steels.

In the Brinell-ball hardness test corresponding hardness values are obtained only when the ratio L/D^2 is kept constant. For ferrous materials this value equals 30 (*e.g.* 3,000 kilograms with a 10 mm. ball), where L is the load in kg. and D is the diameter of the ball. For certain non-ferrous materials the following relationships are obtained.

Material.	$k = \frac{\text{M.S. (tons/sq. in.)}}{\text{Brinell No.}}$	$\frac{L}{D^2}$
Brasses containing less than 30 % zinc	0·20–0·30	10
α-β brass of 60/40 type	0·25–0·40	10
Monel metal	0·20–0·23	10
Copper-aluminium alloys containing less than 8% Al	0·20–0·35	10
Alloys rich in nickel, *e.g.* nichromes, etc.	0·30–0·40	10
Aluminium-copper alloys containing between 4% and 14% copper	0·15–0·22	5
Magnesium alloys	0·27–0·33	5
Lead-tin and white metal bearing alloys	0·17–0·21	5

Comparative Hardness of Cold-rolled Steel Strip Stock.

Grade.	Brinell No.	Erichsen. mm.	Sclero. No.
Hard	210	7·8	34·5
Half-hard	156	10·1	27·5
Medium soft	126	11·0	24·0
Dead soft	99	12·0	19·0
Special dead soft	94	12·6	18·0

General analysis: carbon 0·07–0·11, manganese 0·30–0·40, silicon 0·01–0·03, sulphur and phosphorus about 0·01.

Relative Wear of Various Metals on Steel and Cast Iron.

Metal.	On Steel. (0·95 C. Brinell 192.)	On Cast Iron. (Brinell 163.)
1. Austenitic steel	0·78	3·55
2. Ball-bearing steel	0·61	0·79
3. Oil hardening steel	0·59	0·88
4. Tool-steel	0·51	1·56
5. High-speed steel	0·41	0·90
6. Case-hardening steel	0·39	1·21
7. Nitralloy	0·32	0·71
8. Chromium plating	0·22	0·43
9. Tungsten carbide (Widia)	Could not be measured.	Could not be measured.

Composition.

No. 1. 13 per cent. Mn, 1·26 C.
„ 2. 1 per cent. C, 1·25 Cr.
„ 3. 0·85 per cent. C, 2·1 Mn.
„ 4. 1 per cent. C.
„ 5. 0·65 per cent. C, 4·0 Cr, 15·5 W.
„ 6. 0·4 per cent. C.

Annealing.

To Prevent Oxidation and Scaling during Annealing.

It is sometimes desirable that steel should be thoroughly annealed without any oxidation and scaling whatever taking place. A popular method of doing this is simply to pack the stampings in sand, but this method is not good, because, unless the sand be extremely fine, air will eventually percolate through the grains of the sand. The following method has been found to give first-grade results, and steel articles may be heated for many hours and come out with a perfectly bright surface. The packing mixture consists of finely powdered burnt lime containing a trace of powdered charcoal. A thick layer of lime is tightly rammed down in the bottom of a box, similar to those used for case-hardening purposes. The articles are placed in position on the lime and a further layer of lime is tightly rammed down on top of them, with a sprinkling of powdered charcoal on top of the lime. Several layers of articles may be packed in this way till the box is full, when it is covered with a lid and luted up with clay. Although this process is not so quickly performed as some, it gives excellent results.

If such articles as small die castings or brass or bronze extruded sections require to be annealed, the articles are dipped in a saturated solution of borax or boric acid in methylated spirit. The spirit evaporates at once, and leaves a sound protective coating of the solid which coagulates on heating and effectively resists the deleterious action of oxidising and other gases at temperatures up to some 750 deg. C., which is as high a temperature as is usually employed for the general heat treatment of non-ferrous metals and alloys.

The coating of the borax or boric acid is so extremely thin that there is usually no necessity to remove it after heating, but should this be necessary, hot water will suffice.

(*Met.*, '*Machinery.*')

The only real means of minimising oxidation and scale formation is by controlling the nature of the atmosphere within the reheating chamber (see p. 1160). It has been shown that the amount of scaling is greater the higher the sulphur dioxide and water vapour contents of the furnace atmosphere.

Non-Metallic Inclusions in Steel.

Particles of non-metallic matter may exist in steel due to several causes, viz. (i) occluded slag during tapping and casting; (ii) erosion of the nozzle, stopper, ladle lining and (with bottom poured ingots) of trumpet and runner bricks; (iii) oxides formed as the result of the final deoxidation of the molten steel with ferro-silicon, ferro-manganese, ferro-titanium, aluminium, etc. Of these (iii) is the most fruitful source of non-metallic inclusions and here is one reason why electric furnace steel which may be almost completely deoxidised in the furnace instead of in the ladle, is generally cleaner than that produced by most of the other processes.

It has been shown that failure, particularly by fatigue, may originate from non-metallic particles and some idea of the amount and distribution of the non-metallic matter in steel is thus of importance to the engineer. An attempt to place the degree of cleanness on a quantitative basis has been made by G. R. Bolsover, who devised the method known as the 'Fox' Inclusion Count. Hardened specimens of the steel are microscopically examined at a magnification of 130, and the particles in each field graded according to number and size. About 60 fields are examined for each steel and an average value thus computed. The method is approximate only, but certainly does serve to discriminate between clean and dirty steel.

CORROSION OF METALS.

Protection of Iron and Steel from Rust.

The types of coating applied to iron and steel for their protection may be classified as follows:
1. Adherent coatings. 2. Compound coatings.

1. Adherent Coatings.

These may be metallic or non-metallic in character. Of methods for the production of metallic coatings of this type the following are the most noteworthy examples:—

(*a*) *Galvanising.*—This is a process which has for its object the coating of the material with zinc. Two processes, known respectively as hot galvanising and electro-galvanising, are employed for attaining the desired object.

Hot galvanising consists in the immersion of the object to be protected from corrosion in molten zinc or spelter, the surface of which is largely protected from oxidation by a supernatant layer of ammonium chloride. Modifications of the simple process briefly described in the last sentence are known under the following names, Winiwarter's process, Kufler's process, Bedson's process, Reese's process, Porter's process, etc.

The amount of zinc on modern galvanised sheets varies from 1 to 2·5 oz. per square foot coated. The coating generally contains from 0·25 to 1·00 per cent. of lead and from 2 to 6 per cent. of iron.

The most obvious disadvantages of galvanising are two in number: first, the iron or steel coated has to be raised in temperature for the desired effect to be attained; second, a layer of weak zinc-iron alloy forms at some point intermediate between the true zinc layer and the iron or steel upon which it is designed to rest. This thin lamina of weak alloy, mainly the compound $FeZn_7$, may be insufficient in a majority of cases to cause trouble, but a serious reduction in tensile strength is frequently brought about as a result of the presence of this weak material in fine gauge wires and sheets.

Electro-galvanising (cold galvanising) consists in the deposition of zinc electrolytically upon the object to be protected from corrosion, a soluble zinc salt being employed as electrolyte. A suitable electrolyte contains zinc cyanide 12 to 14 oz., sodium or potassium cyanide, 7 oz., and caustic soda 3 oz., to 1 gallon of water. The anodes are of pure zinc and the electrolyte is regenerated by small periodic additions of sodium cyanide. The current density is between 2 and 4 amps. per sq. ft. of cathode surface, and the best deposits are obtained with a bath temperature of about 40° C.

After galvanising in this way the plated parts should be heated to between 100° and 200° C. to remove brittleness due to plating.

The electrolytic process leads to the formation of an intermediate lamina of only minute thickness. In this particular the coating produced by the electrolytic process is an improvement upon that obtaining as a result of hot galvanising. It is, however, porous, and in this respect is inferior to the zinc coating produced in the dipping process.

By means of electro-galvanising a wide range of articles may be coated with a film of zinc which, while ample to protect from oxidation, does not affect ductility or tensile strength. The fact that perfect cleanliness must be achieved before the adherence of zinc can take place prevents dirty patches on the article becoming coated. The main cause of failure with hot galvanised work is the covering up, through the surface tension of molten spelter, of dirty or scaly places, which blister or otherwise fail in a comparatively short time. A commercially protective coating is obtained with a deposit of 4 oz. per square yard of surface. The deposit is practically pure, lead being the chief adulterant, then only in a fraction per cent. Accurately machined parts can be assembled after coating, this overcoming a difficulty ever present in the past. Hot galvanised bolts must have the thread bared before the nut will fit. Cold galvanising coats the thread so that the configuration is not destroyed, and unless the nut is a very tight fit the zinc remains on the thread and thus protects it against corrosion.

Admiralty specifications require a deposit of 0·86 oz. of electro-deposited zinc per sq. ft. of surface of the articles treated.

(*b*) *Tin plating* is a process similar to hot galvanising, tin being the coating metal. The protection afforded to iron and steel by tin is less effective than is that of zinc.

(*c*) *Terne plating* consists of the coating of the material to be protected with an alloy of lead containing from 50 to 75 per cent. of tin (generally 70 per cent. tin).

(*d*) *Lohmanising*—in this process the metals to be protected are (i) pickled in an acid bath, (ii) dipped into a salt bath which deposits a metallic salt over the entire surface of the body, (iii) immersed in a bath of molten alloy of zinc, lead and tin heated to a temperature of from 500 to 600° C. The composition of the tin-lead-zinc alloy is varied to suit the material being treated.

(*e*) *Electro-plating*—the coating of iron and steel electrolytically with copper, nickel, etc., is well known. For nickel-plating, a suitable solution contains 40 oz. nickel sulphate, 2·5 oz. sodium chloride and 5 oz. boric acid to 1 gallon of water. The p_H value of the solution should be about 4·5. The anodes generally consist of oxidised nickel. Best deposits are obtained at a temperature of 25–35° C. with a current density of 8–15 amps. per sq. ft.

(*f*) *Cadmium plating* consists in the electro-deposition of cadmium from a soluble solution of one of its salts. A suitable solution contains cadmium cyanide 2 oz., sodium cyanide 5 oz., per gallon of water, and a current density of about 10 amps. per sq. ft. is required. As with other articles coated by electro-deposition a subsequent reheating to a temperature between 100° and 200° C. is advisable in order to reduce plating brittleness.

(*g*) *Copper 'clothing'*—the casting of copper round or the welding of copper to steel and the rolling of the resultant material to the required shape and dimensions has been practised, *e.g.* Monnot process, Willis process, etc.

(*h*) *Nickel 'clothing'*—processes similar to the Monnot and Willis processes have been successfully applied to the coating of iron and steel with nickel and cupro-nickel.

(*i*) *Clad steel.*—In this case a thin veneer of corrosion resisting material is inseparably bonded on to a thicker and cheaper backing iron or steel, *e.g.* austenitic 18–8 nickel chromium steel on mild steel. The thickness of the coating may be varied between 5 and 25 per cent. of the total thickness according to requirements. Two distinct processes, perfected and patented by F. F. Gordon, are employed for this purpose.

1. *Ingot casting method.*—Two identical stainless alloy sheets or slabs are fastened together along their edges by welding or mechanical means and with a separating layer between them of some refractory such as green chromic oxide or sodium silicate mixtures. The exposed surfaces are then covered with a layer of iron or special alloy or coated with iron or nickel, electrolytically deposited. The sandwich thus prepared is inserted in a suitable mould and mild steel cast around it so that a complete ingot of normal external appearance is obtained. This ingot is then

roll-cogged, slabbed and finally finished in a sheet mill to the requisite thickness. The edges are then pared away and two sheets of clad steel obtained.

2. *Slab method.*—In this method a single or multiple assembly of mild steel slab and alloy sheets may be made with a bonding layer of iron, nickel, cobalt or special alloy, either electrolytically deposited or as thin sheet. For 3-ply material, a stainless steel plate is placed on both sides of a mild steel slab, and each such assembly is separated with a coating of refractory material to prevent sticking during the subsequent heating and pressing. The heating of the composite slab is preferably conducted in an inert or controlled atmosphere furnace. After pressing the slab is roll-cogged and finished in a sheet mill.

These processes are suitable, not merely for the making of corrosion resisting sheets, but for the manufacture of compound strip composed of mild steel and Invar, etc.

(*j*) *Metallic spraying*—the application of liquid metal in finely subdivided form by a spraying machine has been accomplished by the 'Schoop process,' allows of the coating of iron or steel with an adherent layer of tin, lead, aluminium or zinc, the material being coated remaining meanwhile at the ordinary temperature. The material under treatment must be scrupulously clean if adhesion of the finely-divided spray is to be perfected. The Mellozing process is similar.

(*k*) Thin films of a suitable grease or oil may also be employed as a protection against corrosion, particularly when machine parts or tools are shipped or placed in storage. For this purpose lanoline is very satisfactory. Lard, sperm and olive oils gradually generate fatty acids which may give rise to corrosive attack. Vaseline is satisfactory over relatively short periods, but pitting may result when the steel is covered by a vaseline film for a very long time.

2. Compound Coatings.

The term 'compound' is applied here to such protective coatings as are produced by chemical action, the ultimate result of which is the formation of a chemical compound, such, for example, as magnetic oxide of iron (Fe_3O_4), iron-zinc (FeZn), etc. The compounds formed may be, as already noted, either chemical (in the general sense of the term) or intermetallic (in the specialised sense of the term).

Chemical Compounds.

Barff's process consists in passing superheated steam over the objects, which are heated in an air-tight retort to about 260° C.; for the formation of a more perfect protective coating higher temperatures are necessary (650° C.). Articles so treated are said to be 'barffed.' Relatively small objects only can be treated as above. The coating formed consists of the mixed oxides of iron (Fe_2O_3 and Fe_3O_4).

The *Bower-Barff* process is an improvement on the above, and consists in the heating of the objects to be protected to about 900° C., at the attainment of which temperature superheated steam is introduced to the same, which has the effect of producing a coating of the mixed oxide already referred to. This operation having continued for about twenty minutes, carbon monoxide is injected into the retort for a period of from fifteen to twenty-five minutes, during which time the entire reduction of the coating to the condition of magnetic oxide of iron is completed.

The *Gesner process* treats iron and steel at a moderately high temperature (550° C. to 650° C.) in a retort wherein may be introduced an atmosphere of steam. The preliminary heating occupies twenty minutes; the steam is then allowed to react with the oxide formed in the first operation for about half an hour, when naphtha is introduced and allowed to remain in contact with the material under treatment for about ten minutes. Finally, steam is again introduced. The whole sequence of operations occupies about one hour and twenty minutes.

The *Parker 'dry' process.*—Under a patent assigned to the Parker Rust-proof Company, of Detroit, the formation of an adherent coating of oxide may be obtained by first heating the objects to be treated to about 600° F. and then subjecting them to fumes of the acid metaphosphates of strontium, molybdenum, and tungsten.

The *Hans Renold process.*—Remove all grease or oil from the articles to be treated by suspending in a bath of hot lyco (11 oz. per gal.) for about three minutes, wash in cold running water for a few seconds, then suspend from cathode brass bars by steel hooks in an electrolytic bath (KON, 98 per cent. pure, 4 oz. per gal. of water), iron plates forming the anodes. Leave here for five minutes, and when a current of 25 amps. at 8-10 volts. is passed through the bath all foreign matter on the surface will be removed. Wash off the cyanide by means of hot water to facilitate drying.

Heat the articles to 700° C., then quench in cotton-seed oil by immersing in the main body of the quenching tank for seven or eight seconds before allowing to fall to the bottom.

Dry by heated sawdust and hot-air blast (dried and filtered by passing through calcium chloride and cotton wool).

Insufficient oxidation results in the finished bodies having a grey appearance, due to the atmosphere of the furnace containing too much gas. Blistering is due to overheating.

Villon's process consists in subjecting the iron or steel to be protected to the vapour of sulphur as a result of which treatment the monosulphide of iron is formed.

The *Bertand process* requires that the articles to be treated be thoroughly cleaned by sand-blasting or similar means before they are immersed in a bath, such as will lead to the deposition of copper upon the same as a result of reaction between the iron and a suitable electrolyte. The deposition may be achieved by the application of electricity or by more purely chemical means. After thorough washing the articles are heated in an oxidising atmosphere to a temperature of about 600° C., which treatment leads to the formation of a fairly adherent coating of magnetic oxide of iron.

The *De Meietin's process* depends upon the formation of a coating of magnetic oxide (Fe_3O_4) upon iron or steel which is placed as the positive pole in a bath of distilled water, maintained at a temperature of from 70° C. to 80° C., a current of electricity being passed through it. The oxide coating is adherent, provided the current density is not over great, and may be rubbed and polished. This method is of value in the treatment of such articles as cannot be submitted to any processes which might tend to temper or anneal the objects under treatment.

The *Parker 'wet' process* (patent) consists in the immersion of the iron or steel to be protected in a bath of 1½ per cent. solution of the acid metaphosphates of tungsten, molybdenum, or any of the metals of the third, fourth or fifth group of elements.

The *Coslett process* (*Coslettising*).—Cast iron or steel articles treated by this process are not only rendered rust-proof but improved in appearance.

The articles to be treated should be perfectly clean and free from tin, nickel, or grease.

The best results are obtained on a frosted surface, such as is left by sand-blasting or pickling. Sand-blasting is to be preferred, as any traces of acid render the treatment ineffectual.

The coslettising bath is prepared by heating in a cast-iron or welded steel tank a sufficient quantity of distilled water to easily cover the articles to be treated. When boiling add for every gallon of water in the bath (1) two ounces of phosphoric acid (syrupy sp. gr. 1·5); (2) one ounce of iron filings. This should be carried out in the open or under an uptake, as the reaction results in the formation of disagreeable fumes. When this reaction has ceased the bath is ready for use.

The articles should be placed in the bath as soon as possible after sand-blasting, and should be gently boiled for two hours.

A second tank should be prepared containing distilled water, which should be heated in time to wash the articles as taken from the coslettising bath.

At the expiration of the specified time the articles should be taken from the coslettising bath and placed directly into the hot water and thoroughly scrubbed with a stiff bristle brush.

They may then be dried off, and if heavy will dry quickly themselves, but if light should be placed in an externally heated oven.

As soon as dry and while warm they should be dipped in a mixture of lubricating oil and turps substitute, in the proportion of 2 to 1, and left to drain. When cold, surplus oil may be removed with clean waste or wiper.

To allow for evaporation of bath a mixture similar to that in the bath should be made up and added as required.

To keep the bath at the correct strength it is necessary to titrate a sample of the bath with $\frac{1}{10}$ normal sodium hydroxide. To make this test, 10 c.c. of the bath should be placed in a conical flask, a few drops of phenolphthalein solution being introduced as an indicator. The $\frac{1}{10}$ normal sodium hydroxide should be added from a burette, until a permanent pink tinge is given to the liquid in the flask. The amount of sodium hydroxide dropped into the flask should be noted, and if the bath is correct this amount will be 10 c.c. If more sodium hydroxide is used, the bath is too strong and will require diluting with distilled water; if less is used, the bath is weak and will require the addition of phosphoric acid and iron filings in the given proportion.

The appearance of the deposit obtained on the article is a rough guide to the state of the bath. The proper deposit is of a uniform dull black colour, but if the bath is too strong this black coat is covered with a pearly white deposit which will not scrub off. If the bath is too weak the articles appear unevenly covered with a thin grey coat.

It is stated that in the present coslettising process (which cannot be used except under licence from the Coslett Anti-Rust Syndicate) phosphorised zinc, made under special conditions, is used instead of phosphorised iron, the solution otherwise being prepared in much the same manner. This newer process is said to give much better results, and the depth to which the articles are rust-proofed to be considerably greater.

The need for the absolute cleanliness of the articles before treatment cannot be too strongly emphasised, and they should not be handled after cleaning, or before they are finally oiled.

Intermetallic Compounds.

Sherardising.—The articles to be coated are first cleaned and dipped in an acid bath as for galvanising; they are then thoroughly washed with water, dried, and packed in zinc dust in a suitable retort which is preferably air-tight. The retort is heated to a temperature below 425° C. for a few hours. If the retort cannot be made air-tight about 3 per cent. of finely powdered charcoal should be mixed with the zinc dust to prevent the formation of zinc oxide. The coating consists mainly of the intermetallic compound $FeZn_3$.

Calorising.—A somewhat similar process to the above, in which aluminium powder is utilised as the agent and wherein the resultant protective coating consists chiefly of an intermetallic compound of iron and aluminium.

RELATIVE CORROSION. (Friend.)

Bars exposed to sea-water for 4 years, totally submerged for about 93 per cent. of the time, showed the following results :

Material.	Relative Corrosion
Stainless steel	54
Nickel chrome	69
Wrought iron	100
Cast iron	110
Carbon steel	126

Stainless Steel.

Many different kinds of stainless steel are now available having diverse chemical compositions. Chromium is an essential constituent of all of them, however, and a rough classification according to whether they contain very low or very high proportions of nickel is as follows :—

No.	C.	Cr.	Ni.	W.	Ti.	Mo.
1	0·10 max.	12·0–15·0	0–1·0	—	—	—
2	0·15 ,,	12·0–15·0	0–1·0	—	—	—
3	0·25–0·35	12·0–14·0	—	—	—	—
4	0·40–0·50	12·0–14·0	—	—	—	—
5	0·15	16·0–20·0	—	—	—	—
6	0·10	12·5	12·5	—	—	—
7	0·15 max.	18·0	8·0	0·5–1·0	—	—
8	0·15 ,,	18·0	8·0	0·5–1·0	0·5–1·0	—
9	0·07 ,,	18·0	8·0	—	—	2·5–4·0
10	0·07 ,,	18·0	10·0	—	—	1·0–1·5
11	0·15 ,,	15·0–16·0	10·0–11·0	—	—	—
12	0·20–0·30	10·0–14·0	35·0–37·0	—	—	—
13	0·25	20·0	9·0	—	1·3	—

(*W. H. Hatfield.*)

Typical mechanical and other physical properties are shown in the table on page 1197.

Steel No. 1 is described as 'stainless iron' and is a product of the basic-lined electric furnace. Owing to its greater softness it forges more easily than the harder varieties of stainless material; it works probably as easily as a 0·4 carbon steel, and hence may be forged, rolled, or drop-stamped. Stainless iron does not air-harden so intensively as the higher carbon varieties. It does air-harden, but the degree of hardness is considerably less, being in the range 280–380 Brinell instead of 450–550, as obtained with a stainless steel with 0·3 carbon.

It may be readily cold-worked, and provided the amount of distortion produced is not too great, the resistance to corrosion is not seriously affected. It can be drawn into wire and tubes. The general character of the resistance of stainless iron to various corroding media is similar to that of stainless steel. It is attacked readily by hydrochloric, sulphuric, and sulphurous acids. It is practically unaffected by saturated or superheated steam, lubricating oils, petrol, benzol, paraffin, greases, etc. It resists oxidation well at elevated temperatures. Above 700°–750° C. up to about 825° C. a polished surface becomes covered with a grey film without, however, losing its polished appearance, and the specimen neither gains nor loses weight appreciably. Above 825° C. it begins to scale appreciably.—(*Monypenny.*)

Steel No. 2 is described as mild stainless steel. No. 3 is ordinary stainless steel and is used for cutlery, certain kinds of tools, ships' propellers, etc. No. 4 is used for valves and No. 5 for purposes where superior resistance to corrosion is desired. All these steels are of the pearlitic type, *i.e.* they are magnetic and may be heat-treated in much the same way as ordinary steels but, of course, different treatment temperatures must be used.

Hardening.

For hardening they are quenched in water, oil or in air, according to thickness of section, from temperatures between 950° and 1,000° C. Thin sections such as knife-blades are effectively hardened by cooling in air. In the hardened state their structures are martensitic; they are therefore hard and magnetic, and in this condition offer maximum resistance to corrosion. Even when maximum hardness is desired, as for tools, cutlery, ball bearings, etc., the steel should be lightly tempered immediately after hardening, otherwise it may crack. Slow and uniform heating for tempering is essential. Tempering at higher temperatures lowers the hardness and tensile strength and is carried out when a combination of toughness, high shock-resistance, and corrosion-resistance is desired. The corrosion-resistance is adversely affected by tempering within the range 520°–580° C. The susceptibility or immunity to this kind of corrosion, often referred to as *line-corrosion*, may be revealed by immersion for about 48 hours in sodium chloride or common salt solution. This kind of corrosion is inhibited by increasing the chromium to 16–17 per cent.

No.	Condition.	Mechanical Properties. Yield Point. Tons per sq. in.	Max. Stress. Tons per sq. in.	El. % on 2 ins.	Red. of Area %	Young's Modulus of Elasticity, Tons per sq. in.	Brinell Hardness.	Physical Properties. Specific Gravity.	Coeff. of Thermal Expansion 20°–100°	Thermal Conductivity (C.G.S. Units).	Electrical Resistivity. Microhms per cm.³	Maximum Permeability (C.G.S. Units).
1, 2	Hardened and fully tempered or annealed .	15–25	30–40	30–40	50–60	13,400	140–180	7·73	0·0000107	0·046	52–57	500
3	Hardened and fully tempered	25–35	40–50	20–30	50–60	13,700	200–240	7·75	0·0000107	0·050	50–55	650
	Do. and lightly tempered .	—	—	—	—	—	450–550	7·74	—	—	65–70	75
4	Do. and lightly tempered .	—	—	—	—	—	550	7·74	—	—	—	—
5	Hardened and tempered .	35–50	50–60	15–25	40–60	13,500	240–280	7·70	0·0000104	0·040	72	210
6	Fully-softened	13–17	35–40	40–60	40–60	12,900	130–140	8·01	0·0000181	0·030–0·035	70	1·005–1·03
7	”	15–18	37–45	40–60	40–60	12,900	160–180	7·93	0·0000170	0·030–0·035	73	1·005–1·03
8	”	16–20	40–50	35–50	40–50	13,000	160–200	7·90	0·0000168	0·030–0·035	73	1·5–2·0
9	”	15–18	35–45	40–60	40–60	13,200	150–180	7·96	—	—	—	1·005–1·03
10	”	15–18	35–45	40–60	40–60	13,200	150–180	7·96	0·0000170	0·030–0·035	73	—
11	”	14–17	35–45	40–60	40–60	12,900	130–150	7·98	0·0000175	0·030–0·035	72	1·005–1·03
12	”	22–26	40–50	25–40	40–60	13,000	160–200	8–10	0·0000136	0·020–0·025	98	Slightly magnetic. BH = 3,500 for 150
13	”	26	45	30	55	13,000	230	7·88	0·0000150	0·030–0·035	82	4–10

(*W. H. Hatfield.*)

One of the valuable properties of stainless steel is its behaviour at high temperatures. When pieces of the material which have been polished are heated up gradually, temper colours appear on the surface similar to those obtained when ordinary carbon steels are heated up, but in the case of mild stainless steel these temper colours appear at much higher temperatures, as shown by the following table:—

Colour.	Stainless Steel.	Ordinary Steel.
	° C.	° C.
Pale straw	290	225
Straw	340	235
Brown purple . . .	390	265
Purple	450	280
Bright blue	530	290
Dark blue	600	315

Up to temperatures of about 825° C., ordinary stainless steel resists scaling to a remarkable degree, but above this temperature the material begins to scale to some extent.

Hot Rolling and Forging of Ordinary Stainless Steel.

The most suitable hot rolling temperature for this material is 1050° C. to 1100° C., and it is essential that for rolling and forging the material should be brought up slowly to about 900° C. and then slightly quicker to the rolling and forging temperature. It should be allowed to soak for a long period to ensure a uniform temperature through and through.

For forging, it should be worked quickly and with rapid blows between the temperatures of 1150° C. and 900° C. Below 900° C. it is not so easily deformed, and if any heavy work is put on the material after it has fallen below these temperatures it is very apt to be badly stressed or burst.

For drop stamping, it is advisable to work at a slightly higher temperature, and all stampings must be trimmed while hot, or annealed before trimming on account of the air hardening properties of the steel. Otherwise, if an attempt to trim is made, cracking and splitting will occur, as well as damage to tools.

Normalising.

In view of its marked air-hardening properties ordinary stainless steel cannot be normalised in the ordinary way and for softening must either be annealed or fully tempered.

Annealing.

To anneal ordinary stainless steel to obtain the maximum softness, it should be heated up slowly to about 1000° C. and held at this temperature for a period of time in proportion to the mass of metal, and then allowed to cool off slowly in the furnace.

Although steel annealed this way has the maximum degree of softness, it is not in the best condition for machining. The steel in this state is very soft and tough, and so drags and tears when machined, leaving a very rough surface.

To soften the material for good machining properties or cold drawing and cold rolling, it is best to anneal at 700°C. to 800° C., preferably about 750° C., and then to cool in the furnace or draw and cool in the open air.

Stainless steel has distinct air hardening properties, so that it should always be annealed or high temperature drawn at about 750° C. as stated above, after it has cooled down from a hot rolling or forging temperature, before it can be successfully machined or cold worked.

Cold Rolling and Drawing.

The material will cold roll, cold draw and deep draw, or press very well, but owing to the percentage of chromium in the metal, the depth of each draw, when the steel is being deep pressed or drawn, obtained at each operation is only about half that got when working on mild steel, before the material must be annealed.

In cold rolling and cold drawing, the reduction of one gauge at a pass can be used, but it is found necessary to anneal after every second or third pass.

As with all materials, a distorted surface is always more liable to rust, and the same applies to stainless steel, especially if the surface happens to be unevenly distorted.

In cold working, strains are always set up in the material, and sometimes strained parts are not confined to the surface where they can be ground off, but are inherent in the steel and the strains must be removed by heat treatment. These strains are not only liable to cause rust, but are apt to cause the steel to crack if put into service. It is thus evident that it is advisable to anneal or high temperature draw at 750° C. all material which has been cold-worked in any way.

Pickling.

The proper pickling of stainless steel is a very important factor in the manufacture of articles of this material. If the steel is left coated with scale, or has small pieces of scale left on the surface it will rust wherever the scale is present. It is therefore essential that the finished surface should be free from all scale, roakes, seams, pits, or cracks, or even stamp marks, as otherwise rusting will result.

The acid pickling baths generally used for this material are as follows:

1. 50 per cent. solution of hydrochloric acid used hot at 60° C. to 70° C.
2. 15 per cent. solution of sulphuric acid used hot at 60° C. to 70° C.
3. Ten minutes in a 50 per cent. commercial hydrochloric acid, then transfer without washing to 20 per cent. nitric acid for 25 minutes, both these solutions being worked cold.

Of these acid pickling solutions, No. 3 is the one which has given most satisfactory results. Other methods which have been used for cleaning this steel are electrolytic pickling in an alkaline bath, which has been more or less successful for bars, etc., and articles of simple shape. Acid pickles containing chemical salts to accelerate the action, have also been used with considerable success.

Grinding.

The grinding of stainless steel must be carried out very carefully, as this material is more difficult to grind than ordinary steel. This metal dissipates heat very slowly, and great care should be taken in grinding in order that grinders' 'scorch' is not produced. Grinders' 'scorch' is due to harsh grinding and produces a brownish discoloration on the steel. This discoloration is really a temper colour produced by the heat generated on the surface of the work, and will cause rusting, due to the local distortion set up. Wet grinding is much better than dry grinding.

Etching.

In etching ordinary stainless steel, the following solutions have given very good results:

1. A saturated solution of ferric chloride in hydrochloric acid to which a little nitric acid has been added. This solution is used full strength.
2. A mixture of hydrochloric acid and nitric acid in the proportion of three to one and used full strength after allowing to stand for about 24 hours.
3. A saturated solution of copper sulphate in hydrochloric acid. This solution is used for light etching.

Physical Properties.

The physical properties of mild stainless steel in the annealed condition, and also in the hardened and tempered state, cover a very wide range; this will be seen from the following table: —

Tests on Tempered Bars.

Tempered at.	Yield Point. Tons per sq. in.	Tensile Strength. Tons per sq. in.	Elongation. Per Cent.	Reduction of Area. Per Cent.	Izod Impact. Ft.-lbs.	Brinell Hardness. No.
° C.						
200	66·5	73·0	12·0	37·5	35	340
300	66·0	72·5	12·5	37·0	39	332
400	65·5	72·3	16·0	50·0	38	332
500	59·0	72·5	18·0	52·0	36	340
600	38·0	48·0	22·5	62·0	66	235
700	31·0	41·0	27·0	66·0	80	192
750	28·0	36·6	30·0	69·0	88	174

These tests were made on 'as rolled' and 'annealed' bars quenched from 950° C. in oil and tempered at various temperatures.

Hardness of Stainless Steel at Elevated Temperatures.

Temperature. ° C.	Brinell Hardness Number. A.	B.
15	235	440
100	212	420
200	190	414
300	160	345
400	130	360
500	110	345
600	100	175
700	85	89
800	70	70

A. Oil-quenched, 950° C.; water-quenched, 600° C.
B. Oil-quenched, 950° C.: tempered, 250° C. (*Hadfield.*)

Welding.

Ordinary stainless steels can be welded by either the electrical process or by the oxy-acetylene method. It will not weld by the usual method of heating in a smiths' fire.

In welding, the temperature of the metal at the weld is raised to a fairly high temperature, so that the material on cooling down will air harden and should be annealed before any turning, grinding, or polishing is done. (*H. S. Primrose.*)

Austenitic Stainless Steels.

Steels numbered 7–13, p. 1196, are described as austenitic nickel-chromium steels. Unlike the ordinary stainless steels they offer their best resistance to corrosion when in the softest state. In this condition they are completely austenitic and non-magnetic, and are softened by rapid cooling or quenching from above 1,000° C. In all respects but resistance to corrosion their properties are entirely different from those of the pearlitic plain chromium type. The austenitic stainless steels cannot be hardened by heat-treatment but only by cold-work. They are readily rolled into sheets and can be deeply pressed, and drawn into wire and tubes. After cold-working softness, ductility, machinability, and maximum resistance to corrosion are restored by reheating, slowly at first and then more rapidly, to 1,100° C., followed by cooling in air.

For forgings the steel is heated to 1,100°–1,150° C. and worked down to about 900° C. Typical physical properties are shown on p. 1157, and the relatively low yield-point values will be noticed. These are improved by slightly increasing the carbon contents and by increasing very considerably the nickel or chromium as indicated by steels 12 and 13.

These steels are often used for castings. They resist oxidation and retain their strengths quite well at elevated temperatures.

'Weld-Decay.'

Austenitic nickel-chromium steels, if reheated to temperatures between 600° and 900° C., are susceptible to inter-crystalline attack by corrosive media, a form of corrosion known as weld-decay, since it was first observed in welded articles in zones which had been reheated within the above range. The susceptibility to weld-decay can be overcome by reheating to the softening temperature (1,100° C.) and cooling in air, or by modified composition. Thus carbon is kept as low as possible, chromium and nickel increased, and small proportions of tungsten, molybdenum, titanium or columbium (niobium) added,* as indicated on p. 1196. Such steels are said to be entirely free from weld-decay. The standard Air Ministry test for weld-decay consists in boiling suitable specimens for 72 hours in the following solution:—

111 grams copper sulphate crystals.
98 grams (about 54 ml.) concentrated sulphuric acid.
Water up to 1 litre.

The specimens are then cleaned and subjected to a bend test. Inferior specimens are brittle and crack, whilst satisfactory samples can be bent double without the slightest sign of fracture.

Austenitic stainless steel is readily welded, but as indicated above, the welded articles must be normalised or else a weld-decay-free steel used. A suitable welding rod has the following composition:—

0·25–0·55 per cent. C; 15–19 per cent. Cr; 24–28 per cent. Ni; 2·0–3·5 per cent. Si; 0·6–1·25 per cent. Mn.

Selenium Rustless Steels.

Austenitic steels, average analysis: C 0·1, Cr 18·0, Ni 9·0, Se 0·25. The outstanding effect of selenium is to render the steels free cutting, replacing sulphur in this respect. They can be machined at about 60 to 70 per cent. of the speed used for ordinary screw stock. Selenium has a slight beneficial effect on tensile strength at elevated temperatures, also increases resistance of steel to the action of boiling solutions of acetic acid and aluminium sulphate.

Physical Properties of Stainless and Selenium Steel at Elevated Temperatures.

(*a*) Stainless steel: C 0·11, Mn 0·29, Si 0·20, Cr 17·91, Ni 8·06.
(*b*) Selenium steel: C 0·08, Mn 0·39, Si 0·48, Cr 17·65, Ni 8·45, Se 0·23.

Temperature. ° C.	Steel.	Elastic Limit. Tons per sq. in.	Tensile Strength. Tons per sq. in.	Elong. Per cent.	Reduction of area.
20	a	24·5	52·0	42·2	64·2
	b	23·4	49·6	41·6	65·0
425	a	17·3	35·6	42·8	64·4
	b	17·6	37·5	27·4	61·9
480	a	15·8	33·6	41·0	64·8
	b	16·3	36·0	24·5	46·6
540	a	14·5	31·7	37·3	65·1
	b	14·0	33·3	21·0	57·4

* When columbium is introduced to prevent 'weld-decay,' the columbium content must be at least ten times that of the carbon content of the steel.

Up to 3·5 per cent. of copper may be added to austenitic nickel-chromium steels; its addition improves the resistance to attack by dilute hydrochloric acid and chloride solutions.

Molybdenum-bearing stainless steels of the 18/12 type containing about 3 per cent. Mo are particularly useful for resistance to attack by sulphuric acid and sulphate solutions.

Heat-Resisting Steels.

Although the austenitic nickel-chromium steels previously mentioned possess excellent resistance to corrosion and oxidation, even at elevated temperatures, the modern demand for materials possessing greater strengths and at the same time possessing great resistance to oxidation and scaling by heat at high temperatures has led to the evolution of the so-called heat-resisting alloys. The compositions and physical properties of some of these steels are given in the following tables:—

No.	C.	Cr.	Ni.	W.	Ti.	Si.
14	0·4–0·5	6·0–10·0	—	—	—	3·0–4·0
15	0·1–0·5	12–14	—	—	—	—
16	0·1–0·2	25–30	0–5	—	—	1·0–2·0
17	0·5 max.	10–15	10–15	2–3	—	1·0–2·0
18	0·5 ,,	12–16	25–35	0–4	—	1·0–2·5
19	0·15 ,,	18	8	0·5–1	0·5–1	—
20	0·2–0·4	20–25	20–25	—	—	1·0–2·0
21	1·0 max.	15–25	60–80	0·4	—	1·0

No.	Condition.	Mechanical Properties. Yield Point. Tons per sq. in.	U.T.S. Tons per sq. in.	El. % on 2 ins.	Red. of Area. %	Brinell Hardness.	Physical Properties. Specific Gravity.	Coeff. of Thermal Expansion 20°–100°	Thermal Conductivity (C.G.S. Units).	Electrical Resistivity. Microhms per cm.³
14	Hardened and tempered	45–50	65–70	15–20	35–45	250–285	7·60	0·0000130	0·030	75–85
15	Hardened and tempered	15–40	30–60	20–40	40–60	140–280	7·74	0·0000106	0·045	52–65
16	Softened	20–35	30–45	15–25	30–50	150–225	7·50–7·90	0·0000104	0·030	80–86
17	Softened	30–45	50–65	20–35	30–50	220–290	8·00	0·0000169	0·033	78
18	Softened	30–40	50–60	20–30	40–50	180–270	8·00	0·0000151	0·030	85
19	Softened	16–20	40–50	35–50	40–50	160–200	7·90	0·0000168	0·030–0·035	73
20	Softened	22–35	40–55	30–50	40–60	180–260	7·90	0·0000154	0·030	92
21	Softened	35–45	50–60	15–25	25–35	180–250	8·10	0·0000121	0·020	105

(*W. H. Hatfield.*)

Steel No. 14 is used for high-duty exhaust valves and for purposes where non-scaling properties are required. Its impact resistance at ordinary temperatures is relatively low, so that it is essentially a steel suitable for high-temperature work only. This steel also exhibits a marked resistance to attack by leaded fuels.

Steels 15 to 20 are also used for exhaust valves and for baffle plates, oil or gas burners, enamelling trays, etc.

Alloy No. 21 is essentially a non-ferrous alloy and is particularly suitable where considerable strength and great resistance to scaling at high temperatures are required. It has found considerable application in the manufacture of annealing and case-hardening boxes, baffle plates, furnace muffles and doors, pyrometer sheaths, dampers, etc.

The above alloys are characterised by their relatively high maximum stress and limiting creep stress values at elevated temperatures.

Temp. °C.	0·17 % C. Steel.		0·23 % C. Steel.		C 0·15–0·30* Cr 18·0–22·0 Ni 6·0–8·0 W 2·0–2·5		C 0·35–0·45* Si 2·00–3·00 Mn 1·00–1·50 Cr 18·00–22·00 Ni 24·00–26·00		C 0·35–0·45* Si 1·50–2·50 Mn 1·00–1·50 Cr 14·00–16·00 Ni 25·00–30·00		C 0·40–0·50* Si 1·20–1·50 Mn 0·50–0·60 Cr 12·00–14·00 Ni 10·00–12·00 W 2·00–2·50		C 1·0 max.* Si 0·5–1·0 Cr 13·0–23·0 Ni 63·0–65·0 (Cast alloys)	
	Ultimate Tensile Strength.	Limiting Creep Stress.	Ultimate Tensile Strength.	Limiting Creep Stress.	Ultimate Tensile Strength.	Limiting Creep Stress.†	Ultimate Tensile Strength.	Limiting Creep Stress.	Ultimate Tensile Strength.	Limiting Creep Stress.	Ultimate Tensile Strength.	Limiting Creep Stress.	Ultimate Tensile Strength.	Limiting Creep Stress.
20	—	—	—	—	60	—	26	—	—	—	64	—	—	—
500	19·4	4·8	—	—	—	—	—	14·0	40·0	—	—	—	—	—
530	—	—	16·4	4·0	—	—	—	—	—	—	—	—	—	—
550	15·0	2·4	—	—	36	—	19	—	—	—	39	13·5	—	—
600	11·0	1·2	—	—	33	—	16·4	5·0	33·0	13·5	35·5	9·5	39	—
700	7·0	—	—	—	24	2·5	15	1·5	24·0	7·0	26·0	—	27·5	4·0
725	—	—	—	—	—	—	—	—	—	—	—	2·0	—	—
800	3·5	—	—	—	14	1·25	7·5	—	17·0	2·0	14·5	—	23	—
900	2·0	—	—	—	—	—	—	—	—	—	—	—	12	1·75
1000	—	—	—	—	—	—	—	—	—	—	—	—	7·5	—

Aluminium up to 4·0 per cent. is sometimes added to heat-resisting steels. Like silicon and molybdenum it improves the scaling resistance at high temperatures.

Heat-resisting steels of the above types should be forged between 1,150° and 900° C.

Copper Steels.

The beneficial influence of small proportions of copper on the corrosion-resisting properties of mild steel has been known for a long time and it is on this account that the element is added to tube steels.

The addition of 0·25–0·50 per cent. Cu raises the yield point and maximum stress without adversely affecting the ductility and working properties. The following are average comparative values for 0·25 per cent. carbon-steel in the form of hot-rolled sheet.

Copper Content. Per cent.	U.T.S. Tons/sq. in.	Yield Point. Tons/sq. in.	Elongation. Per cent.
0·015	30·8	16·6	30·5
0·2	31·5	18·4	29·75
0·45	32·4	19·9	27·4

(*Iron and Steel Industrial Research Council.*)

0·25 per cent. of copper and 0·25 per cent. molybdenum considerably improves the limiting creep stress of mild steel at temperatures up to 450° C.

Steel containing 0·25–0·50 per cent. copper and 1·0 per cent. chromium is used as a high-tensile, corrosion-resisting structural steel. Its tensile strength is about 40 tons per sq. in. with elongation of 20 per cent.

The addition of more than 1·0 per cent. of copper to dead-mild steels, suitable for pressings, etc., confers 'temper-hardening' properties upon them. In the normalised condition their

* After normalising from 850°–900° C.

† Limiting creep stress values based on a rate of 10^{-6} ins. per in. per hour.

properties are not much different from those of normalised copper-free steels, but a considerable increase in strength results from tempering between 450° and 600° C. Smith and Palmer have reported the following data on a mild steel containing 0·08 per cent. carbon and 1·50 per cent. copper.

	Maximum stress. Tons/sq. in.	Elongation. Per cent.	Brinell Number.
Normalised . . .	33·5	33	160
Normalised and tempered	45·0	26	215

(*Smith and Palmer.*)

Up to 3 per cent. copper is often added to austenitic stainless steel of the 18/8 type to further improve corrosion resistance.

A high-carbon chromium-copper steel has been developed by the Ford Motor Company for cast and heat-treated crankshafts. This steel contains 1·35–1·60 per cent. carbon ; 0·85–1·10 per cent. silicon ; 0·6–0·8 per cent. manganese ; 0·4–0·5 per cent. chromium and 1·5–2·0 per cent. copper.

Effect of Molten White Metal on Properties of Steel.

When steel is stressed in contact with liquid non-ferrous metals and alloys its ductility may be seriously affected owing to inter-crystalline penetration. The following table, due to Swinden, shows the effect of white metal on the ductility of several steels.

Loss of Ductility under Stress at 250° C. in Molten White Metal.

Type of Steel. Per cent.	Condition. ° C.	U.T.S. Tons per sq. in.	Percentage Reduction in Elongation at Fracture.
0·3 C, 3·25 Cr, 0·6 Mo .	O.H. 900/T. 630	62·7	3
0·35 C, 1 Mn, 0·5 Ni .	O.H. 850/T. 630	48·2	30
0·10 C, 0·7 Mn . .	Norm. 900	33·0	40
0·33 C, 2·75 Ni, 0·75 Cr, 0·5 Mo . . .	O.H. 830/T. 610	67·7	59
0·38 C, 0·70 Mn . .	Norm. 850	45·7	61
0·36 C, 3 Ni, 0·5 Cr . .	O.H. 830/T. 600	61·5	63

(*Swinden.*)

The tensile pieces were pulled at a rate of $\frac{1}{24}$th of an inch per min. in (*a*) oil and (*b*) in molten white metal. The values given above represent the percentage lowering of the elongation in (*b*) as compared with (*a*). The superior properties of the 3 per cent. chromium-molybdenum steel in this respect are well shown.

COPPER.

Copper containing at least 99·95 per cent. of the pure element is produced on a commercial scale in large quantities. The average tenacity of cast copper is about 10 tons per square inch. Rolling, forging or other kinds of hot or cold working, followed by annealing, have the effect of raising its tensile strength to more than 14 tons per square inch, with approximately 50 per cent. elongation. Cold working by rolling, drawing, pressing, spinning or hammering progressively hardens copper and raises its tensile strength although its ductility is decreased. Average values for high-purity copper are as follows :—

Condition.	U.T.S. Tons/sq. in.	Elongation. Per cent. on 2 ins.	Brinell Number.	Density.
As cast	10–11	25–30	40–45	8·5
Cold-worked . . .	20–26	5–20	80–100	8·9
Annealed after cold-working	14–16	50–60	45–55	8·9

(*Copper Development Association.*)

Very heavily cold-worked copper, which is generally in the form of wire, may have a tensile strength as high as 30 tons per sq. in., but its elongation is then between 1·0 and 5 per cent. Ductility is restored by annealing, but the annealing temperature required to induce maximum softness depends on its purity. High-purity copper is effectively softened by reheating to 200°–250° C., but if impurities are present in any appreciable amount temperatures between 300° and 400° C. may be required.

High-purity copper (not less than 99·9 per cent.) is used for electrical purposes. The relative electrical and thermal conductivities and coefficients of expansion of the metals are indicated in the following table:—

Metal.	Relative Electrical Conductivity.	Relative Thermal Conductivity.	Coefficient of Linear Expansion at 20° C. $\times 10^{-6}$
Silver	100	108	19
Copper	100	100	16·6
Gold	72	76	14
Aluminium	62	56	23
Magnesium	39	41	26
Zinc	29	29	30
Nickel	25	15	13
Cadmium	23	24	31
Cobalt	18	17	12
Iron	17	17	12
Steel	13–17	13–17	12
Platinum	16	18	9
Tin	15	17	21
Lead	8	9	28
Antimony	4·5	5	11

(*Copper Development Association.*)

The electrical conductivity of copper is reduced by cold-work. Hard-drawn has a conductivity of about 97 per cent. of that of annealed copper. Impurities, notably phosphorus, silicon, iron, and aluminium, reduce the electrical conductivity.

The thermal conductivity of copper is about 0·92 C.G.S. units and is decreased by impurities. Its electrical resistivity coefficient at 20° C. is about 0·00395 per ° C. The melting point of pure copper (graphite or charcoal covered) is 1,083° C.

Impurities in Commercial Copper.

Arsenic increases the hardness and the tenacity of copper. It improves the rigidity and strength of copper at high temperatures, hence, for fire-box plates, the use of copper containing from 0·3 per cent. to 0·5 per cent. of arsenic is frequently specified. Arsenic adversely influences both the electrical and thermal conductivities and about 0·40 per arsenic may reduce these properties by as much as 50 per cent.

Antimony is similar in action to arsenic, but it should be limited to traces in such copper as is to be used for the manufacture of brass that is to be cold-worked.

Bismuth being the most deleterious element that can be present in copper, owing to its tendency to induce red-shortness, not more than traces of this element should be allowed.

Iron is rarely present in copper in more than traces.

Lead is, after bismuth, the most objectionable impurity likely to be present in copper, although it is of value in metal that is to be rolled into sheets, on account of its softening effect on copper. Not more than the smallest quantities of this element should be allowed in copper for cold-worked brass.

Nickel hardens copper, but is in no way deleterious when present in such amounts as are usual in commercial copper.

Oxygen in small quantity does not have any very radical effect on the mechanical properties of copper. It is important in bringing copper up to the 'pitch,' and in combining with deleterious impurities to form insoluble and infusible compounds, thus separating and distributing these impurities and rendering them harmless.

Copper which has been deoxidised by the addition of either phosphorus, magnesium, boron, calcium or beryllium, is used for castings. To remove all the oxygen, a slight excess of the deoxidiser is required and this may adversely affect the electrical and thermal conductivities. The main advantages gained by the deoxidation of copper are soundness in castings, improved ductility and enhanced resistance to the action of reducing gases, such as hydrogen and carbon monoxide, at high temperatures. The resistance to attack by sulphurous gases in the atmosphere is improved by preliminary heating in air at 70° C.

Sulphur tends to embrittle copper, but is rarely present in refined copper.

Tellurium is similar to sulphur in its action on copper.

Silver, tin, and zinc are sometimes present in copper, but are rarely present in such quantity as to have much effect on the metal. Of these the least harmful is silver.

ZINC.

Zinc is a bluish-white metal, and, when pure, melts at 419·4° C., a point which has been accurately determined and frequently used in the standardisation of pyrometers. The boiling-point of zinc is 906° C. and its specific heat 0·10 grm./cal. per ° C. The thermal and electrical conductivities are about 25 per cent. of the corresponding values for silver.

The pure metal is mainly used as a protective coating on the surfaces of iron and steel articles, and may be applied by either galvanising (p. 1192), sherardising (p. 1195), or by spraying (p. 1194).

The most important zinc alloys are the brasses, the nickel-silvers or 'nickel-brasses,' and the aluminium-base and zinc-base alloys used mainly for 'gravity' and 'pressure' die-castings. Cadmium-zinc alloys containing about 70 per cent. cadmium and 30 per cent. zinc are used for special solders.

Impurities in Commercial Zinc.

Arsenic is a common impurity in spelter. It does not occur in such quantity as to have a deleterious effect upon the mechanical properties of spelter or of such alloys as contain it. It is, however, an objectionable ingredient in such zinc as is to be employed in the production of hydrogen.

Cadmium is nearly always present in zinc; in electrolytic varieties of the metal it rarely exceeds 0·07 per cent., in other varieties as much as 0·40 per cent. is found. The effect of this element is to harden spelter and to make it more brittle. Cadmium is, on this account, objectionable when the zinc has to be rolled into sheet or to be used for galvanising telegraph wires. In zinc for the manufacture of brass subsequently to be cold-worked as much as 0·20 per cent. of cadmium may safely be allowed.

Copper tends to harden zinc, but is rarely present in quantity sufficient to effect radical change in the metal.

Iron hardens and embrittles zinc. Its presence tends to the formation of dross in the galvanising process.

Such zinc as is to be used in the manufacture of cold-worked brass should not contain above 0·10 per cent. of iron. Electrolytic spelters rarely contain above 0·025 per cent.; as much as 2·5 per cent. of iron has, however, been found.

Lead is present in certain brands of zinc up to 2·5 per cent. The purer brands of spelter contain up to 0·20 per cent. of lead. The presence of lead is not deleterious where the spelter containing it is to be used for general brass manufacture, but where brass is to be cold-worked or specially treated care must be taken that the spelter used has not too high a content of lead.

Lead in moderate quantity tends to make zinc softer in rolling. When present in quantities in excess of 0·7 per cent. it tends to the production of serious cracks in slush castings of zinc.

Other elements occasionally present in spelter in traces are sulphur and carbon.

TIN.

The London Metal Exchange form of contract (1911) for the purchase of tin recognises two classes of tin—Class A, tin assaying not less than 99·75 per cent. of pure metal; Class B, tin assaying not less than 99 per cent. of pure metal. Chempur tin contains more than 99·99 per cent. of the element.

The tensile strength of cast tin varies (according to Thurston) from 2,000 to 6,000 lbs. per square inch. The elongation is, for all practical purposes, nil.

The principal impurities in commercial tin are lead, iron, copper, and arsenic. Sulphur, silver, bismuth, and antimony are of frequent occurrence. Tin offers excellent resistance to corrosion under many different conditions, and it is for this reason that it is so widely used in containing vessels for food, fruit and milk and for protecting copper wire and cable from attack by the sulphur in the vulcanised rubber used for insulating purposes.

An important application of tin is in connection with the electro-deposition of tin coatings to the cast iron and aluminium-alloy pistons in automobile engines in order to facilitate 'running-in.' Tin-coated bearings are also employed on some rolling mills.

The melting-point of pure tin is 232° C., but its boiling-point is extraordinarily high (about 2,260° C.).

NICKEL.

Nickel containing small quantities only of impurities is a commercial article. After rolling and annealing commercial nickel has the following properties :—

Yield point 9–13 tons per square inch, maximum stress 27–32 tons per square inch, elongation 40–55 per cent. on 2 inches. By cold-working after annealing the tensile strength may be raised to as much as 60 tons per square inch.

Impurities in Commercial Nickel.

Carbon may be present in nickel up to 0·20 per cent., though it rarely exceeds 0·10 per cent. It has an embrittling effect on nickel when in the form of graphite, but when combined with nickel as nickel carbide it has a strengthening effect.

Cobalt has a strengthening and hardening effect on nickel, but in quantities—up to 1·00 per cent.—usually found in commercial nickel, this element has but little effect. Nickel free from cobalt is obtained by the Mond process.

Copper is similar to cobalt in its action, but rarely exceeds 0·20 per cent. in amount.

Arsenic should not exceed about 0·025 per cent. in such nickel as is to be subjected to cold-work, or in alloys for a similar purpose.

Iron has an effect on nickel similar to that of cobalt and chromium, but, present in quantities less than 0·75 per cent., its presence is scarcely noticeable.

Sulphur is occasionally noted in samples of nickel.

Nickel, if it is to be satisfactory for the manufacture of copper-nickel alloys for cold-working, should be free from all unreduced oxides.

Carbonless nickel containing as much as 1 per cent. of oxygen in the form of nickel oxide is used, however, for anodes. Even though it contains some of the nickel nickel-oxide eutectic the material is readily hot-rolled to any requred section. The oxide in it serves to minimise polarisation effects.

CHROMIUM.

The colour of chromium is silver-white. Nickel has a little more of a yellow tinge to it than chromium. On the other hand, chromium will hold its colour indefinitely. Nickel will turn almost black in a laboratory atmosphere, and chromium will stay bright indefinitely. It is about the only metal known that will stay bright in a laboratory atmosphere. Not even platinum will stay as bright.

It is for this reason that chromium-plating has become so popular. A suitable electrolyte for plating contains 30–33 oz. of chromic acid and 0·3 oz. sulphuric acid per gallon of water. The best deposits are obtained at temperatures between 45° and 55° C. with a current density of about 140 amps. per square foot of surface. To avoid any tendency to peeling it is customary to undercoat the chromium with a deposit of nickel. The melting point of chromium is 1,830° C.

ALUMINIUM.

For electrical conductors, and other special purposes, a grade of aluminium having a minimum purity of 99·5 per cent. is available. The lower grade, having a purity of about 99·2 per cent., is used for rolling into sheets for the manufacture of cooking utensils, chemical plant, etc. The

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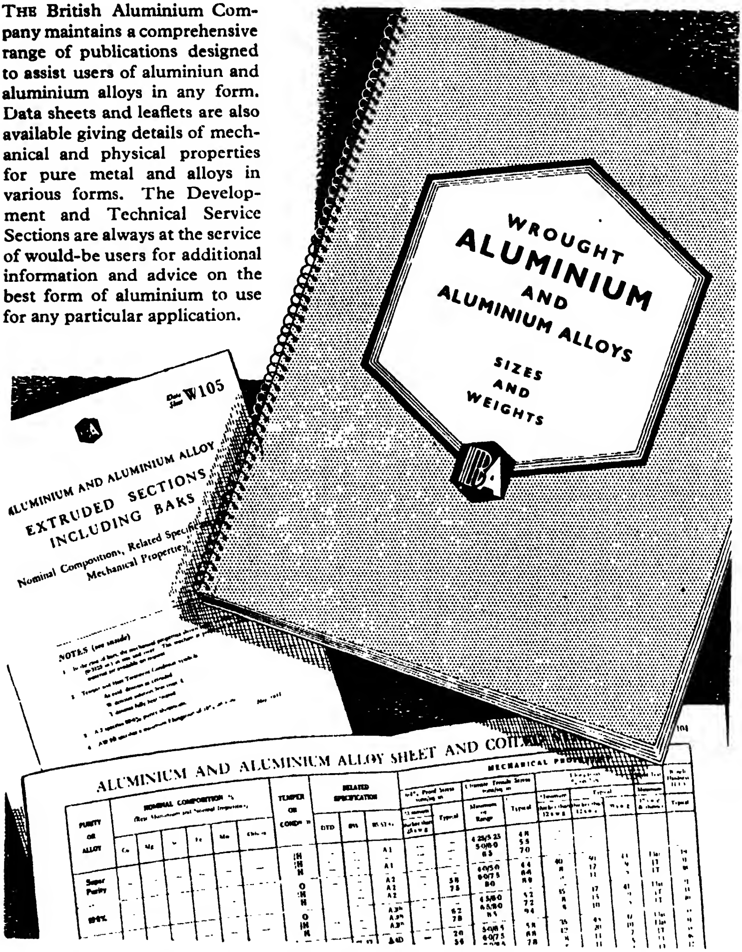

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impurities generally present in primary aluminium are: copper, iron, silicon, sodium, zinc, nitrogen, and a typical analysis of high-purity aluminium is as follows:—

Aluminium	99·5 per cent.
Iron	0·3 „
Silicon	0·2 „

Aluminium is one of the lightest of metals and has a specific gravity of about 2·70. The mechanical properties of cast and annealed wrought aluminium are as follows:—

Condition.	Ultimate Tensile Strength.	Elongation. Per cent. on 2 ins.	Brinell No. (500 kgrm.)
Cast	5–7 tons/sq. in.	3–4	20
Rolled and annealed .	10–15 „	30	24

The tenacity of cold-rolled aluminium can be between 10 and 18 tons per square inch.

Primary aluminium is not used in the cast condition for constructional purposes. It is used as a deoxidiser in the manufacture of steel, and in the form of powder and dust in the thermit process and in paint. The principal use is in the wrought form, such as sheets, tubes, mouldings, rods, wire, etc. As commercially pure aluminium has extensive application in the brewing and chemical industries, and also for domestic purposes, it is important to consider the action of acids upon it, which may be briefly stated as follows:—

Hydrochloric Acid.—Attacked with great rapidity hot or cold and at all concentrations.

Hydrofluoric Acid.—Rapidly attacked.

Sulphuric Acid.—Attack slow with concentrated acid. Appreciably attacked with hot dilute acid.

Nitric Acid.—Attack very slow, the metal becoming passive (can be used for containers for cold strong HNO_3).

Acetic Acid.—Attack slow with cold acid, increases with use of temperature.

Lactic Acid.—Attacked very slowly (suitable for milk containers).

Oleic Acid.—Practically no effect.

Action of Air.—Attack very slow in fairly pure air, increases considerably in impure atmospheres. (The oxygen forms a thin film of oxide which tends to retard further action.)

Action of Alkalies.—Potassium and Sodium Hydroxide.—Rapidly attacked. Ammonium Hydroxide.—Attack slow. Alkaline Water.—Attacked appreciably by water in which alkalies or soap have been added.

Most metals will alloy with aluminium, but only a few definite alloys (aluminium—copper, iron, manganese, magnesium, nickel, silicon, tin, zinc) are used to a large extent. These include the binary alloys Al-Zn, Al-Cu, ternary alloys Al-Cu-Zn, Duralumin and related alloys, silicon and certain complex alloys. The so-called heavier alloys are the aluminium brasses and bronzes. Certain iron-aluminium brasses and manganese-aluminium brasses are also used. Aluminium-chromium-iron alloys are used as electrical resistances for heating elements (p. 1252).

Annealing Sheet Aluminium.

1. Prepare a mixture of ordinary whiting and oil, mixed to the consistency of thick paste. Coat the metal on both sides with this preparation, and place it on a clean fire or in a furnace, allowing it to remain until the oil in the paste becomes ignited. When this occurs withdraw the work from the furnace and allow it to cool gradually. The coat of whiting should then be removed from the surface of the metal by washing and brushing.

2. Heat the metal to a dull red colour, perceptible in the dark, and cool slowly. With this method care must be exercised to get an equal annealing temperature distributed over the entire surface of the work.

3. Heat the sheet aluminium until pine-wood sawdust just glows when sprinkled on it, which indicates a temperature of about 400° C. At this temperature a dry match-stick will just char when rubbed on the metal. (*A. Eyles.*)

The Protection of Aluminium from Corrosion.

Aluminium and its alloys can be efficiently protected against atmospheric corrosion under the most severe conditions by the use of an enamel or varnish paint. Oil paints are very inferior in this respect. The best results are obtained by anodic oxidation (see p. 1239).

MAGNESIUM.

PROPERTIES OF PURE MAGNESIUM.

	Pure Magnesium.		Pure Aluminium.	
	Cast.	Worked.	Cast.	Worked.
Ultimate strength. Tons per sq. in.	7·5	14·5	5·2	10·1
Proof stress „ „	—	—	—	5·03
Elongation on 2 in., per cent.	6	6	35	16
Modulus. Lbs. per sq. in.	—	$6·74 \times 10^6$	—	$10·1 \times 10^6$
Rigidity modulus. Lbs. per sq. in.	—	$0·58 \times 10^6$	—	$3·45 \times 10^6$
Impact, Izod. Ft.-lbs.	—	4·0	—	24
Endurance limit, range of stress. Tons per sq. in.	—	± 3·48	—	± 4·69
Hardness, Brinell No.	20	42	23	45
Density. Grammes per c.c.	—	1·72	—	2·70
Thermal expansion per ° C.	—	25×10^{-6}	—	24×10^{-6}
Electrical resistivity. Microhms per cm.^{3}	—	4·59	—	2·84
Thermal conductivity (C.G.S. units)	—	0·34	—	0·52

Magnesium alloys (see p. 1231).

ANTIMONY.

Antimony is a bluish-white lustrous metal. It is extremely brittle, readily powdered, and is a poor conductor of heat and electricity. The pure metal melts at 631° C., and is sometimes used in the standardisation of pyrometers, but otherwise has little or no industrial application. Its most important alloys are the copper-tin-antimony, lead-tin-antimony and lead-tin-antimony-copper bearing metals, 'Britannia metal' (p. 1243) and type metal (p. 1248). Antimony is also used in the manufacture of shrapnel bullets and in cable sheathing. A notable feature of the antimony alloys is their property of expanding during freezing.

BERYLLIUM.

Beryllium is a whitish metal and one of the lightest. Its specific gravity is 1·82 compared with 2·70 for aluminium and the metal melts at 1,284° C. Although brittle when cold, beryllium is capable of being hot-worked. The pure metal has limited applications, although it is used in X-ray work and for the electrodes in neon signs. It is also used as a deoxidant for molten copper. Some of its alloys, notably copper-beryllium (pp. 1227 and 1241), are of considerable importance, since the resulting alloys may be heat-treated in consequence of the great difference between the concentrations of the solid solutions at ordinary and elevated temperatures. After quenching from high temperatures and 'ageing' at lower temperatures, the alloys with copper and nickel yield extraordinarily high hardness and tensile strength values. Ferro-beryllium, containing about 80 per cent. of the metal, has been used in the case-hardening of steel and it is claimed that an equivalent Brinell hardness of more than 1,000 can be obtained by this means.

CAESIUM.

Caesium is a silvery-white metal, resembling potassium. It is a soft metal, melting at 28·6° C., and reacts readily and explosively with water and oxygen. It is used in the manufacture of photo-electric cells for television and modern cinematography, and certain other automatic processes.

CALCIUM.

Calcium is an exceedingly soft, silvery-white metal which readily oxidises and actually burns when even gently heated in air. It is for this reason that the metal and its alloys are used for the deoxidation of molten metals and their alloys. Lead-calcium alloys, generally containing barium also, are used for bearings.

CADMIUM.

Cadmium is a soft white metal, with a melting point of 321° C. The pure metal finds little industrial application apart from electro-deposition and metal-spraying; both these processes are often applied to iron and steel articles with the primary object of resisting corrosion.

Copper-cadmium alloys are used for overhead trolley wires (p. 1227); lead-cadmium alloys for bearings, and the metal is a constituent of certain solders, notably of the special solders used for aluminium alloys.

CERIUM.

Cerium is a steelish-grey metal, resembling the alkali metals, sodium and potassium, in many of its properties. It is somewhat harder than lead but is readily cut. The metal reacts slowly with water and rapidly burns when heated in air.

Probably the most important application of cerium is in connection with the modern 'petrol lighter' and similar devices. The flints used for this purpose consist of ferro-cerium, which emits copious sparks when rubbed with a sharp file or toothed wheel. Ferro-cerium has also been used as a deoxidant for molten steel.

COBALT.

Cobalt is a bluish-white metal, having general properties intermediate between those of iron and nickel. Thus, apart from iron, it is the most magnetic of metals and actually retains its magnetic properties at higher temperatures; iron loses its magnetism above 900° C., but cobalt remains magnetic up to about 1,100° C.

The melting point of cobalt is 1,466° C.

The metal is readily electro-deposited but, apart from plating, the element itself has little or no practical applications, although many of the alloys containing cobalt are of the greatest interest and importance. It is a constituent of modern 'super' high-speed cutting steels, the nickel-aluminium-cobalt and nickel-cobalt-titanium steels used for permanent magnets of high coercivity, and as a bond or matrix for tungsten carbide for the production of 'tipped' machine tools of the 'Widia' and Carboloy type. The well-known 'Stellite' alloy (p. 1161) contains about 45 per cent. of cobalt, and about 3 per cent. of the metal is contained in the 'Kanthal' alloys employed for electric resistance-heating purposes. In the latter respect, the cobalt alloys are claimed to have a longer life at higher temperatures than the usual nickel-chromium alloys used for similar purposes.

COLUMBIUM (NIOBIUM).

Columbium is a greyish metal, having a hardness comparable with that of pure iron, and is available in wire, rod and sheet form. The element is associated in nature with the oxides of tantalum. Separation of the two metals is not easy, so that columbium alloys generally contain some proportion of tantalum. Both metals have been used in the manufacture of rectifiers in radio installations. The most important use of columbium is its introduction into the 'austenitic' nickel-chromium stainless steels for the purpose of inhibiting 'weld-decay,' a phenomenon due to carbide precipitation along the grain boundaries when the steel has been heated to a temperature within the range 600°–900° C. Weld-decay has proved to be a very real problem in the past in connection with austenitic stainless steels and the employment of columbium has gone a long way towards the solution. To be really effective, the amount of columbium needed is at least ten times the amount of carbon present, and the metal is generally added in the form of its ferro-alloys.

MOLYBDENUM.

Molybdenum is a dense white metal (sp. gr. 10·0). It oxidises and volatilises when heated at about 600° C., although the melting point (in an inert atmosphere) is 2,500° C. Ordinarily the metal is brittle, but by high-temperature swaging and heat-treatment, large single crystals can be obtained which are, then, ductile enough to permit rolling into sheets or drawing into wire. In the latter form, together with tungsten wire produced in a similar way, it is used for thermo-couples in very high temperature work. Tungsten-molybdenum couples can only be used in inert atmospheres, nitrogen and hydrogen mixtures, generated from 'cracked' ammonia vapour generally being used.

Molybdenum is mainly used as an alloying element in special steels and cast-irons, and may then function in several different ways. Thus it reduces the susceptibility of nickel-chromium steels to 'temper-brittleness,' increases their strengths at elevated temperatures, and improves the cutting efficiencies of 'high-speed' steel both at ordinary and elevated temperatures. (See also p. 1184.)

MANGANESE.

Pure manganese is generally regarded as a hard, brittle metal having no real practical use although it is an important constituent of many ferrous and non-ferrous alloys. In the great majority of commercial steels manganese must be regarded as an essential constituent and, indeed, of similar importance as their carbon contents. In steel, manganese fulfils several most useful functions. In the first place it acts as a deoxidiser and desulphuriser of the molten steel. In the solid steel, manganese negatives the adverse influence of sulphur by converting the latter into the relatively harmless ' manganese sulphide ' particles which do not appear to have much effect on the usual mechanical properties.

A most important influence of manganese on steel is that of ' hardenability.' With carbon steels of very low manganese content, the hardness depth, due to quenching is shallow.

Manganese also has a marked effect on the volume alterations which occur when steel is quenched. The volume of a hardened piece of carbon steel is greater than that of the soft unhardened material, but in many of the so-called ' non-shrinking ' steels containing about 1 per cent. each of carbon and manganese, this difference in volume is almost negligible.

Manganese exerts a similar effect on the usual mechanical properties of normalised or hardened and fully tempered steels as does carbon, but to a milder degree. Thus its general effect is to increase tensile strength, but lower the ductility.

Austenitic manganese steel contains about 1 per cent. of carbon and 12–14 per cent. manganese. This steel, which is non-magnetic, exhibits a remarkable resistance to abrasion. Its Brinell hardness is generally about 200 only and the steel is exceedingly tough, but when subjected to pressure or abrasion, the surfaces rapidly work-harden, a property which renders the steel particularly suitable for rock-crushing machinery, dredger-buckets, tramway and railway crossings and on bends where wear is heavy, etc. The life of high manganese steel rails used on sharp bends with very heavy traffic is given as nine years, as against nine months for ordinary steel rails. Manganese steel is used for bullet-proof helmets, as it is very resistant to penetration and does not shatter.

In its toughest condition, obtained by quenching from about 1000° C., the steel has a tensile strength of about 55–60 tons/sq. in. with 40 per cent. elongation. Even in this condition, it can only be satisfactorily machined by tools made from high-speed steels of special quality or ' tipped ' tools of the Widia or Carboloy type.

High manganese steel of this kind should never be annealed.

Manganese is also a constituent of ' manganese bronze,' the well-known resistance alloy Manganin,' and a number of aluminium-base light alloys, etc.

SILVER.

Pure silver is one of the whitest of metals and will yield a most brilliant finish when properly polished. It is for these reasons that silver-plated articles are so pleasing. It is a very malleable and ductile metal and in the annealed state is the best conductor of heat and electricity known. The density of silver is about 10·5, but the casting of the pure metal is difficult, owing to silver ' spitting ' giving rise to the evolution of gas during freezing, which considerably reduces its specific gravity. The resulting pores are closed by subsequent cold-work, however, and the metal then possesses its normal density. The melting or freezing point of pure silver, melted under charcoal, is 961° C., and this point is often used in the calibration of pyrometers.

Silver finds considerable application in industrial processes in view of its marked resistance to attack by caustic alkalis and certain acids. Thus, it is used in the distillation and handling of acetic acid and vinegar. The metal finds wide application in electro-plating, and is also used to harden copper utilised for certain electrical purposes as it does not adversely affect its electrical conductivity. Small proportions of silver have been introduced into the so-called lead-bronze bearing alloys used in aircraft (p. 1223).

SELENIUM.

Selenium is an allotropic element, capable of existing in three different forms. Although in many of its properties it closely resembles sulphur, the grey form of the element conducts electricity and is therefore known as ' metallic selenium.' The electrical conductivity of this metallic form of selenium increases enormously when exposed to light and the change is reversible, in the dark the resistivity returns to its previous maximum. It is assumed that the element is a mixture of two forms having greatly different conductivities and that the equilibrium between them is displaced by light. Practical advantage is taken of this in the photo-electric cell, utilised in quite a number of automatic processes.

Selenium has also been introduced into copper alloys and the austenitic stainless steels in order to improve machineability. In the latter case it is claimed that, unlike sulphur, it does not materially influence the corrosion resistance of the steel.

TANTALUM.

Tantalum is a white metal when polished but unpolished surfaces often appear steely-blue, due to oxide films. Although relatively soft when pure, it rapidly work-hardens. As indicated, tantalum is always likely to contain some proportion of columbium, and like the latter has been used to prevent weld-decay in austenitic steels.

Tantalum resists attack by some of the most powerful acids, e.g. hydrochloric and nitric acids, aqua-regia, and sulphuric acid, so that it has been used as a substitute for platinum for certain purposes. Like columbium, it readily combines with a number of gases, e.g. oxygen, hydrogen, nitrogen, etc., when heated.

Tantalum has been introduced, in the form of its ferro-alloys, into certain high-speed steels and acid-resisting alloys. Tantalum carbide (cemented in a nickel base) alloys have been used in place of the tungsten-carbide (cobalt base) alloys of the Widia and Carboloy types. Another alloy of this type (Miramant) contains tungsten, molybdenum, carbon and tantalum (p. 1161).

TELLURIUM.

Tellurium is a white, crystalline-looking metal. It is a poor conductor of heat and electricity. The element is undesirable in most commercial alloys. For instance, even minute amounts exert a detrimental influence on the properties of copper and its alloys, although, for certain purposes, it has been introduced in order to improve their machining properties. The metal has been added to steel for the same purpose. A useful application of tellurium is in connection with 'tellurium-lead.' (See p. 1242.)

TITANIUM.

Titanium is a hard brittle metal which burns in air when strongly heated and when heated in nitrogen combines with it to yield a stable titanium nitride.

The element, in the form of ferro-titanium, is a powerful deoxidant of molten steel and small proportions are often added for this purpose alone; it also serves to remove nitrogen in the steel by converting it into the insoluble titanium nitride. Larger proportions, up to 1·5 per cent., are added to the 'austenitic' stainless steels to decrease their susceptibility to weld-decay. From 0·1 to 0·2 per cent. titanium is frequently added to certain light aluminium alloys in order to improve their properties, as also those of other non-ferrous alloys. A newer application of titanium is the incorporation of ferro-titanium in the coatings of electrodes for the welding of steel.

TUNGSTEN.

Tungsten is generally obtained in the form of a hard, brittle, greyish powder. By repeated sintering and swaging in an inert atmosphere, with intermediate heat-treatments, the aggregate eventually becomes one single crystal, sufficiently ductile to be drawn into wire for use in thermocouples and electric light bulbs. Apart from the latter, its principal use is as an alloying element in high-speed steel and other cutting alloys (see pp. 1160–1161). Tungsten confers the property of 'red-hardness' i.e. the ability of tools to continue cutting even at a temperature of a red heat, and even small proportions refine the grain of high carbon tool steels. Steels used for taps and dies frequently contain about 1·0 per cent. carbon, and up to 2 per cent. tungsten.

Magnet steels containing 5–6 per cent. of tungsten are still made, where exceptionally high coercive force values are not essential. These steels are considerably cheaper than the cobalt-chromium, nickel-aluminium, nickel-aluminium-titanium and nickel-aluminium-cobalt steels used for permanent magnets. Steels containing about 2 per cent. carbon and 5–10 per cent. tungsten are used for drawing dies. The element has also been introduced to the extent of about 2 per cent. into certain alpha-bronzes in order to increase the hardness and strength.

Tungsten is a very heavy element and, on this account, when it is desired to add it to molten steel, the less dense ferro-alloys are generally preferred. The melting point of tungsten is about 3400° C., and the element dissolves very slowly in molten iron. The melting point of ferro-tungsten is considerably lower than this, so that homogeneous liquid metal is more easily obtained.

VANADIUM.

Pure vanadium is a brilliant white metal of great hardness. It is chiefly used as a deoxidiser and as an alloying element in steel. For the former purpose the equivalent of about 0·05 per cent. only may remain in the solid steel. Higher proportions (0·15 to 0·4 per cent.) may be present in chrome-vanadium spring steels and still higher percentages (0·5 to 2·0 per cent.) in modern 'super'-high-speed cutting steels containing tungsten and chromium. Vanadium exerts an important influence on the secondary hardness of high-speed steel. It is an expensiv eelement and this, in part, accounts for its limited applications.

Vanadium is generally introduced into steel in the form of ferro-vanadium, containing about 40–45 per cent. of the element.

ZIRCONIUM.

This metal has been added to steel and appears to exert a similar influence to titanium. It is added in the form of ferro-zirconium and it is claimed that the element counteracts the adverse effects of sulphur on the mechanical properties of steel, although any harmful influence of sulphur is carried out more cheaply, and perhaps more efficiently, by proper control of the manganese-sulphur ratio.

NON-FERROUS ALLOYS. (See also page 1249.)

COPPER AND ZINC.

The alloys of copper and zinc containing over 50 per cent. of copper are generally termed brasses.' Average mechanical properties are as follows:—

PROPERTIES OF STRAIGHT COPPER-ZINC ALLOYS.

Composition. Copper. Per cent.	Composition. Zinc. Per cent.	Condition.	Tensile Strength. Tons per Sq. In.	Elongation. Per cent. on 2 ins.	Reduction of Area. Per cent.	Brinell Hardness.	Scleroscope Hardness.
70·0	30·0	As cast . .	16·7	58	57	55	15
		As forged and annealed .	21·5	68	65	57	15
59·0	41·0	As cast . .	24·9	45	49·7	90	14
		As forged .	26·0	47	62	90	14
		Forged and annealed at 650° C. for 1 hour	24·0	49	55	79	12
53·3	46·7	As cast . .	29·7	24	21·5	108	18
		As forged .	32·8	28	30·6	114	18
		Forged and annealed at 650° C. for 1 hour	29·1	22	31·8	108	18
51·2	48·8	As cast . .	26·9	19	21·5	108	18
		As forged .	33·4	37	33·5	114	18
		Forged and annealed at 650° C. for 1 hour	29·0	25	27·0	108	18
50·19	49·81	As cast . .	8·8	1	1·5	108	18
		As forged .	15·8	5	5·5	117	19

(*Smalley.*)

The casting temperatures of these alloys are of exceptional moment. The following tests show the influence of varying casting temperature on the tensile properties of zinc-copper alloy castings poured from one crucible within a short interval of each other. Chemically the three castings were identical.

Casting Temperature °C.	U.T.S. Tons per Square Inch.	Elongation. Per cent.
1,038	12·45	6·0
943	16·28	9·5
934	18·88	15·0

In every case with other alloys similar temperature variations have a similar effect on the mechanical properties.

In judging a suitable pouring temperature the appearance of the alloy in ladle or crucible may be used as a guide. With manganese bronzes this appearance is somewhat confused by the aluminium present, for, as is well known, this imparts a characteristic skin to the surface of the molten alloy. The practical effect of this is to make the alloy appear colder and less fluid than is really the case. Careful observation readily overcomes this difficulty.

There is little doubt that the pyrometer is, in the hands of careful workers, the best means of determining the temperatures of metal or alloy in the ladle or crucible.

Cast Brass.

Cast brass is somewhat variable in composition, but usually contains about 66 per cent. of copper, this percentage being that found in English standard brass.

The metals employed in the production of cast brass are not generally selected with such care as are those used in the manufacture of malleable brass.

Lead in proportions varying from 1 to 3 per cent. is frequently added for the purpose of facilitating machining. It reduces the tenacity and hardness and increases the corrodibility of the resulting alloy, and is liable to segregate in those parts of the castings that solidify latest on cooling.

Tin tends to harden brass and is added in quantities up to 3 per cent., though 2 per cent. is not generally exceeded.

Iron hardens and increases the tenacity of brass.

The presence of more than one of these elements in quantity is to be deprecated, as their combined effects on the mechanical properties of the resulting alloy are likely to be injurious.

The mechanical properties of cast brass of average composition (66 per cent. of copper) and containing only small quantities of impurities or added elements are :—

Tenacity, 14 tons per sq. in.; elongation, 40 per cent.

A number of alloys, adapted for particular purposes, are tabulated below :—

Alloy.	Cu. Per cent.	Zn. Per cent.	Other Metals. Per cent.
Very delicate castings	86·00	14·00	—
Good yellow brass	75·00	25·00	Sn 2·00
Name plates	86·00	9·00	Sn 3·00; Pb 2·00
Small castings (red)	90·00	10·00	" "
Taps, cocks, etc.	78·00	14·00	Sn 3·00; Pb 3·00
Fine castings (yellow)	71·00	24·00	Sn 2·00; Pb 3·00
Hydraulic pumps	58·00	38·00	Sn 2·00; Fe 2·00

Ductile Brass.

The brasses containing upwards of 63 per cent. of copper, known as α brasses, are ductile and malleable at ordinary temperatures, provided that the metals used in the production of the alloys be of the best quality, and that the alloys be well annealed during working. The most suitable annealing temperatures for ductile brass are between 600° C. and 700° C.

The annealing temperatures required are raised by even small proportions of impurity, notably iron. If annealed at too high a temperature large crystals are formed and then subjected to further deformation by cold-work, as in deep pressing, may give rise to the 'orange-peel' effect. This can be avoided by keeping the grain-size between 0·03 and 0·04 mm. Over-annealing also tends to spoil the surface of brass due to scaling. In this connection atmosphere control is important.

Brasses containing from 5 to 20 per cent. of zinc are known as *gilding metal*; when properly annealed they are exceedingly malleable and have a very pleasing colour.

Lead is admissible in alloys that are to be rolled into sheet, but is injurious in alloys that are to be drawn into wire. It should not be present in such quantity as is liable to create trouble at the annealing furnace. Lead is deliberately introduced solely to facilitate machining. The most ductile of the brasses contains 70 per cent. of copper and 30 per cent. of zinc (known as 'cartridge' brass) and may be expected to have the following properties :—

Condition.	Proof Stress (0·1 %) Tns./sq. in.	U. T. S. Tns./sq. in.	Elongation. Per cent on 2 in.	Brinell Number.
Chill castings (*i.e.* strip ingots prior to rolling)	6	16	60–70	60
Hard-rolled sheet	Over 25	30–40	10–15	150–200
Sheet annealed after rolling	6	20–23	65–75	60

(*Copper Development Association.*)

Cold-working sets up internal strains, and if these are unevenly distributed spontaneous cracking may subsequently occur when exposed to even mild corrosive atmospheres, particularly those containing traces of ammonia. This phenomenon is described as *season-cracking* and is inhibited by tempering the cold-worked brass for about an hour at 250°–270° C. The susceptibility to season-cracking is revealed by immersion in a solution of mercurous nitrate, containing a little free nitric acid.

Like many other non-ferrous alloys the brasses show a marked resistance to corrosion-fatigue. Cold-worked brass, nickel-silver, and phosphor-bronze are used for springs, and the following comparative values are of interest.

Material.	Modulus of Rigidity. Lb./sq. in.	Torsional Stress. Lb./sq. in.
Steel	$11 \cdot 0 \times 10^6$	—
Phosphor-bronze	$6 \cdot 5 \times 10^6$	45,000
70/30 brass	$4 \cdot 8 \times 10^6$	40,000
Nickel-silver	$6 \cdot 5 \times 10^6$	40,000

Hot-forgeable Brass.

The alloys of copper and zinc containing from 57 to 61 per cent. of copper, described as α-β brasses, are characterised by their ability to withstand hot rolling at temperatures ranging from 600° C. to 800° C. They are employed in sheet form for the sheathing of wooden ships and in bar form for the manufacture of bolts, fastenings, etc. They are forgeable under the drop hammer and, hence, are frequently used in the manufacture of stampings. In this last connection it is particularly noteworthy that constancy of composition must be attained, on account of the large variations in machining properties accounted for by small changes in the copper content. The alloys in the range 57 to 63 per cent. of copper are forgeable, though not capable of withstanding rolling, in the temperature range already stated, and may be considered here.

All the above alloys are retained in a harder, stronger, and less ductile condition if quenched from temperatures between 750° C. and 800° C. than if allowed to cool slowly. They are liable to spontaneous cracking if worked at temperatures below 600° C., which liability, if thought to be inherent, can be lessened if the alloys are annealed at temperatures between 200° C. and 300° C., subsequent to working. The best temperatures for annealing, with a view to reducing variations in the mechanical properties due to variations in the mechanical and thermal treatment, are below rather than above 650° C. The temperature range (750° C.–800° C.) is, according to Stead, a dangerous one in which to conduct the annealing of hot-forgeable brass.

The average mechanical properties of fully annealed cast 60/40 brass are as follows :—

Yield point	10 tons per sq. in.;	Elongation	35 per cent.;
Tensile strength	25 ,, ;	Hardness (Brinell)	85.

Drawn rod of 60/40 composition is obtainable having the following average mechanical properties :—

Yield point	28 tons per sq. in.;	Elongation	20–30 per cent. on 2. ins.;
Tensile strength	32 ,, ;	Hardness (Brinell)	140.

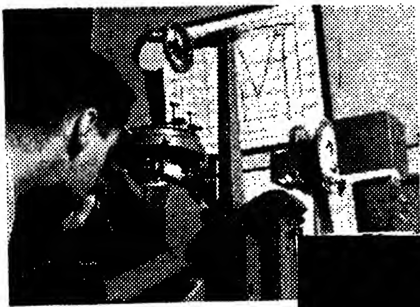
The last factor appears to be dependent on the diameter of the rod in question, being least when the diameter of the rod is greatest. On account of the almost infinite variety of hot- and cold-working possible in the treatment of these alloys almost infinite variations in mechanical properties are obtainable.

Hot Brass Pressings.*

Pressings are made from metal heated to the plastic state and squeezed to shape by heavy pressure. Accuracy can be kept to within close limits, within a few thousandths of an inch, the closeness depending on and varying according to the shape and size. Plain and straight cored work presents no difficulty, while parts that are undercut can, within reason, be produced by species dies. Tail pipes and right-angled and obtuse bends can be produced with clean cores and sound walls of even thickness. Walls of less than $\frac{3}{32}$ in. should be avoided.

The correct design of the dies is of extreme importance. For work which is plain in design, carbon steel dies are suitable. For designs sharp in outline with squares or hexagons, sharp corners or for extrusion work, the cobalt and nickel chrome, tungsten, vanadium and manganese steels are used. The life of the dies varies greatly ; approximately 2,000 heavy pieces and 10,000 light pieces may suffice to fully cover the cost of the dies.

* Willis Beard.



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F. 1214

The following alloys can be used for hot pressings :—

(a) Forging quality brass, 58–60 per cent. Cu, up to 2 per cent. Pb, balance Zn.
(b) Manganese bronze, 60 per cent. Cu, 1 per cent. Al balance Zn (total impurities under 0·75 per cent.).
(c) Naval brass, 61 per cent. Cu. min., 1 per cent. Sn, total impurities 0·75 per cent. max. balance Zn.
(d) High conductivity copper, 99·9 per cent. pure.
(e) Special alloys: nickel brass, aluminium brass, silicon copper, aluminium zinc, aluminium silicon.

'DELTA' BRAND ALLOYS.

The metals and alloys sold by the Delta Metal Co., Ltd. (of London and Birmingham), are employed for a very great variety of purposes, the composition varying widely according to the requirements and purposes for which each of these metals or alloys is more particularly adapted. Some of them are based upon the introduction and chemical combination of definite quantities of iron and other elements in copper-zinc alloys. In most of the alloys which may be classed under the 'Brasses,' the copper content lies between 57 and 69 per cent., that of zinc between 42 and 38 per cent. whilst some contain iron, phosphorus, manganese, or other elements up to a total of about 2 per cent.

Delta Bronzes Nos. I and IV are practically incorrodible; they are not attacked by sea-water, and resist acids much better than other copper alloys. On account of their great strength and resistance to corrosion, these alloys are largely used for parts of marine engines and mining and hydraulic plant which require a strength not less than that of the best steel, but at the same time freedom from corrosion—such as pump rods, piston rods, shafts, cranks, plungers, etc.

These alloys are malleable brasses of great strength, exceeding, even in the cast state, the strength of mild steel. They can be cast with facility, and will produce perfectly sound and homogeneous castings. When heated to a dull red heat, they become soft and highly malleable, and can be readily forged, stamped, rolled, or pressed, to any extent required. These operations increase the strength considerably, as will be seen by the following copies of tests.

Description of Metal.	Diameter. In.	Area. Sq. in.	Ultimate Strength. Tons per sq. in.	Elongation in 2 ins. Per cent.	Reduction of Area. Per cent.
		Delta Bronze, No. I.			
Cast . . .	0·500	0·1963	41·30	20·0	20·4
Extruded bars	0·502	0·1979	49·82	26·0	24·9
		Delta Bronze, No. IV.			
Cast . . .	0·611	0·293	23·89	21·0	20·1
Extruded bar .	0·500	0·1963	37·1	27·0	20·0

Extrusion Process.

The process of extruding copper and other alloys consists of the pressing or squeezing of heated metals through dies of any desired cross-section, by hydraulic pressure varying according to the section to be produced and the character of the metal operated upon.

Extruded bars and other sections such as tubes have a smooth and uniform surface, free from cracks or other defects; size and shape are of great accuracy, and owing to their superior finish, machining is reduced to a minimum.

Extrusion dies and mandrels must withstand abrasion, resist the influence of heat on their mechanical properties and dimensions and the effect of sudden changes of temperature in regard to cracking and distortion. Special steels have been evolved for this purpose (see p. 1185).

EFFECTS OF OTHER ELEMENTS ON THE PROPERTIES OF BRASSES.

Iron is a constituent of almost all high-tensile brasses and, as indicated, is present in certain Delta bronzes. It refines the structures of cast α-β alloys and 1·0 per cent. of iron results in an increase of between 2 and 3 tons per sq. in. in the tensile strength without, however, any serious influence on the ductility. More than 2 per cent. of iron is definitely harmful, particularly in regard to machining properties.

Lead, as in the ductile α-brasses, is often introduced into '60/40' brasses in order to facilitate machining. Up to 0·4 per cent. does not appear to exert much influence on their mechanical properties, but higher proportions are definitely harmful, apart from easier machining.

Aluminium is often added in small amounts to molten brass. Higher proportions improve the corrosion-resistance of both α (70/30) and α-β (60-40) brasses, although it is usually added to alloys of the latter type to increase their tensile strengths. The influence of aluminium on the mechanical properties of brass is indicated in the following tables:—

Composition.			U.T.S. *or* Tensile Strength Tons/sq. in.		Elongation. Per cent.		Proof Stress. Tons/sq. in.	
Cu.	Zn.	Al.	Extruded.	Hard-drawn.	Extruded.	Hard-drawn.	Extruded.	Hard-drawn.
76	22	2	22	40	70	10	6	28

EFFECT OF ALUMINIUM ON CHILL CAST BRASSES.

Composition.			Yield Point. Tons per sq. in.	U.T.S. Tons per sq. in.	Elongation. Per cent. on 2 in.	Reduction of Area. Per cent.	Brinell Hardness.
Copper. Per cent.	Zinc. Per cent.	Aluminium. Per cent.					
58·96	41·04	Nil	8·8	24·9	45·0	49·7	90
59·48	39·52	1·00	14·8	32·0	30·0	33·5	114
58·35	40·11	1·54	16·4	35·2	17·0	18·5	129
58·26	38·56	2·18	16·0	36·4	16·0	21·5	138
59·85	37·13	3·02	22·3	42·0	18·5	21·5	159

(*Smalley.*)

Even in the presence of lead, aluminium improves the properties of 60/40 brass as shown by the following data for an alloy of cast brass containing 38 per cent. of zinc, 2 per cent. of aluminium, and 2 per cent. of lead.

Yield point	18 tons per sq. in.
Tensile strength	34 ,, ,,
Elongation	12 per cent.

Aluminium brass for condenser tubes contains 2 per cent. of aluminium, 22 to 28 per cent. of zinc.

Tin is often added to brasses of the 60/40 type. It greatly improves the corrosion-resistance, notably against sea-water corrosion. Up to 1 per cent. of tin has little influence on the ductility but raises the tensile strength by about 2 to 3 tons per sq. in. Above 1 per cent. the ductility rapidly falls away although the effect is less marked the higher the copper content.

The well-known *Naval brass* has the following composition and properties:—

Cu.	Sn.	Total Impurities.	Zn.	Minimum Tensile Strength (Tns./sq. in.)		Min. Elongation. Per cent.
				Up to ½ in. Bar.	Over ½ in. Bar.	
61 min.	1·0 min.	0·75 max.	Remainder	26	22	20

Naval brass is used for bolts, nuts, etc., for the under-water fittings of ships.

Manganese is invariably present in high-tensile brasses, *e.g.* in the so-called 'manganese bronze' (this is really a brass). It increases the tensile strength without seriously affecting the ductility if less than 1 per cent. is added. By virtue of its powerful deoxidising properties manganese leads to the production of sounder castings, even when only small amounts are added. In the cast state manganese 'bronze' has a tensile strength of between 28 and 33 tons/sq. in., and an elongation of about 30 to 35 per cent.

Nickel in small proportions has little effect on the tensile properties of brass, and higher proportions are usually precluded on account of additional cost. When added with aluminium and/or silicon, nickel brasses may be heat-treated and temper-hardened (precipitation-hardened).

Thus one proprietary alloy contains 72 per cent. of copper, 6 per cent. of nickel, 1·5 per cent. of aluminium and 20·5 per cent. of zinc, and yields the following properties :—

Treatment	Proof Stress (Tns./sq. in.)	U.T.S. *or* Tensile Strength (Tns./sq. in.)	Elongation. Per Cent.
Softened by quenching from 850° C.	5·0	23·0	60
Quenched and reheated to 500° C.	22·0	36·0	30
Cold-worked and then quenched and tempered	45·0	48·0	11

The addition of zinc to copper lowers its conductivity and increases the expansibility. Average **coefficients of thermal expansion per ° C. (range 25–100° C.) are as follows :—**

H.C. copper	0·000017 (17×10^{-6})
80/20 gilding metal	0·000018 (18×10^{-6})
65/35 brass	0·000019 (19×10^{-6})
60/40 brass	0·000020 (20×10^{-6})

An aluminium-nickel-silicon alloy, known as *Tungum*, is covered by the Air Ministry Specification D.T.D. 283 ; copper 81–86 per cent., aluminium 0·7–1·2 per cent., nickel 0·8–1·4 per cent., silicon 0·8–1·3 per cent., impurities not more than 0·5 per cent., zinc the remainder.

This alloy may be cold-worked and heat-treated. It offers marked resistance to corrosion and possesses useful tensile properties.

Iron-Nickel Alloys.

These alloys generally contain 25–35 per cent. nickel. They are austenitic, and not subject to heat-treatment. They are tough and strong, while some have a low thermal expansivity and are very resistant to corrosion in air, fresh and sea water. They may be forged or rolled, but not so readily as ordinary steel.

Average tensile properties in the normal condition (not heat-treated):—

	25–28 per cent. Nickel.	30–35 per cent. Nickel.	35–38 per cent. Nickel.
Tensile strength, tons per sq. in.	38–41	38–42	45–51
Yield point, tons per sq. in.	15·5–22	18–22	28–35
Elongation on 2 in. per cent.	30–35	30–40	25–35
Reduction of area, per cent.	50–60	40–60	50

Alloys containing about 25 per cent. nickel are used for electrical resistance wire in the construction of rheostats and electrical heaters. Within the range of composition from 20–30 per cent. nickel, the ferro-nickel alloys may be readily obtained in a non-magnetic condition by cooling at normal rates from rolling or forging temperatures.

***Permalloy.*—A nickel-iron alloy, nickel 78·5 per cent., iron 21·5 per cent., with impurities as low as possible. Heat-treatment. Heat to 200° C. for 1 hour, cool slowly, reheat to 600° C., cool at a definite rate in air. This alloy possesses high magnetic properties, the maximum permeability is largely in excess of the highest values found for silicon steels. The presence of impurities affects the permeability and carbon is especially harmful, but the effect of changes in the heat-treatment are much greater than those due to small quantities of impurities.**

Typical composition of Permalloy : nickel, 78·23 ; iron, 21·35 ; carbon, 0·04 ; silicon, 0·03 ; manganese, 0·22 ; sulphur, 0·035 ; phosphorus, trace ; cobalt, 0·37 ; copper, 0·10.

μH values for Permalloy : 50 per cent. nickel alloy, 4 per cent. silicon-iron alloy, and pure iron.

Magnetising Force H. Gilberts per sq. cm.	Permeability $\mu = \frac{B}{H}$			
	Permalloy.	50 % Ni.	4 % Si-Fe.	Pure Iron.
0·05	74,000	69,000	5,000	2,400
0·1	50,000	60,000	7,000	3,600
0·2	30,000	39,000	13,000	10,000
0·4	18,000	23,000	15,500	26,500
0·6	13,000	16,500	14,500	21,500
0·8	10,000	13,000	11,000	16,500
1·0	8,000	10,000	10,000	14,000

Mumetal.—A nickel-iron alloy containing copper and manganese. Its initial permeability is somewhat less than that of the binary alloy, but has a higher electrical resistivity which is of value in keeping down eddy-current losses.

The advantage of the nickel-iron alloys are pronounced in very low magnetic fields and are particularly well marked in respect of 'initial permeability'—*i.e.* the permeability in fields whose strength approaches zero. Ternary nickel iron alloys containing such elements as aluminium, cadmium, chromium, cobalt, copper, magnesium, molybdenum, tantalum, titanium, tungsten, vanadium, etc., have been developed to meet special requirements.

In general, the alloys must be made from as pure metals as possible, the presence in the alloy of certain impurities, such as sulphur and carbon, being very detrimental. Heat-treatment is of great importance, also care must be taken to protect metal from stress or work after treatment. An alloy with 81 per cent. nickel is less sensitive in this direction, and the addition of molybdenum also helps.

Permax.—Nickel-iron alloy containing added elements. It does not possess the high permeability of some of the other alloys, but is remarkably constant in certain of its magnetic properties—*e.g.* under fields varying from 200 gauss to 1 gauss, the coercive force changes only from 0·48 to 0·45.

Invar.—See p. 1183.

Elinvar.—An alloy of 36 per cent. nickel and about 12 per cent. chromium (or its equivalent where small quantities of manganese, tungsten, or carbon are used in conjunction with the chromium additions).

At ordinary temperatures has a practically invariable elastic modulus coupled with a very low thermal expansivity.

Newer alloys of this type have been developed by Chevenard, Huguenin, Waché and Villachon. One such alloy has the composition: nickel 40 per cent., aluminium 2 per cent., titanium 2 per cent. It is quenched before drawing, after which it is reheated to 600–650° C., and then has an elastic limit of about 87 tons per sq. in.

Dilver.—Forty per cent. nickel, has a coefficient of thermal expansion approximately that of ordinary glass.

Platinite.—Forty-six per cent. nickel, has a coefficient thermal expansion equal to that of platinum.

A.M.F. Alloy.—Fifty-seven per cent. nickel, has the same thermal expansion as steel but more stable and less liable to corrosion.

Nickel Silver.

The most harmful impurities are lead (except in rods or castings intended for machining, in which case lead up to 3 per cent. is purposely added to improve the machining quality of the alloy), sulphur and carbon, and precautions should be taken that these elements are kept within very narrow limits. Tin, iron, and unreduced oxides may also be present as impurities.

Melting operations are carried out usually in coke-fired pit type crucible furnaces. Electric furnaces are also used, and the metal is cast into iron moulds in forms suitable for rolling sheets or rods. The rolling and drawing of nickel silver is carried out at normal temperatures, since none of the ordinary commercial alloys are capable of being hot-worked.

Annealing temperature: Nickel, less than 14 per cent., 600–650° C.
Nickel, 16 per cent. and upwards, 700–750° C.

Average mechanical properties (nickel 18 per cent. and up) cold-worked (rolled or drawn) and fully annealed. Yield stress, 15 tons per sq. in. Ultimate tensile stress, 26–30 tons per sq. in. Elongation on 2 ins., 35–40 per cent. Reduction of area, 45–50 per cent. With nickel less than 18 per cent. Yield stress, 12–14 tons per sq. in. Ultimate tensile stress, 23–26 tons per sq. in. Elongation on 2 ins., 50–55 per cent. Reduction of area, 50–55 per cent.

The mechanical properties of nickel silver are not as important as other physical properties—*i.e.* whiteness of colour and brilliance of polished surface, resistance to corrosion.

The alloy possessing the finest silvery whiteness is that produced from a specification of approximately equivalent weights of nickel, copper and zinc. This composition is only suitable for castings. It is not so important commercially as the alloy coming next to it known as B.B. and containing 30 per cent. nickel. In general, the colour varies directly with the nickel content, those alloys of highest nickel giving the whitest colour and most lustrous polish.

Results of research show that the tendency of commercial nickel silver to 'burn' is increased (*a*) by increasing the nickel content, (*b*) as the ratio of the zinc to the copper is increased, and (*c*) with the amount of impurities present.

The following alloy is stated to possess the best all-round properties:—

Copper, 46 per cent.; Nickel, 34 per cent.; Zinc, 20 per cent.

Nickel-Chromium Alloys.

These alloys have a composition which varies considerably according to the purpose for which they are required, but in those of commercial utility the nickel content lies within the limits of 60 per cent. and 80 per cent., and the chromium content between 12 per cent. and 22 per cent. The balance consists mainly of iron, with small percentages of manganese, silicon, aluminium, magnesium, and titanium, other elements being present only as traces of impurity.

A typical alloy has the approximate composition of nickel 66 per cent., chromium 20 per cent., iron 10 per cent., the balance consisting of the above-mentioned minor components.

Physical Properties.

Tensile strength in cast state	25 to 30 tons per sq. in.
Melting-point	1300° to 1350° C.
Density	·29 lbs. per cu. in.
Specific heat	·11

These alloys have two outstanding properties which have great commercial value, namely: strength is well maintained at high temperatures, and they are chemically inactive under a wide range of conditions even at high temperatures. Thus they resist the attack of furnace gases to a marked degree, and unless sulphur is present in unusually large quantities, their life at temperatures up to 1200° C. is considerable.

The actual life of articles made of this alloy is greatly dependent upon conditions, but a period approaching 10,000 hours at such temperatures is by no means unusual.

The above properties make the alloys eminently suitable for the manufacture of such articles as pyrometer sheaths, case-hardening boxes, furnace parts, retorts, muffles, in which rôles they have been of great assistance to engineers in overcoming difficulties due to the severe conditions of temperature and corrosion which modern works practice produces.

These alloys when drawn into wire or rolled in strip form, are extremely valuable for such electrical purposes as windings for electric furnaces and stoves.

The alloys are frequently of service to chemical engineers for use with corrosive fluids at normal temperatures, and have found application in the manufacture of such articles as acid pumps and valves subject to attack by acids and acid vapours.

Inconel contains 80 per cent. nickel, 14 per cent. chromium and 6 per cent. of iron and is both heat- and corrosion-resisting. Its tensile strength when annealed is between 35 and 40 tons per sq. in., but may be drawn into wire of 75–80 tons strength. Its coefficient of expansion is 0·0000115 per ° C. (range 40–100° C.).

Heat- and Acid-Resisting Alloys.

The following table gives the compositions of some typical commercial alloys :—

Alloy No.	C.	Mn.	Si.	Cr.	Ni.	Co.	Al.	Cu.	W.	Mo.	Fe.
1	0·94	1·1	1·9	21·0	68·0	—	—	—	—	—	7·0
2	0·27	0·06	3·92	29·1	62·4	—	—	—	—	—	4·9
3	0·09	0·02	0·46	15·6	81·2	1·56	0·42	—	—	—	0·8
4	0·50	0·79	0·80	13·68	57·5	—	—	—	—	—	27·3
5	—	0·98	1·04	21·97	60·7	—	1·09	6·42	2·13	4·7	0·76
6	1·13	2·2	0·63	15·5	64·9	—	—	—	—	—	15·8
7	1·37	1·90	0·49	8·5	40·0	—	—	5·5	—	—	42·0
8	0·60	1·88	1·17	13·2	59·9	—	—	—	—	—	23·2

(*Kayser.*)

Alloy No. 1 has the most general useful composition. Silicon rather higher than necessary. Narrow freezing range; solidification complete about 1325° C.

No. 3 alloy is suitable for the manufacture of wire, but is expensive. The cobalt probably due to impurity in the nickel. Melting range, 1230–1390° C.

No. 4 alloy is rather poor, suitable for resistance wire.

No. 5 alloy, generally known as Parr's Alloy, developed specially for the manufacture of Mahler bombs.

No. 6 alloy is an average alloy. The carbon is high, showing that a low-grade ferro-chrome was used, and the manganese is also unnecessarily high.

No. 7 is a poor type of alloy. Iron too high and nickel very low. The copper should not be added to an alloy intended for general purposes; it is useless in a heat-resisting alloy.

Maximum tensile strength of alloy No. 1 in cast state, 25–30 tons per sq. in.; alloy No 4 in rolled sheets; yield stress, 28 tons per sq. in.; ultimate strength, 50 tons per sq. in. Elongation 8 per cent. Reduction of area, 12 per cent.

Practically all the alloys are capable of being softened and rendered more ductile by heat-treatment, and in that respect behave as many non-ferrous alloys—*i.e.* they become more ductile when quenched.

The corrosion resisting properties vary according to composition, but if the silicon alone does not exceed 2 per cent., and the combined carbon, silicon, and manganese 4 per cent., the properties of these alloys are very similar. Nickel-chromium alloys offer a perfect resistance to the corrosive action of the atmosphere in the presence of either hard or soft water, or sea-water. In the case of stainless steel it is necessary that it be correctly heat-treated and free from surface scale or pit marks.

No heat-treatment is necessary for nickel-chromium alloys. (Kayser.)

Nickel-Barium Alloys.

Alloys of nickel containing up to 0·2 per cent. barium are used for sparking plugs. The addition of barium to nickel increases its resistance to the action of hot corrosive gases, at the same time improving its hardness and tensile strength. The increased thermionic electron emission of the barium nickel is an added advantage for sparking plugs, and this property can be improved further by the addition of 2 to 4 per cent. chromium, or 1·5 to 2 per cent. manganese.

'Adnic' White Nickel Alloy.

This alloy is used for diaphragms for thermostatic steam traps and similar devices. The normal composition is: copper 70, nickel 29, tin 1. It resists corrosion, has good strength at elevated temperatures and is not susceptible to season cracking. It withstands the action of alkalies, hot gases, salt solutions, dilute organic and inorganic acids.

Electric Resistance-Heating Alloys.

Nickel-chromium and nickel-chromium-iron alloys (Nichrome, Brightray, etc.) are commonly employed for this purpose. The best types contain about 80 per cent. of nickel and 20 per cent. of chromium, but the ternary alloys are cheaper. The more important physical properties of such alloys are as follows:—

Composition.			Specific Resistance. (Microhms per cm.³)	Resistivity Coefficient per ° C.	Coefficient of Linear Expansion.	Maximum Temperature for which suitable.*
Ni.	Cr.	Fe.				
80	20	—	105–110	0·000098	0·000015	1150° C.
85	15	—	90–95	0·000254	0·000015	1000° C.
65	10	25	95–105	0·000210	0·000012	825° C.

Newer alloys developed for this purpose are 'Kanthal' and 'Smith Alloy No. 10.' Kanthal contains about 25 per cent. of chromium, 5 per cent. of aluminium, 3 per cent. of cobalt, and the remainder of iron. Its resistivity is about 30 per cent. greater than that of nichrome, and may be used for temperatures as high as 1250° C. Smith No. 10 Alloy contains approximately 37·5 per cent. chromium, 7·5 aluminium, and iron the remainder. Its resistivity is more than 80 per cent. greater than nichrome and it may be used for temperatures up to 1350° C.; on this account the alloy is being used as the heating element in high-speed steel hardening furnaces. Alloys of this type should not be heated in contact with siliceous refractories, but should be wound on alundum.

Platinite.

Platinite is an alloy of iron, nickel, and carbon, which is used extensively for scientific instruments for standard measurements of length because of its peculiar quality of being practically non-expansive when heated to high temperatures. It is also used in incandescent electric lamps where the wire connections fused into the glass must have so small a coefficient of expansion as not to cause fracture in the glass. This alloy contains about 46 per cent. of nickel and 54 per cent. of iron, with a very small percentage (0·15 per cent.) of carbon.

Monel Metal.†

This is a natural nickel alloy, and which, after refining, has an average composition of nickel 67 per cent., copper 28 per cent., iron 3 per cent., manganese 2 per cent.; a small percentage of carbon may also be present. In appearance it is very similar to pure nickel and takes the same high finish, but has a slightly softer and somewhat more silvery lustre. It is as strong, tough, and ductile as steel, and under corroding influences far superior to copper, gun-metal, and bronze, It machines readily, and can be rolled, drawn, cast, forged, soldered, brazed, and welded.

* The maximum temperature depends on the section of the wire or tape, so that the above values are approximate only. Thicker wires withstand high temperatures better than thin wires but the latter are bent and twisted more easily.

† Registered Trade Mark (*Henry Wiggin & Co., Ltd.*).

Average tensile properties:

	U. T. S. *or* Tensile Strength Tns./sq. in.	Yield Point. Tns./sq. in.	Elongation per cent. on 4 $\sqrt{\text{area}}$.	Brinell No.
Hot-rolled metal .	34–38	15–18	35	120–140
Cold-drawn bars .	40–45	35–40	18–20	190–210
Cold-drawn and annealed . . .	30–35	14–17	35	110–120
Cold-rolled strip . .	45–50	40–45	15	—
C.R. strip annealed .	29–30	14–16	30	—
Cold-drawn wire for springs . . .	55–60	50–55	5–10	—
Ditto, and annealed .	29–33	14–16	35	—
Castings (as cast) .	19 23	12–15	12	110–130

Monel metal resists sea-water, alkalies, and some dilute acids, but is not suitable where nitric acid is present. One of the most valuable properties is its tensile strength at elevated temperatures. For example, at a temperature of 400° C., a forged rod has a tensile strength over 35 tons (see page 1186).

Average Physical Properties.

Melting point, 1,350° C. (2,460° F.).
Pouring temperature of castings, 1,500° C.
Forging temperature, 1,040–1,100° C.
Annealing temperature, 700–900° C.
Specific gravity, 8·82.
Weight per cub. in. = 0·323 lb.
Shrinkage allowance for castings, ¼ in. per foot.
Coefficient of expansion, 0·000014 per ° C. (25–100° C.); 0·000015 per ° C (25–300° C.).
Electrical resistivity, 48 microhms per cm.³
Electrical conductivity, 3·4–5 per cent. of copper.
Heat conductivity, ⅛ that of commercial copper.
Elastic modulus, in tension, 25–26 × 10^6 lbs. per sq. in.
in torsion, 8–9 × 10^6 lbs. per sq. in.

Monel metal in the rolled condition is magnetic at ordinary temperatures, but becomes non-magnetic when heated from 100–150° C. The magnetic change is reversible, the metal becoming magnetic on cooling. Hysteresis tests show that under a magnetising force (H) of 100 C.G.S. units the number of lines per cm.² (B) is 2,460. The remanence is found to be 990, and the coercive force 1·1.

Casting.—Melting practice, also gates and risers similar to that used for steel. The care required for moulding and ramming more nearly resembles that for non-ferrous metals. Cores must be collapsible to allow free shrinkage of the metal.

Metal can be melted in graphite crucibles, using (*a*) oil-fired pit furnaces; (*b*) oil-fired clay brick-lined reverberatory furnaces; or (*c*) in clay brick-lined or basic-lined electric arc furnaces. All ladles and crucibles used for transferring the metal should be preheated to conserve the heat in the metal.

Forging.—Practice similar to that for steel. The metal is harder under the hammer than mild steel and approaches closer to the nickel steels in behaviour. Oxide scale formed is strongly adherent and will not free itself even under hammer blows with steam and salt. Metal becomes hot-short just above maximum forging temperature.

Annealing should be carried out in a reducing atmosphere. General practice is to anneal in a cast-iron box covered with fine charcoal and a cover sealed with fireclay.

Silicon Monel Metal.

Prepared by the addition of approximately 2·75 per cent. of silicon to remelted monel metal. Centrifugal castings made from this alloy are sound and free from defects.

The following properties may be expected from sand castings:—

Silicon.	2·75 per cent.	3·75 per cent.
Tensile strength, tons per sq. in. .	38	45
Yield point, tons per sq. in. .	23	40
Elongation on 2 ins. . .	16 per cent.	5 per cent.
Brinell number . . .	210	270

Standard Monel metal cannot be hardened by thermal treatment. Its mechanical properties can, however, be considerably enhanced by cold working, and for certain purposes it is definitely advantageous to employ it in a hard rolled or hard drawn condition.

'K' Monel Metal.

Heat-treatable alloys have been developed by the addition of between 3 and 5 per cent. of aluminium to Monel metal. An alloy of this type, made by H. Wiggin & Co., Ltd., is described as 'K' Monel metal and has the following properties:—

Density	8·58
Weight, lb. per cubic in.	0·31
Melting point	1315/1345° C.
Specific Heat (20–400°)	0·127

Coefficient of Expansion—
- 25–100° C. 0·000014 per 1° C.
- 25–300° C. 0·000015 ,, ,,
- 25–600° C. 0·000016 ,, ,,

Magnetic transformation.—Below minus 79° C.

Elastic modulus—
- in tension . . 26,000,000 lbs. per sq. in.
- in torsion . . 9,500,000 ,, ,,

Mechanical Properties at Ordinary Temperatures.

Condition.	Ultimate Strength. Tons/sq. in.	Yield Point. Tons/sq.in.	Elongation Per cent. on 2 in.	Izod. Ft./lb.	Brinell.
Hot-rolled and softened	39	19	35	100	140
Hot-rolled, softened and thermally hardened	60	43	30	70	270
Cold-worked and thermally hardened	72	60	15	50	320

Mechanical Properties at Elevated Temperatures.

Temperature. ° C.	U. T. S. *or* Tensile Strength Tons/sq. in.	Yield Point. Tons/sq. in.	Proportionality Limit. Tons/sq. in.	Elongation. Per cent. on 2 in.	Reduction in Area. Per cent.
25	73·6	55·8	46·9	21·0	38·9
95	72·8	55·4	46·4	21·0	37·0
205	69·6	52·7	44·6	20·0	35·0
315	65·9	48·5	37·5	19·5	32·8
425	55·8	47·3	36·3	18·5	29·8
540	55·6	45·9	31·5	9·5	9·8

To soften 'K' Monel metal it is heated to about 800° C., soaked, and then quenched in water or oil. To harden it is then reheated to 580°–590° C. from 4–8 hours and then slowly cooled. The alloy is hot-rolled or hot-forged at temperatures between 950° and 1150° C. It must be softened before cold-working.

When heat-treated 'K' Monel metal has a fatigue limit of more than ± 17·5 tons per sq. in. (10^7 reversals) and its creep-strength at elevated temperatures is relatively high.

Inconel.

This alloy, also developed by H. Wiggin & Co., Ltd., is a useful corrosion- and heat-resisting alloy, and can be mechanically worked, either hot or cold. Its approximate composition is: nickel 80 per cent.; chromium 14 per cent.; iron 6 per cent.

Forgings will yield the following mechanical properties: proof stress (0·5 per cent.), 15–22 tons per sq. in.; ultimate tensile stress 38–45 tons per sq. in.; elongation per cent. (2 ins.), 40–25; reduction of area 70–50 per cent.; Brinell hardness number (3,000 Kgm. load), 125–180. In forging, the range 650°–875° C. must be avoided. Best forging range is between 1,000° and 1,250° C., but light work at temperatures between 650° and 400° C. results in improved strength.

Annealing is carried out at 980° C. In this condition has the following properties: yield stress 15–20 tons per sq. in.; ultimate tensile strength 35–40 tons per sq. in.; elongation per cent. 55–35.

In the hard-drawn or cold-drawn condition will give 40–60 tons per sq. in., with 30–20 per cent. elongation.

The alloy is much more resistant to attack by sulphurous gases than pure nickel.

COPPER AND TIN ALLOYS. (See page 1250.)

Gun-metal.

The alloys of copper and tin containing upwards of 86 per cent. of copper are possessed of high tenacity and elasticity. Those containing from 89 to 91 per cent. of copper are the most important and are known as 'gun-metals.'

The addition of tin to these bronzes results in an increase in the hardness of the resulting alloy. Gun-metal is rarely manufactured from specially selected metals and frequently contains lead, zinc, etc. The Admiralty specification for gun-metal is as follows:—

Copper	88 per cent.
Tin	10 ,,
Zinc	2 ,,

Average mechanical properties being:—

U.T.S. or Tensile Strength	17 tons per sq. in.
Elongation	20 per cent. on 2 ins.
Brinell number	65.

Bell-metal.

The alloys of copper and tin containing from 75 to 86 per cent. of copper are used in the manufacture of bells; iron, zinc and lead are frequently added to the alloys either for cheapness or variety of tone.

Ductile Bronzes.

Copper-tin alloys containing upwards of 93 per cent. of copper consist of a single solid solution and, after annealing, may be cold-worked in the same way as the α brasses. To ensure soundness and freedom from tin oxide these bronzes are almost always deoxidised by small proportions of phosphorus added as phosphor-copper. Phosphorus-deoxidised bronzes are made into rods, bars, sheet and strip and are used for coinage. In the hard-rolled and hard-drawn condition they have found considerable engineering application. A bronze of this type, after annealing and cold-working, may have a tensile strength of 25–40 tons per sq. in.

Effect of Certain Elements on Bronze.

ZINC, LEAD AND PHOSPHORUS.

Gun-metal, Bearing Metal, etc.

The alloys of copper, tin and zinc are rarely, if ever, free from elements such as lead, iron, etc. These alloys may be divided into two classes:—

(i) Alloys containing from 7 to 12 per cent. of tin, up to 6 per cent. of zinc, and up to 2·5 per cent. of lead. These alloys, among which may be noted gun-metal to the Admiralty specification, are used in the manufacture of castings, such as cocks, stuffing-boxes and pump barrels, and for bearings which are subject to but little friction.

(ii) Alloys containing from 12 to 20 per cent. of tin, up to 5 per cent., though rarely more than 2·5 per cent., of zinc and up to 5 per cent. of lead. These alloys are used in the manufacture of heavy bearings, *e.g.* railway wagon and locomotive axle boxes. The majority of alloys of this class contain from 14 to 16 per cent. of tin and from 1·5 to 2·5 per cent. of zinc.

Zinc combines with the oxides and free oxygen of the bronzes, producing greater fluidity in the molten state, and hence freedom from 'pinholes' in the solid state. Bearings rich in zinc have a poor wear value and induce a tendency to seizing and should only be used with low speeds.

Copper-lead alloys (known as 'lead-bronzes') containing 25–30 per cent. lead are used for high-duty bearings. These are not alloys in the true sense but are really mixtures of copper and lead, and the chief manufacturing difficulty lies in preventing segregation of the lead. This is overcome by centrifugally casting, but where this is not possible, small additions of tin, zinc or nickel are added to inhibit segregation. The addition of 1–2 per cent. of nickel also slightly increases the strength and toughness. One specification for an alloy containing 30 per cent. lead includes 0·4–0·7 per cent. of silver.

Although the Brinell hardness of lead-bronze is low(about 30) it permits of about 20 per cent. higher loading than white-metal bearing alloys, but to avoid spreading under load only a thin layer (0·06–0·02 in.) is generally employed. The thermal conductivity of lead-bronze is about half that of high-conductivity copper, as against 10 to 20 per cent. of that of copper for other bearing bronzes.

Phosphor Bronze.

Typical compositions and properties are as follows:—

Nominal Percentage Composition by Weight.					Average Mechanical Properties.			Description and Applications.	Equivalent British Standards Institution or Air Ministry Specifications.
Copper	Tin.	Lead	Zinc	Phosphorus.*	Tensile Strength. Tons/sq. in.	Elongation. Per cent. on 2 in.	Brinell Hardness.		
85	15	—	—	(0·1)	14	2	100	Hard-wearing bronze suitable for heavy compressive loads. Employed for locomotive slide valves, bearings for turntables, etc.	
89	10 min.	—	—	(0·5 min.)	18	4	100	*Phosphor - bronze*, suitable for heavy loading, and very widely employed.	2B8
88	10	—	2	—	17	20	65	*Admiralty Gun-metal.* A bronze for general casting purposes, especially to resist marine corrosion. Suitable for bearings when lubrication is good.	383
80	10	10	—	(0·05)	15	15	65	Possesses good anti-friction properties combined with plasticity. May be applied where lubrication is doubtful.	
77	8	15	—	(0·05)	14	15	60	Suitable for use where lubrication or alignment is still less satisfactory than for the above.	
85	5	5	5	—	13	16	55	An alloy suitable for general castings, such as hydraulic fittings not requiring high strength. Only occasionally used for bearings but suitable for bearing shells.	
74	1·2 max.	25	—	—	8	15	30	This alloy has high thermal conductivity and is capable of carrying higher loads at high speeds than 'white metals' and is therefore used for high-duty aeroplane and other engine crankshaft bearings, etc. Special technique in casting is required.	D.T.D. 229

(*Copper Development Association.*)

* Phosphorus content varies widely and values given are typical only.

Bronzes of the above types are used for bearing shells which carry thin linings of 'white-metal' bearing alloys. Castings for bearings should never be annealed.

Phosphorus acts as a deoxidiser and is added in small quantities for the purpose of deoxidation to copper. Not more than 0·15 per cent. is allowable.

The term 'phosphor bronze' is applied to the alloys grouped below.

(i) Bronzes to which phosphorus has been added as a deoxidiser and containing either no phosphorus or merely traces of the element. These are better described as *phosphorus deoxidised bronzes.*

(ii) Bronzes containing less than 9 per cent. of tin and up to 0·25 per cent. of phosphorus. These alloys are forgeable and are relatively non-corrodible. They are employed for rolling and drawing into wire and rod, and in the manufacture of general castings; in the latter case lead is occasionally added in quantities up to 5 per cent. The forgeable alloys of this group generally contain less then 5 per cent. of tin.

(iii) Bronzes containing from 8 to 14 per cent. and from 0·5 to 1·0 per cent. of phosphorus. These alloys are employed for casting purposes in the manufacture of machine parts likely to be exposed to great friction. Lead up to 5 per cent. is added in certain cases.

Phosphorus is added to these alloys in the form of phosphor copper or phosphor tin.

Silicon Bronze.

Typical analysis: copper 94 per cent., silicon 4 per cent., zinc 2 per cent.

Physical properties:—

Tensile strength (Cast alloys) . .	16 to 22 tons per sq. in.
Yield point	7 to 10 tons per sq. in.
Elongation	10 to 25 per cent.
Modulus of elasticity	$1 \cdot 8 \times 10^6$ lbs. per sq. in.
Coefficient of expansion between 15–400° C.	$1 \cdot 8 \times 10^6$ lbs. per sq. in.
Brinell No.	100 to 120.
Specific gravity	8·4.
Melting point	950° C. (approx.).

Shrinkage approximately midway between that of gun-metal and manganese bronze.

Melting Points of Commercial Bronzes.

(*Gillett & Norton.*)

	Copper.	Tin.	Lead. (as desired).	Zinc.	Copper.	Tin.	Lead. (by analysis).	Zinc.	No. of Tests.	Melting Point.
Gun-metal . .	88	10	—	2	—	—	—	—	4	995° C.
Leaded gun-metal	85·5	9·5	3	2	85·4	9·7	3·0	1·9	6	980° C.
Leaded bronze .	80	10	10	—	—	—	—	—	3	945° C.
Bronze with zinc .	85	10	—	5	84·6	19·4	—	5·0	4	980° C.

COPPER AND ALUMINIUM ALLOYS.

Aluminium Bronze.

The alloys known as the 'aluminium bronzes' are solid solutions of copper and aluminium containing not less than 88 per cent. of copper. Of these alloys that containing 10 per cent. of aluminium is the strongest, though the alloys containing less than this percentage are of industrial importance. These 'bronzes' are of use, firstly, on account of their valuable mechanical properties and, secondly, on account of their resistance to corrosion. They are malleable both when hot and when cold; the alloys containing between 8 and 12 per cent. of aluminium are affected in a marked degree by heat-treatment.

Following are tabulated the mechanical properties of some of these alloys:—

Percentage of Aluminium.	Yield Point. Tons/sq. in.	U.T.S. Tons/sq. in.	Elongation. Per cent. on 4 in.	Remarks.
2·5	—	20·0	40·0	Cast.
5·0	—	25·0	55·0	Cast.
10·0	18·0	35·0	25·0	Cast in chill.
10·0	18·0	46·0	25·0	Cast rod reduced 25 per cent. by forging hot. Annealed.
10·0	15·0	40·0	30·0	Rolled bar.
10·0	40·0	50·0	2·0	Rolled bar quenched at 900° C.

The following are tests of the 'R3' and 'R4' aluminium bronzes manufactured by the British Aluminium Company :—

R3 Bronze.—Average breaking load, 34·04 tons; average elongation, 21 per cent.; maximum breaking load, 37·6 tons; maximum elongation, 35 per cent.; minimum breaking load, 32·8 tons; minimum elongation, 12·5 per cent.

R4 Bronze.—Average breaking load, 41·8 tons; average elongation, 9·75 per cent.; maximum breaking load, 43·7 tons; maximum elongation, 13 per cent.; minimum breaking load, 39·8 tons; minimum elongation, 6·5 per cent.

Aluminium Bronze Die Castings.

Alloys suitable for die casting may contain anything from 2 to 16 per cent. of aluminium, with small percentages of iron, manganese, or nickel, the balance copper; the majority of alloys in use contain from 5 to 11 per cent. aluminium. With less than 7 per cent. Al the alloy displays characteristics more resembling copper, while with Al exceeding 12 per cent. the alloy develops brittle properties. Alloys containing 7 per cent. Al have a similar colour to 18-carat gold.

Alloys containing 10–12 per cent. Al have been found more resistant to attack by dilute sulphurous acid than the more complex compositions.

An aluminium bronze containing about 10 per cent. Al and 1·0–1·5 per cent. of nickel is used for forgings for exhaust valve seats. Hardened by quenching in water from about 850° C. and tempering at the optimum temperature to a Brinell hardness of 190–230.

COPPER AND MANGANESE ALLOYS.

Manganese Bronze, Manganin.

The term 'manganese bronze' is applied to copper-zinc alloys of 60/40 composition to which small quantities of manganese have been added. There are, however, alloys of copper and manganese of industrial application which rightly should be admitted under this heading. The alloys of copper and manganese containing up to 5 per cent. of the latter element are harder and tougher than is copper. Those containing from 2 to 4 per cent. of manganese are frequently employed in situations where high temperatures are existent, on account of the retention of their mechanical properties under these conditions.

The following figures represent the results obtainable by stamping certain of these alloys at a red-heat (800° C.) :—

Copper.	Manganese.	Tensile Strength.	Elongation. Per cent.
97	3	18·5	32
96	4	21·0	60

Manganin is the name applied to an alloy of copper, manganese, nickel and iron, characterised by low electrical conductivity and almost zero temperature coefficient. It is employed in the construction of electrical resistance coils. The composition is somewhat variable, but the following figures will serve to indicate its approximate analysis :—

Copper . .	80 to 84 per cent.	Nickel . . .	up to 12 per cent.
Manganese .	4 to 15 ,,	Iron . . .	Difference.

COPPER AND NICKEL ALLOYS.

Cupro-nickel, Constantan, etc.

The alloys of copper and nickel containing upwards of 50 per cent. of copper are of industrial importance. The alloys may be divided into three classes :—

(i) Alloys containing up to 5 per cent. of nickel and having the property of resisting high temperatures without undue deterioration. Of these alloys those containing from 2 to 3 per cent. of nickel have been employed in the manufacture of locomotive firebox tubes. Following are the mechanical properties of the 2 per cent. alloy :—

Tensile Strength.	Elongation. Per cent.	Remarks.
45·0	5·0	Hard rolled.
30·0	45·0	Hard rolled and annealed at 650° C.

Those of the 5 per cent. alloy are very similar in character. Low nickel alloys are used for armature slip-rings.

(ii) Alloys containing from 15 to 25 per cent. of nickel are valuable on account of their ability to withstand cold working.

For the best results with these alloys only the purest metals should be employed, as small quantities of such elements as iron, silicon and sulphur are detrimental to the malleability and

ductility of the resultant alloy. The maximum percentages of impurities allowable in nickel for this high-class work should be 0·050 per cent. of arsenic, 2 per cent. of cobalt, and 1·5 per cent. of other metallic impurities. The metal should be free from all unreduced oxides. Cupro-manganese, 25–50 per cent., is commonly employed as a deoxidiser in the manufacture of these nickel copper alloys, upwards of 0·25 per cent. being added to charges containing 30 per cent. of scrap. Following are the mechanical properties of the 20 per cent. alloy :—

Tensile Strength.	Elongation. Per cent.	Remarks.
40·0	5·0	Hard rolled.
20·0	35·0	Hard rolled and annealed.

The alloy containing 25 per cent. of nickel has been successfully employed in the manufacture of firebox plates and for condenser tubes.

(iii) Alloys having low electrical conductivities and whose temperature coefficients of electrical conductivity are practically nil. The 40–45 per cent. nickel alloy, known as *constantan*, can be drawn into wire, and is employed in the construction of resistance coils and base-metal thermo-couples. Another alloy of this type, containing only 25 per cent. of nickel and 0·5 per cent. of iron, is known as *nickeline*.

All the above alloys can be hardened by cold-work only, but newer copper-nickel alloys containing aluminium are heat-treatable, as shown by the following data.

EFFECT OF HEAT-TREATMENT ON A NICKEL-COPPER ALLOY CONTAINING 30 PER CENT. NICKEL WITH THE ADDITION OF 1·5 PER CENT. ALUMINIUM.

Treatment.	Maximum Stress. Tons/sq. in.	Limit of Proportionality. Tons/sq. in.	Elongation. Per cent. on 2 in.	Reduction of Area. Per cent.	Brinell Hardness Number.
Water-quenched .	26·9	3·2	45·0	65	90
Water-quenched and re-heated . .	48·8	18·0	32·0	43	184
Water-quenched, cold-drawn, 25 per cent. reduction in area and re-heated . .	58·1	31·2	16·0	38	210

(*Bureau of Information on Nickel.*)

Copper and Silicon.

Alloys of copper and silicon are used for electrical purposes and in the manufacture of telephone and telegraph wires. The effect of silicon is to increase the tensile strength, but even 0·20 per cent. of silicon reduces the electrical conductivity by nearly 50 per cent.

COPPER-CADMIUM ALLOYS.

The addition of cadmium to copper increases its tensile strength, hardness and resistance to wear without much influence, however, on its electrical conductivity. The conductivity of an alloy containing 1·0 per cent. Cd is more than 90 per cent. of that of pure copper. Cadmium-copper is thus most suitable for overhead cables, tramway trolley wires and other purposes where strength or resistance to wear combined with excellent conductivity is desired. It is a solid solution alloy so that it is hardened by cold-work, and softened by annealing, although it is to some extent heat-treatable. Air Ministry Specification D.T.D. 208 gives the following details: Cadmium 0·8–1·2 per cent., total impurities less than 0·2 per cent, copper the remainder, and the following minimum physical properties :—

Proof stress (0·1 %): 25 tons/sq. in. min. Maximum stress: 35 tons/sq. in. min.

Torsion test (wires)—No. of turns = 25 per length of 100 times diameter. Bend test (strip)—180° over radius equal to twice the thickness.

Electrical resistivity at 15 5° C. (60° F.) not greater than 2·25 microhms per cm.[3]

COPPER-BERYLLIUM ALLOYS.

Small amounts of beryllium are sometimes used to deoxidise copper. Larger amounts, up to 2·5 per cent., render the alloys heat-treatable and extraordinarily high hardness and strengths may then be obtained. When properly heat-treated their strengths are comparable with those of high-tensile steel. A 2·5 per cent. beryllium-copper alloy may yield, when suitably cold-worked and heat-treated, a tensile strength of 90–95 tons/sq. in. with elongation of 1–2 per cent. To soften these alloys are quenched from a temperature of about 800° C. In this condition they are readily cold-worked and are then temper-hardened by reheating to temperatures between

250° and 300° C. The following data, furnished by the Anaconda Copper Co., are representative of the properties of Anaconda Beryllium-Copper wire.

Condition.	Tensile Strength. Tons/sq. in.	Young's Modulus. Lbs./sq. in.	Elongation. Per cent. on 2 ins.	Approximate Brinell Number.
Annealed soft . .	34·5	$17 \cdot 5 \times 10^6$	39·8	155
Annealed soft then heat-treated . . .	78·1	$18 \cdot 0 \times 10^6$	3·6	363
Hard drawn . .	61·0	$16 \cdot 8 \times 10^6$	2·0	281
Hard drawn then heat-treated . . .	81·1	$18 \cdot 5 \times 10^6$	2·0	375

The elastic properties and fatigue resistance are exceptionally high so that beryllium-copper is particularly suitable for springs, either in the form of wire or strip. For springs it is infinitely superior to phosphor-bronze.

Beryllium-copper is also used for castings which, after heat-treatment, have tensile-strengths of the order of 50–55 tons/sq. in.

The corrosion-resisting properties of these alloys are about the same as those of copper; they have a pleasing colour, and their electrical and thermal conductivities are relatively high although inferior to the copper-cadmium alloys.

ALUMINIUM-ZINC ALLOYS.

The alloys of aluminium and zinc are of importance as light alloys, their specific gravities varying uniformly from 2·7 to 3·3 as their zinc content increases up to 30 per cent., which may be considered the limiting percentage for those alloys of this series that may be termed 'light.' All those alloys containing upwards of 75 per cent. of aluminium are forgeable at temperatures between 300° C. and 400° C.

These alloys do not allow of their use in casting into intricate shapes, but for all plain casting purposes they are quite satisfactory. Under certain conditions these alloys corrode rapidly; the use of high-grade spelter in their manufacture eliminates this defect. The machining properties of these alloys which are, at all times, superior to those of the pure metal, improve as the zinc content increases.

Below are tabulated certain of the results of tests reported by Messrs. Rosenhain and Archbutt to the Alloys Research Committee of the Institution of Mechanical Engineers.

Zinc. Approx. Per cent.	Condition of Alloy.	Yield Point. Tons per sq. in.	Max. Stress. Tons per sq. in.	Elongation. Per cent. on 2 ins.
5	1	2·70	5·20	16·0
	2	2·80	6·60	29·0
	4	4·30	7·48	33·0
11	1	6·40	9·41	8·0
	2	5·00	10·24	16·0
	3	9·42	13·78	33·0 (on 1 in.)
15	1	9·60	11·14	2·0
	2	5·80	11·69	8·5
	3	11·61	17·92	31·0 (on 1 in.)
	4	6·80	16·38	33·0
20	1	10·00	13·07	0·7
	2	7·80	13·87	4·0
	4	17·30	22·64	20·5
25	1	nil	16·26	1·0
	2	9·10	13·50	3·5
	3	20·24	24·03	27·0 (on 1 in.)
	4	25·00	27·09	16·5

1. Sand-casting.
2. Chill-casting.
3. Hot-rolled bars, ½ in. diam.
4. ditto 1½ in. diam.

The straight Al–Zn alloys are subject to hot-shortness and high shrinkage and thus tend to crack after casting.

The addition of copper to aluminium-zinc alloys improves their casting and mechanical properties. Alloy 2L. 5 is a well-known casting alloy used for gear-box and crank-case castings, etc. This alloy contains 2·5–3·0 per cent. copper and 13·0–14·0 per cent. zinc, and typical mechanical properties are given on p. 1237.

ALLOYS HAVING A ZINC BASE.

Zinc-base alloys are widely employed in the production of die-castings, and particularly for pressure-castings. They have superior casting properties to the aluminium-base alloys, and are readily nickel or chromium plated, enamelled or lacquered in order to improve their surfaces. These alloys, developed in the U.S.A., are manufactured in England by National Alloys, Ltd., under the proprietary name 'Mazak Alloys.' Typical compositions and properties are given in the following table :—

	Mazak 3.	Mazak 5.
Aluminium	4·1	4·1
Copper	—	1·0
Magnesium	0·04	0·03
Zinc	Remainder	Remainder.
Ultimate tensile	15–17	18–19
Elongation (on 2 ins.)	2–5	3–6
Brinell Numeral	62	73

It has been shown that even traces of tin, cadmium and lead adversely influence the properties of these alloys, so that zinc of high-purity must be used as the base.

Typical properties are given in the following tables :

Tensile Strength (Tons per Sq. In.).

Temp.	Mazak 3.	Mazak 5.
20° C.	18·2	22·5
0° C.	19·2	24·2
−40° C.	20·5	24·2

Elongation (Per Cent. on 2 Ins.).

Temp.	Mazak 3	Mazak 5.
20° C.	11	8
0° C.	9·5	8·5
−40° C.	4·5	3·5

Impact Strength (Ft./Lb.).

Figures Average of 20 Tests.

Temp.	Mazak 3.	Mazak 5.
20° C.	42	44
10° C.	31	41
0° C.	7·6	39
−10° C.	3·3	18·3
−20° C.	2·5	3·6
−40° C.	2·1	2·4

Property.	Mazak 3.	Mazak 5.
Specific gravity	6·6	6·7
Weight of 1 cub. in./lb.	0·238	0·242
Melting point ° C.	380·9	380·6
Melting point ° F.	717·6	717·1
Thermal expansion per ° C.	$27·4 \times 10^{-6}$	$27·4 \times 10^{-6}$
Thermal expansion ° F.	$15·2 \times 10^{-6}$	$15·2 \times 10^{-}$
Electrical conductivity Phos/Cm. cube at 20° C.	157,000	153,000
Solidification shrinkage ins./ft.	0·14	0·14

The Mazak alloys have good casting properties, together with relatively high impact strengths at normal temperatures. For this reason designers incline to reduce the whole thickness of castings to the greatest possible extent.

It must be borne in mind, however, that wherever these castings are likely to undergo shock loading at sub-normal temperatures, those sections of the castings liable to be severely stressed should have thicker walls.

ALUMINIUM-COPPER ALLOYS.

Like the zinc alloys, the commercial alloys are not simple binary alloys, but generally contain iron and silicon, and may also contain Mg, Mn, Ni, Zn, etc.

Al.	Cu.	Use.
93–91·5	7–8·5	General castings.
92	8	General castings, tendency to be brittle.
96	4	General castings, less brittle than 92 : 8 alloy ; suitable for rolled sheets, sand-cast cooking utensils.
91–89·5	9–10·5	Light alloy pistons } American automobile practice.
89–86·5	11–13·5	Manifolds } American automobile practice.
90	10	Die castings. Permanent mould castings.
88	12	Die castings. Permanent mould castings.

The specific gravity increases with increase of copper from 2·7–2·85 for sand-cast alloys for 0–8 per cent. copper. The specific gravity of sand-cast alloys is less than that of chill castings, and that of rolled and drawn alloys is approximately equal to chill cast. In sand-cast alloys the strength increases with increase of copper at the expense of the elongation. In the hot-rolled condition the tensile strength increases up to about 6 per cent. copper and then decreases. The physical properties of these alloys are indicated by the following figures :—

92–8 alloy. Ultimate tensile strength, 18,000 lbs. per sq. in. ; elongation, 1–1·5 per cent.

97–3 alloy (forged). Yield point, 16,000 lbs. per sq. in. ; ultimate strength, 24,000 lbs. per sq. in. ; elongation, 4 per cent.

96–4 alloy (rolled, annealed). Ultimate tensile strength, 15,000–28,000 lbs. per sq. in. elongation, 4–16 per cent.

Tensile strength of hot-rolled bars, 21,000–37,000 lbs. per sq. in. (for 1–6 per cent. copper); elongation varies from 30–18 per cent. For cold-drawn bars—tensile strength, 29,000–45,000 lbs. per sq. in. (for 1–4 per cent. copper) ; elongation, 14·5–6 per cent.

Well-known alloys of this type for castings are 3L. 8 and 4L. 11. Typical mechanical test values for these alloys are given on p. 1196, and their compositions are as follows :—

	Cu.	Fe.	Zn.	Si.	Ti.	Pb.	Al.	Sn.
3L. 8	11–13	<0·8	<0·1	<0·7	<0·2	<0·1	Remainder	—
4L. 11	6–8	<0·8	<0·1	<0·7	<0·2	<0·1	,,	<1·0

An alloy of German origin, described as *Ultralumin*, has the approximate composition : 4·7 per cent. copper, 0·2 per cent. nickel, and traces of thorium and cerium, having a specific gravity of 2·2. The tensile strength and elongation are given as 24 tons/sq. in. and 14–24 per cent. respectively.

Aeral **is an aluminium-copper alloy of French origin containing 2·0 to 2·5 per cent. copper, 0·5–2·5 per cent. cadmium, 0·2–1·5 per cent. magnesium, 0–2·5 per cent. silicon, and small proportions of manganese, iron and titanium. Sand-castings have the following properties : maximum stress 14–20 tons/sq. in., elongation 3–8 per cent. The alloy is heat-treatable.**

Designation.	Equivalent D.T.D. Specification.	Composition. Si.	Cu.	Mg.	Fe.	Ti.	Mn.	Zn.	Condition.	Maximum Stress. Tons/sq. in.	Elongation. Per cent.	Brinell Number.
					Max.	Max.	Max.	Max.				
N.A. 125/67 .	276	4·5/5·5	1·0/1·5	0·4/0·6	0·6	0·25	—	—	Q. 515–527° C. T^d 150–160° C.	15 min.	—	—
N.A. 125/W. 60	272	4·5/5·5	1·0/1·5	0·4/0·6	0·6	0·25	—	—	W.Q. 515–527° C.	11 ,,	2·0 min.	—
M.V.C. . .	231	8·0/12·0	—	—	0·6	0·2	0·5	0·1	Modified (a) sand-cast	9–11	5–10	40
									(b) chill-cast	11–13	7–11	—
Alpax Beta .	240	10·0/13·0	—	Max. 0·6	0·6	—	0·6	—	Sand-cast and heated to 150–170° C.	11–13	5–10	55–60
Alpax Gamma	245	10·0/13·0	—	0·6	0·6	—	0·6	—	W.Q. 520–535° C. T^d 150–170° C.	15·5 min.	—	—

Designation.	Equivalent D.T.D. Specification.	Mg.	Mn.	Si.	Fe.	Condition.	Proof Stress (0·1 %) Tons/sq. in.	Maximum Stress. Tons/sq.in.	Elongation. Per cent.	Specific Gravity.
			Max.	Max.	Max.		Min.	Min.	Min.	
BSS.–L. 31 .	303	4·5/5·5	0·6	0·5	0·5	Rolled and annealed	—	21	—	—
' Birmabright ' .	165	3·0/6·0	0·25/0·75	—	0·6	Castings	—	9	3	2·68
,, .	170	3·0/6·0	0·25/0·75	—	0·6	Hard-rolled	15	20	—	2·68
,, .	180	3·0/6·0	0·25/0·75	—	0·6	Rolled and annealed	6	14	—	2·68
M.G. 7 . .	194	6·5/7·5	0·6	—	0·75	Rolled bar	10–15 *	21–25 *	15	2·63
N.A. 350 and B.A./59	300	9·5/10·5	—	0·25	0·30	Sand-cast and heat-treated	—	16	7	—

* According to thickness of section.

ALUMINIUM-SILICON ALLOYS.

A striking property of these alloys is their low contraction, due to the silicon expanding on freezing. Yield exceptionally sound sand castings. Iron should be a minimum, as it has a detrimental effect. These alloys may be forged and both hot and cold formed, and, in their raw condition, are very malleable at the higher temperatures. In addition, they may be welded and will resist all ordinary corrosion, being unaffected by atmospheric conditions. The strength is largely dependent on the actual alloy used, but, in general, is somewhat higher than that of cast iron, and somewhat lower than that of brass. Much higher strengths are possible in the case of 'modified' alloys. Modification consists in treating the molten metal, prior to casting, with sodium, potassium, calcium, the fluorides of these metals, or other of the patented modifying agents. This results in a remarkable refinement of structure and enhanced mechanical properties. Both tenacity and ductility are improved by modification. The alloys most amenable to this treatment contain 10–13 per cent. silicon and are known as *Alpax* alloys. Their specific gravities are between 2·6 and 2·7.

All the Al-Si alloys are amenable to heat-treatment after casting. Typical compositions are shown in table on page 1231.

OTHER ALLOYS OF ALUMINIUM.

Magnalium.

Magnalium is the name applied to a series of alloys having an aluminium-magnesium base, containing from 1·5 to 10·0 per cent. of magnesium and varying proportions of other elements, *e.g.* copper, tin, nickel, lead.

The table below gives the compositions of certain of these alloys, together with the uses to which they are generally put.

Use of Alloy.	Magnesium.	Copper.	Nickel.	Tin.	Lead.
Strong castings .	1·60	1·76	1·16	Nil.	Nil.
General ditto . .	1·60	1·76	nil	traces	traces
Rolling and drawing .	1·58	0·21	nil	3·15	0·72

Other commercial alloys of this class together with compositions and properties are shown in the table on page 1231.

ALUMINIUM-MANGANESE ALLOYS.

Manganese increases the hardness, strength and ductility of aluminium. Alloys containing 1·0–1·5 per cent. Mn are employed for the manufacture of sheet and strip. Their corrosion-resisting properties are similar to those of pure aluminium. The following properties are typical of 1·0 to 1·5 per cent. Mn alloys.

Condition.	U.T.S. *or* Tensile Strength.	Elongation.	Brinell Number.	Specific Gravity.
	Tons/sq. in.	Per cent.	(500 kgrm.)	
Annealed at 410/420° C.	7	30	28	2·7/2·72
Hard-rolled . . .	13	4	55	2·71/2·73

Aluminium-manganese alloys containing from 20–25 per cent. Mn are used for certain kinds of castings.

Aluminium-Magnesium-Manganese Alloy.

This alloy, which contains about 1 per cent. Mg and 1 per cent. Mn, is finding many applications. It possesses unusual combination of strength and corrosion-resisting properties. It is not susceptible to heat-treatment, but is hardened by cold working. It presents no difficulty in fabrication. Compressive strength and compressive yield point are similar to the values obtained in tension. The endurance limit is exceptionally high (6–7 tons per sq. in., depending upon the temper).

Condition.	Tensile Strength. Tons per sq. in.	Yield Point.	Elong. on 2 ins. Per cent.	Brinell No.	Shear Strength. Tons per sq. in.
Annealed . .	11·6	4·5	20	45	7·1
¼ hard . .	13·8	11·1	6	55	7·6
½ hard . .	15·6	13·3	5	65	8·5
¾ hard . .	17·4	15·6	3	73	9·4
Hard . .	18·7	17·0	3	80	9·8

'Duralumin.'

'Duralumin' is the trade name applied to a series of alloys of aluminium containing small proportions of copper, manganese, magnesium, and silicon, and over 94 per cent. of aluminium. These alloys are possessed of great tenacity and hardness, and are also very ductile. They attain their maximum properties after a heat-treatment, *i.e.* they are quenched from a suitable temperature (about 485° C.), and after this quenching operation they increase spontaneously in strength over a period and reach their maximum after several days. Ageing is accelerated by boiling in water.

The following table gives the range of composition of these alloys :—

Copper. Per cent.	Manganese. Per cent.	Magnesium. Per cent.	Silicon. Per cent.	Iron. Per cent.	Aluminium. Per cent.
3·5–4·8	0·4–0·7	0·4–0·7	0·40	0·40	Difference.

The specific gravity is about 2·8. The alloy has a melting range starting at 545° C., but it is not considered wise to subject it to temperatures higher than 500° C. It may be satisfactorily annealed for forming or cold-working processes at 380° C., after which the maximum mechanical properties may again be obtained by the specified heat treatment.

The alloy may be rolled, forged, drawn, and spun, and may be worked either hot or cold, and is supplied in sheets, tubes, bars, wires, strips, forgings, stampings, extruded sections and rivets.

By suitable modifications of the composition the alloy may be adapted for many purposes, but the following table shows the mechanical properties of this alloy as normally supplied by the manufacturers :—

0·1 per cent. Proof Stress. Ton per sq. in.	Ultimate Tensile Strength. Ton per sq. in.	Elongation per cent. on 2 ins.	Brinell Hardness.
16–18	25–28	15–20	100

Young's Modulus.—10 × 10^6 lbs./sq. in.
Impact Value.—15 ft. lbs.
Fatigue Range.—± 9·5 tons per sq. in.

AGE HARDENING OF DURALUMIN.

The rate at which Duralumin will age-harden is controlled to a large extent by the temperature immediately following quenching. It has been found (J. O. Lyst) :—

(1) Duralumin aged at room temperature after quenching began to age-harden appreciably 1½ hours after quenching.
(2) Duralumin stored at 0° C. immediately after quenching began to age-harden appreciably about 36 hours after quenching.
(3) Duralumin stored at − 48° C. does not age-harden appreciably over a period of 14 days.
(4) Duralumin stored for 14 days at − 48° C. and then removed and allowed to age at room temperature, shows approximately the same behaviour as Duralumin stored at room temperature immediately after quenching.

Advantage is taken of this property in practice by storing Duralumin rivets in refrigerators immediately after quenching.

HIDUMINIUM R.R. ALLOYS.

These alloys have been developed by Rolls Royce, Ltd. They can be forged, drop-stamped, pressed, rolled, or extruded.

Alloy.	Cu.	Ni.	Mg.	Fe.	Si.	Ti.	Al.
RR 50	1·30	1·3	0·1	1·0	2·2	0·18	Remainder.
RR 53	2·25	1·3	1·6	1·4	1·25	0·10	"
RR 56	2·00	1·3	0·8	1·4	0·70	0·10	"
RR 59	2·25	1·3	1·6	1·4	0·50	0·10	"

RR 50 is used for sand and die castings for general purposes; particularly suited for gearboxes, differential and back-axle casings, or similar parts where rigidity is essential.

RR 53 retains its strength at elevated temperatures. Used for die cast pistons. A modified form of this alloy (Si 2 per cent.) can be used for very delicate die castings.

RR 56 suitable for general purposes forgings.

RR 59 forging alloy. Specially used for high-quality pistons in aero and large Diesel engines.

Alloys RR 56 and 59 should be preheated to 500 to 520° C. and held at this temperature before forging or rolling. Forging should commence at 480 to 520° C., and continue to a minimum of 350° C. In stamping it is important that the correct dummy size be formed so that the die is filled just prior to the formation of the 'flash.' Cold work can be applied to these alloys provided they are in a hot-worked condition, and have been softened by annealing at 350 to 380° C. For heat-treatment they are re-heated to 510°–535° C., soaked for 2–4 hours, and then quenched in water, followed by tempering at 155°–175° C. for 20 hours, after which they are again water-quenched.

The chief feature of these alloys is their iron, titanium and nickel content.

Titanium refines the grain and reduces oxidation at high temperatures. The iron appears to prevent crystal growth on ageing. R.R. alloys do not age-harden so readily at ordinary temperatures as duralumin and higher artificial ageing temperatures are permissible. This allows of greater latitude in straightening, etc., after quenching.

Typical properties of RR 56 and RR 59 are as follows:—

Alloy.	—	0·1 Per cent. Proof Stress. Tons per sq. in.	Yield Point. Tons per sq. in.	Maximum Tensile Stress. Tons per sq. in.	Elongation on 2 ins. Per cent.	R/A. Per cent.	Brinell.
'RR' 56	Forging material heat-treated	23–25	24·5–27	28–32	10–20	14–25	121–160
'RR' 59	Forged piston alloy heat-treated	19–22	22–24	23–29	6–10	10–20	120–150

The alloys retain their strengths quite well at elevated temperatures. Thus at 200° C. the tensile strength of RR 59 is 21–22 tons per sq. in., at 300° C. it is 13 tons per sq. in., and 8 tons per sq. in. at 350° C.

The coefficient of thermal expansion of RR 59 is given as 22×10^{-6} per ° C. (range 20–100° C.) and its thermal conductivity as 0·428 C.G.S. units.

Heat Treatment and Properties of Aluminium Alloys.

The mechanical properties of a number of aluminium alloys are greatly improved by suitable heat treatment.

Age-hardening.—Increase of hardness in the course of time, which can be explained by the 'Precipitation Theory.'

> 'Age-hardening can occur in an alloy where some constituent is more soluble at a high temperature than at a lower temperature and where rapid cooling fixes this constituent in supersaturated solid solution. Being supersaturated and metastable at lower temperatures, there is a natural tendency—slowly at ordinary temperature and more rapidly if the temperature is slightly raised—to precipitate out of solid solution some of the dissolved constituent. Precipitation of the excess constituent occurs in the form of submicroscopic particles distributed throughout the mass of the solid solution, and this causes the hardening.'
> (*N. F. Budgen.*)

Certain aluminium alloys age 'naturally,' *i.e.* at room temperatures if given sufficient time after solution treatment; others can only be 'artificially' aged by re-heating to temperatures above atmospheric.

For example, alloys with Mg_2Si as the soluble constituent: '*Silmalec,*' Mg 0·6, Mn 0·6, Si 1·0. '*Aludur,*' Mg 0·6, Si 0·88.

Soluble constituent $CuAl_2$. '*Lautal*': Cu 4·7, Mn 0·5, Si 1·0–2·0; U.S.A. '25 S': Cu 4·0, Mn 1·0, Si 0·8.

Soluble constituent $Mg_2Si + CuAl_2$: '*Duralumin,*' '*Y*' *Alloy*; '*Avional*': Cu 4·75, Mg 0·5, Mn 1·0, Si 1·4; 'RR 56' and 'RR 59.'

Age-hardening can occur in wrought and cast alloys, but for similar treatment, cast alloys give lower mechanical properties.

'*Solution*' *Heat-treatment.*—Temperature depends on composition of alloy. Duration of heating depends on mass of article and must be such that all of the soluble constituents of the alloy go into solid solution before quenching. Rapid heating is recommended for solution-treatment, but it must not be forgotten that the material is quite soft during this operation and care must be taken to prevent the articles warping.

Close temperature control is essential. With too low a temperature the full mechanical properties are not obtained; if the temperature is too high overheating or burning may occur with loss of both ductility and tensile strength when neither can be restored by subsequent treatment. Molten sodium nitrate baths are now commonly employed for heating.

Quenching is necessary to prevent the decomposition of the solid solution. Cold water quench is best for alloys subject to corrosion. Boiling water may be used, but air cooling is generally too slow.

Alloy.	Temperature for Solution Heat-treatment. ° C.	Temperature for Precipitation Treatment (Ageing). ° C.
Duralumin (wrought)	510	15 for 4 days.
Super-duralumin (wrought)	495	155 „ 20 hours.
4–5 per cent. Cu alloy (wrought)	520	140 „ 10 „
„ „ „ „ „ (cast)	515	155 „ 10 „
Magnesium-silicon alloy	520	155 „ 20 „

Die Quenching.—Chilling produced by casting in moulds is sometimes sufficient to prevent decomposition of the solid solution. This principle is used in production of die-cast pistons. The castings will age-harden to a certain extent at room temperature, but by accelerating the ageing process, considerable increase of hardness and strength is obtained.

Stabilising Treatment.—Pistons are subjected to additional treatment to prevent warping, growth, or change of hardness in use. This treatment, which consists of heating to about 230° C. for 20 hours, may or may not be preceded by a solution treatment.

COMPARISON OF PROPERTIES OF WORK-HARDENING AND PRECIPITATION-HARDENING OF COPPER-NICKEL-ALUMINIUM ALLOYS.

—	Work-hardening Alloy. Cu 92, Ni 4, Al 4.	Precipitation Hardening. Cu 91, Ni 7·5, Al 1·5.
Melting point. ° C.	1,090	1,120
Density. Lb. per cub. in.	0·302	0·315
Mean coefficient of expansion. Per ° C.	$1 \cdot 7 \times 10^{-5}$	$1 \cdot 7 \times 10^{-5}$
Tensile strength. Lb. per sq. in.	45,000–120,000	48,000–110,000
Yield point. Lb. per sq. in.	20,000–80,000	30,000–80,000
Elongation on 2 ins. Per cent.	60–12	50–10
Reduction of area. Per cent.	80–50	80–35
Elastic modulus. Lb. per sq. in.	$21–18 \cdot 5 \times 10^6$	$21–18 \times 10^6$
Hardness, Rockwell B.	98–15	95–10

Both alloys have excellent corrosion resisting properties, and may be welded, brazed or soldered. The work-hardening alloy has found considerable application for condenser tubes and for a number of structural purposes.

'*Lautal.*'—Aluminium alloy with 4–5 per cent. Cu and 2 per cent. Si. This alloy does not age-harden appreciably at room temperature, and may be stored in the quenched condition and aged after it has been fabricated, by treating in an oil bath.

Mechanical properties similar to those of Duralumin, but the elastic properties are somewhat lower. The resistance to corrosion is also not so good, but can be improved by anodic oxidation.

Tensile strength for sheets and wire: 22–24 tons per sq. in., with 18–25 per cent. elongation. For forgings, tensile strength 24–26 tons per sq. in; 10–18 per cent. elongation.

'*L. IV.*' *Alloy.*—Cast variety of Lautal alloy.

'*Allautal.*'—Lautal coated with pure aluminium, similar to 'Alclad,' which is Duralumin coated with aluminium.

'*Silmalec.*'—Aluminium-magnesium-silicon alloy Mg 0·6, Mn 0·6, Si 1·0.

Solution heat-treatment: heat to 520° C. and quench. Age for about 15 hours at 150° C.

For wire, the material is drawn between the quenching and ageing operations.

This class of alloy is used for telegraph wires. Electrical conductivity, 80 per cent. pure aluminium, but tensile strength nearly double that of aluminium.

Annealed 345° C. Tensile strength 7 tons per sq. in with 30 per cent. elongation.

Quenched from 520° C., aged at 150° C. Tensile strength 21·5 tons per sq. in. with 14 per cent. elongation.

'*Duralumin.*'—British Standards Institution specification (3 L 1) for bar. For composition, see page 1233.

Mechanical Properties.

Nominal Size of Bar (diameter or width across flats) from which test bar was taken.	0·1 per cent. Proof Stress. Tons per sq. in.	Ultimate Tensile Stress. Tons per sq. in.	Elongation per cent.	R/A. per cent
		Not less than	Not less than	Not less than
Up to 2⅝ ins.	15	25	15	20
2⅝ „ 4 ins.	12	22	15	20
4 „ 6 „	10	20	15	20
Untreated material . .	—	17	15	18

Heat-treatment.—Heat in salt bath to 480° C. ± 5° C. and when part has attained the temperature of the bath it should be quenched in water, or oil, and aged 4 days.

On no account should the temperature exceed 490° C. or the material is embrittled.

Annealing.—Aluminium alloys are generally softest when annealed at temperatures below their solution treatment temperatures, *e.g.* duralumin is softest when annealed at 380°–400° C. Subsequent cooling may be in either air or water.

For sheets (Spec. 3 L 3) similar treatment, but the mechanical requirements are as follows:—

Thickness of Sheet.	0·1 per cent. Proof Stress. Tons per sq. in.	Ultimate Tensile Strength. Tons per sq. in.	Elongation on 2 ins. Per cent.
	Not less than	Not less than	Not less than
Below 0·02 in. (25 S.W.G.) . .	15	25	8
From 0·02 to 0·048 in. (25 to 18 S.W.G.)	15	25	12
From 0·048 in. (18 S.W.G.) and thicker	15	25	15

Drop Forging.—Working temperature, 470°–500° C. Dies used for steel stamping may be used for Duralumin. The use of oil during stamping is not necessary, but the dies should be highly finished.

Super-duralumin.—Duralumin containing 0·8 to 1·25 Si. Has superior mechanical properties to normal Duralumin, especially after accelerated ageing.

Treatment for Duralumin: heat to 490° C. for sufficient time for the soluble constituents to be taken into solid solution; *e.g.* ½ hour for sections less than ½ in. thick, quench in cold water and age at room temperature for about 4 days, or in boiling water for 5 hours. With normal Duralumin, ageing above room temperature tends to lower the mechanical properties, but with Super-duralumin the hardness and tensile strength are raised. If material that has been age-hardened at room temperature is cold-rolled and aged at about 100° C., the yield point is raised considerably.

'*Y*' *Alloy.*—Used as a casting alloy but can be extruded, hot and cold worked. Normal composition: Cu 4, Ni 2, Mg 1·5, balance aluminium.

For cold working: (*a*) heat to 500°–520° C. and quench. Work immediately. (*b*) Soften by annealing 350°–400° C. followed by air cooling.

For forging: Preheat at 500° C. for about 6 hours; commence forging at 480°–500° C., finish at 350° C.

There is a tendency for this alloy to crack under the hammer.

B.S.I. specification for wrought heat-treated 'Y' Alloy :—

Composition.	Per cent.
Copper	between 3·5 and 4·5
Nickel	„ 1·8 „ 2·3
Magnesium	„ 1·2 „ 1·7
Aluminium	Remainder.
With impurities : Iron	Not more than 0·75
Silicon	„ „ „ 0·60
Lead, zinc and tin	Not more than a total of 0·10 per cent.

Mechanical Properties.—In the form of the British Standard test-piece, 0·564 in. diam., material is required to give :—

Nominal Size of Bar (diameter or width across flats).	0·1 per cent. Proof Stress. Tons per sq. in.	Ultimate Tensile Strength. Tons per sq. in.	Elongation per cent.	R/A. per cent.
		Not less than	Not less than	Not less than
Up to 2½ ins.	13	22	15	25
2½ to 4 ins.	10	20	15	25
4 „ 6 „	10	17	15	25
Untreated material	—	14	15	25

Heat-treatment.—Heat to 490°–520° C. for not less than ½ hour and quench in boiling water, followed by ageing for 5 days, or the ageing can be accelerated by heating at 100° C. for 1 hour.

'*Magnalite*.'—Alloy similar to 'Y' Alloy. Approximate composition: Cu 2, Ni 1·5, Mg 1, Si 0·6.

CASTING ALLOYS OF ALUMINIUM.

Alloy No. 3 *L.* 5.—High zinc content, 12·5–14·5 per cent.; Cu 2·5–3·0; possesses good casting and machining properties. Susceptible to corrosion and hot shortness. Except for simple shapes should not be used for gravity die-casting.

Typical Physical Properties

	Sand.	Chill.
Ultimate tensile strength. Tons per sq. in.	10·5	12
Elongation on 2 ins. Per cent.	3	5
Brinell (500 kg. — 10 mm. ball)	55	65

Alloy 3 *L.* 8 (11–13 per cent. Cu). Satisfactory for general gravity die-casting. Machining qualities excellent.

	Sand.	Chill.
Ultimate tensile strength. Tons per sq. in.	8·5	11
Elongation on 2 ins. Per cent.	Nil	1
Brinell	70	80

Alloy 4, *L.* 11. (6–8 per cent. Cu.)—Possesses greater ductility and shock resistance than Alloy 3 L. 8. Used for motor manifolds, pump parts, etc.

	Sand.	Chill.
Ultimate tensile strength. Tons per sq. in.	9	11
Elongation on 2 ins. Per cent.	2	3
Brinell	65	70

Alloy 2 *L.* 33 (10–13 per cent. Si).—Has excellent casting qualities and resistance to corrosion. Suitable for aircraft and seaplane castings. Not heat-treated but molten metal is 'modified' by addition of sodium, calcium fluoride or other agent.

	Sand.
Ultimate tensile strength. Tons per sq. in.	10·5 min.
0·1 per cent. proof stress. Tons per sq. in.	3·5 „
Elongation on 2 ins. Per cent.	5·0 „

Alloy L. 35 ('Y' alloy')—3·5–4·5 per cent. Cu; 1·2–1·7 per cent. Mg. and 1·8–2·3 per cent. Ni.—Alloy suitable for pistons. High tensile strength and hardness at temperatures up to 400° C. Castings are heat-treated.

	Sand.
Ultimate tensile strength. Tons per sq. in.	14 min.

Alloy Loex (12 per cent. Si, 2·5 per cent. Ni, 1 per cent. Mg, 1 per cent. Cu, 0·5 per cent. Fe).—Special piston alloy with low expansion.

	Sand-cast.	Chill.
Ultimate tensile strength. Tons per sq. in.	10·5–11·5	16·0–17·5
Elongation on 2 ins. Per cent.	0·0– 0·5	0·0– 0·5
Brinell	120–140	120–140

Alloy D.T.D. 300 (N.A. 350), developed by the Northern Aluminium Co., Ltd., is typical of alloys whose properties are substantially improved by solution and precipitation treatments. It is essentially a binary alloy of aluminium and magnesium (9·5–10·5 per cent.) and is widely employed for stressed aircraft castings since, after solution heat-treatment, it gives the best combination of tensile, ductility and impact properties of any aluminium-base casting alloy, together with a lower density.

Typical properties for heat-treated sand castings are: 0·1 per cent. proof-stress, 11–14 tons per sq. in.; ultimate tensile strength 18–22 tons per sq. in.; elongation (2 in.), 12–14 per cent. The alloy has a wide freezing range so that control of casting temperature is important.

Extruded Sections

(Aluminium Union, Ltd.)

Limits of Manufacture of Extruded Sections.

Alloy.	Minimum Thickness.	Minimum Area.	Minimum Wt. per ft.	Maximum Area.	Maximum Length.	Maximum Dimension.
						Ins.
Pure aluminium	0·040	0·059	0·07	11·4	24	5
3 S.	0·040	0·059	0·07	9·4	24	5
51 SQ.	0·080	0·129	0·15	4·7	24	5
4 S.	0·125	0·213	0·25	6·1	24	5
17 ST.	0·187	0·330	0·40	2·4	13·5	3·75
43 S.	0·187	0·174	0·20	7·3	24	5
No. 1 Turning rod	0·250	0·200	0·24	4·0	24	5

The economic lengths in which the extruded sections can be supplied in order to avoid excessive scrap loss can be determined by dividing the weight of the billet (approximately 40 lbs.) by the weight per foot of the section.

Typical Physical Properties of Extruded Sections.

Alloy.	Yield Stress. Tons per sq. in.	Ultimate Tensile Strength Tons per sq. in.	Elongation Per cent.	Brinell Hardness.
Pure aluminium	3	5–6	37	25
3 S.	3·5	7	38	31
51 SQ.	7·5	14	27	60
4 S.	6	12	24	48
17 ST.	15	23	23	100
No. 1 Turning rod	—	12	25	47

Details of Alloys.

Alloy 3 S.—Commercial pure aluminium with the addition of 1·25 per cent. manganese.

Alloy 51 SQ.—Extrusions of this alloy are specially adapted for architectural purposes, where resistance to corrosion is important.

Alloy 4 S.—Alloy 3 S. plus 1 per cent. magnesium.

Alloy 17 ST.—Properties similar to those of Duralumin.

Anodic Oxidation.—The normal resistance to corrosion of aluminium and its alloys is due to a protective coating of oxide. The natural film is exceedingly thin, however, and processes are available whereby the natural coating is reinforced by a thicker and more strongly adherent film. Improved resistance to corrosion is then obtained. This is usually accomplished by electrochemical means. The articles to be treated constitute the anode when immersed in a suitable electrolyte solution. The well-known *Bengough-Stuart* process uses a 3 per cent. solution of chromic acid in water at a temperature of 40° C. Owing to the non-metallic nature of the coating the resistance gradually increases so that the voltage must be altered as the process continues. The following procedure is recommended: During the first 15 mins. the voltage is gradually increased from 0 to 40, maintained constant at 40 for the next 35 mins., gradually raised to 50 volts during the ensuing 5 mins., and finally maintained at 50 volts for 5 mins. Immediately after anodising the parts are thoroughly washed with cold and then hot water, and dried. The current consumption is about 0·2 kW.H. per sq. ft. of surface treated.

In the *Eloxal* process (German) the electrolyte is oxalic acid plus some other material such as chromium salts, whilst the *Aluminite* (American) process uses dilute sulphuric acid containing glycerine, etc.

The thickness of the film produced by anodic oxidation is about 40 times that of the natural film. The coating is sub-microscopically porous, but the pores can be closed by greases and waxes such as lanoline and beeswax, thus further improving the resistance to corrosion. Alternatively, the pores may be closed and the coating coloured by immersion immediately after rinsing in inorganic or organic dyestuffs.

Most of the light aluminium alloys can be successfully anodically oxidised but difficulty is encountered with alloys containing more than 5 per cent. of copper and alloys relatively rich in silicon.

Aluminium solders cannot be treated satisfactorily, but welded and riveted joints present no difficulty.

Magnesium alloys have also been successfully anodised. Buzzard and Wilson recommend an electrolyte containing 10 per cent. of sodium dichromate and 2 to 5 per cent. of monosodium dihydrogen phosphate, maintained at a p_H value of 4·0–4·8, operating at 5 to 10 amps. per sq. ft. at 50° C.

In all cases the articles must be thoroughly cleaned before being subjected to anodic oxidation. Aluminium and its alloys may be cleaned with 20 per cent. caustic soda solution, and magnesium alloys cleaned electrolytically in sodium carbonate-sodium phosphate solution.

After anodising still better results are obtained by painting and enamelling.

Aluminium Paint.—Protection from corrosion may be improved by painting articles with an emulsion of the following composition: boiled linseed oil 850 parts, turpentine 50 parts, and aluminium powder 300 parts. (All parts by weight.)

Alclad is a composite alloy having a rolled-on coating of pure aluminium to alloys of the duralumin type. The properties of the latter are then retained but the corrosion resistance is improved. The process has been extended to other alloys, now covered by well-known D.T.D. specifications.

MAGNESIUM ALLOYS.*

It would appear from X-ray experiments that the mechanical properties of magnesium and its alloys are intimately connected with lattice distortion. It is possible to produce work hardening in these alloys accompanied by improved mechanical properties, but cold working to the extent of causing lattice distortion at once ruins the material.

Binary alloys of cadmium and magnesium are soft and ductile, the addition of cadmium not materially altering the ultimate strength. The addition of aluminium, up to about 6 per cent., increases the ductility and strength. The addition of zinc and cadmium produces alloys of moderate strength coupled with high ductility. Alloys of magnesium, cadmium and zinc have valuable properties at normal temperatures, but are weak above 300° C.

Magnesium and its alloys are very sensitive to rolling conditions.

(*W. E. Prytherch.*)

* See also p. 1231.

Elektron.—The well-known 'Elektron' or 'Magnuminium' alloys cover a wide range of compositions and are produced as castings, forgings, extruded sections, etc. Details are as follows:—

COMPOSITIONS OF ELEKTRON ALLOYS.

Designation.	D.T.D. Specification.	Al.	Zn.	Mn.	Impurities.
Castings.					
		Max.	Max.	Max.	Max.
AZG	59A	8·5	3·5	0·5	1·7
AM503	140A	0·2	0·2	2·5	0·5
			Cu 0·2 max.	Si 0·4 max.	
AZ31	59A	8·5	3·5	0·5	1·7
AZ91	136A	9–11	3·5	0·5	1·5
Wrought Alloys.					
		Max.	Max.	Max.	Max.
AZM	259	11·0	1·5	1·0	1·5
AM503	118	0·2	0·2	2·5	0·5
A4	120A	9·0	1·5		
			Cu 0·3 max.	Si 0·4 max.	
AZ855	—	7·5–8·5	0·4–0·55	0·15–0·25	

Physical Properties (Castings).

Designation.	Condition.	0·1 per cent. Proof-Stress. Tons per Sq. In.	Tensile Strength Tons per Sq. In.	Elongation. Per Cent.	Brinell No.
AZG	As cast	4·5–5·5	9–11	3–5	50–60
	Solution heat-treated	4·5–5·5	14–16	8–10	45–55
AM503	As cast	1·5	6–7	3–5	35–45
AZ31	"	3·0–4·0	9–11	3–10	40–50
AZ91	"	4·5–5·5	8–10	1–3	55–65
	Heat-treated	7–9	15–17	1–3	75–85
Physical Properties (Wrought Alloys).					
AZM	Extruded	9–12	18–22	16–12	55–60
	Forged	8–10	18–20	12–10	65–70
AM503	Rolled sheets	6–8	12–15	10–3	—
	Extruded	8–9	15–17	10–3	40–50
A4	Rolled sheets	7–9	16–18	12–10	—
AZ855	Forged	11–14	18–22	15–8	65–75

The specific gravities of the above alloys are about 1·80 only with coefficients of linear expansion of the order of 0·000025 per 1° C.

As previously indicated magnesium alloys may be anodically oxidized to improve resistance to corrosion.

Newer improved magnesium alloys have been developed by the National Physical Laboratory. Typical compositions given by Desch and Haughton are 8 Al, 8 Cd and 2 Ag; 8 Al and 8 Cd; 8·5 Al and 3 per cent. Ag, together with 0·3–0·4 Mn and 0·2–0·3 per cent. Ca.

Wrought alloys of this type after solution heat-treatment and age-hardening have given the following results: proof-stress (0·1 per cent.) 16·5–17·5 tons per sq. in., maximum stress 25–26 tons per sq. in., and elongation 4–6 per cent.

For aircraft pistons an alloy containing 10 cerium, 1·5 cobalt, and 1·5 per cent. manganese has given excellent results.

Lithium and Lithium Alloys.

Lithium is the lightest metal known, sp. gr. 0·53. It is a silvery-white metal resembling sodium, melting at 186° C., and burning at higher temperatures. Lithium-calcium alloys have been used to deoxidise molten copper and bronze. It also increases the strength and corrosion resistance of magnesium. Has great affinity for the metalloids and readily absorbs sulphur, phosphorus and gas inclusions from molten metal.

Small additions of lithium to aluminium alloys increase their hardness. With complex aluminium alloys with magnesium, copper, lead and zinc, it accelerates ageing, increases the tensile strength and hardness, and facilitates working.

Alloys of lithium and beryllium, such as 25 per cent. Li, 75 per cent. Be, and 65 per cent. Li, and 35 per cent. Be, are extremely light, sp. gr. 1·0–1·5, but are at present still in the experimental stage.

The German alloy 'Bahnmetal,' which is used for bearings, contains a small percentage of lithium. The approximate composition of the alloy is: 98·6 lead, 0·6 sodium, 0·7 calcium, 0·04 lithium.

The fluidity of molten cast iron is improved by the addition of lithium. It also produces sounder and denser castings with improved physical properties. It also improves the machinability. The addition of lithium to a 0·6 per cent. carbon steel increased the elastic limit and ductility without increasing the hardness.

Beryllium Alloys.

(See pp. 1208 and 1227.)

Beryllium has similar hardness and mechanical properties to pure iron. Alloys of beryllium with magnesium and aluminium have been developed with success. Special beryllium-copper alloys are finding considerable application for instrument and watch balance springs. These alloys can be cold rolled up to about 2·5 per cent. beryllium, beyond which it is only possible to hot roll.

To soften, the alloy should be quenched from 800° C. To harden, heat to 300° C. for 2 hours for softened material, and 275° C. for cold worked material.

The addition of 0·025 per cent. phosphorus to a 2·5 per cent. beryllium copper alloy gives a maximum hardness of 360 Brinell when aged at 350° C. in 1 hour. Without the phosphorus this hardness is not attained under 7 hours. Copper-beryllium springs are non-magnetic and resistant to corrosion.

Strength-Weight Factors of Aircraft Materials.*

Material.	Ultimate Strength. Lbs. per sq. in.	S.G.	Strength-weight factor.
Wrought metals in tension.			
Piano wire 0·01 in. diameter	400,000	7·85	51
Alloy steel, heat treated (high)	200,000	7·85	25
" " " " (medium)	150,000	7·85	19
Duralumin	55,000	2·85	19
Magnesium alloy	34,000	1·75	19
Alloy steel, normalised	85,000	7·85	12
Mild steel, normalised	55,000	7·85	7
Aluminium, annealed	12,000	2·7	4·4
Wood in compression.			
Balsa	2,200	0·12	18
Douglas fir	6,000	0·54	11
Spruce	4,300	0·43	10
Oak, white	5,900	0·74	8

$$\text{Strength-weight factor} = \frac{\text{Ultimate strength (lbs. per sq. in.)}}{\text{Specific gravity} \times 1{,}000}.$$

* H. T. Kerr, American Society for Testing Materials.

MODULUS OF ELASTICITY (E) AT ORDINARY TEMPERATURE AND − 40° C.

Material.	At 15° C.	At − 40° C.
Chrome-molybdenum steel	30×10^6 lbs. per sq. in.	$30 \cdot 5 \times 10^6$ lbs. per sq. in.
Chrome nickel steel	29×10^6 „ „ „ „	$37 \cdot 0 \times 10^6$ „ „ „ „
„ „ annealed	30×10^6 „ „ „ „	$37 \cdot 5 \times 10^6$ „ „ „ „
„ „ cold rolled	28×10^6 „ „ „ „	$36 \cdot 5 \times 10^6$ „ „ „ „
Aluminium alloy	10×10^6 „ „ „ „	$10 \cdot 0 \times 10^6$ „ „ „ „
Magnesium	$6 \cdot 4 \times 10^6$ „ „ „ „	$6 \cdot 3 \times 10^6$ „ „ „ „

Tests carried out at liquid air temperature (− 182° C.) indicate that the general effect is an increase of tenacity, but at the temperature of liquid hydrogen (− 252·8° C.) the number of metals that show an increase in tenacity is greatly reduced. Nickel and copper show a considerable increase in ductility at liquid air temperature, while the ductility of copper still further increases from 45 to 60 per cent. in liquid hydrogen.

STRENGTH OF METALS AT LOW TEMPERATURES.

The physical properties of engineering metals at low temperatures (−40° C.) are such that the designer is safe in using the allowable stresses determined by testing material at normal temperature. The resistance to fatigue, or a suddenly applied load on a part with a sharp change of section, is not reduced except in the case of steels. Greaves and Jones have shown that carbon steels are affected by cold to a much greater extent than nickel steels, but consider that Ni–Cr–Mo steels are affected to the least extent.

Lead.

Lead is one of the softest, but one of the heaviest of all metals. It is easily cut and worked and possesses several other valuable and characteristic properties. Its Brinell number is between 3·2 and 4·5 (1 cm. ball, 100 kgrm. load). It is generally regarded as self-annealing, *i.e.* after deformation it almost immediately recovers its initial softness. *Tellurium-lead* containing 0·05 to 0·10 per cent. Te, is amenable to work-hardening; its strength is then much greater than that of ordinary lead. Ternary alloys of lead containing 0·25 per cent. cadmium and 0·5 per cent. antimony, or 0·5 per cent. cadmium and 1·5 per cent. tin are used for pipes. Since these alloys are also stronger than lead a lesser thickness of the pipe wall is permissible. Lead exhibits a marked resistance to corrosion and attack by certain acids, notably sulphuric acid, although it is readily attacked by some organic acids such as acetic acid.

The coefficient of linear expansion—0·0000293 per 1° C. in the range 17°–100° C. is relatively high.

Other important physical properties of lead are as follows :—

Pure lead :	Fatigue limit	± 403 lbs. per sq. in.
Lead containing 0·25 Cd, 0·5 Sb:	„ „	± 1,658 „ „ „

(*Beckinsdale and Waterhouse.*)

Tensile strength . . 2,000–2,400 lbs. per sq. in. with an extension rate of 0·2 inch/inch/min.

Alloys of Lead.—Lead is a constituent of white-metal bearing alloys, fusible alloys, leaded-bronzes, and many other important alloys. Lead containing about 0·5 per cent. of antimony is used for cable sheathing and an alloy containing 10 per cent. of antimony possesses enhanced acid-resisting properties.

BEARING ALLOYS.*

The requisite properties of good bearing alloys are hardness, to resist wear; strength, to bear compression; and plasticity, to allow of ready adjustment of the supported body on the bearing.

Two types of bearing alloys are known :—

(*a*) Bearing alloys in which the hard constituents of the alloy are distributed indiscriminately throughout a soft matrix.

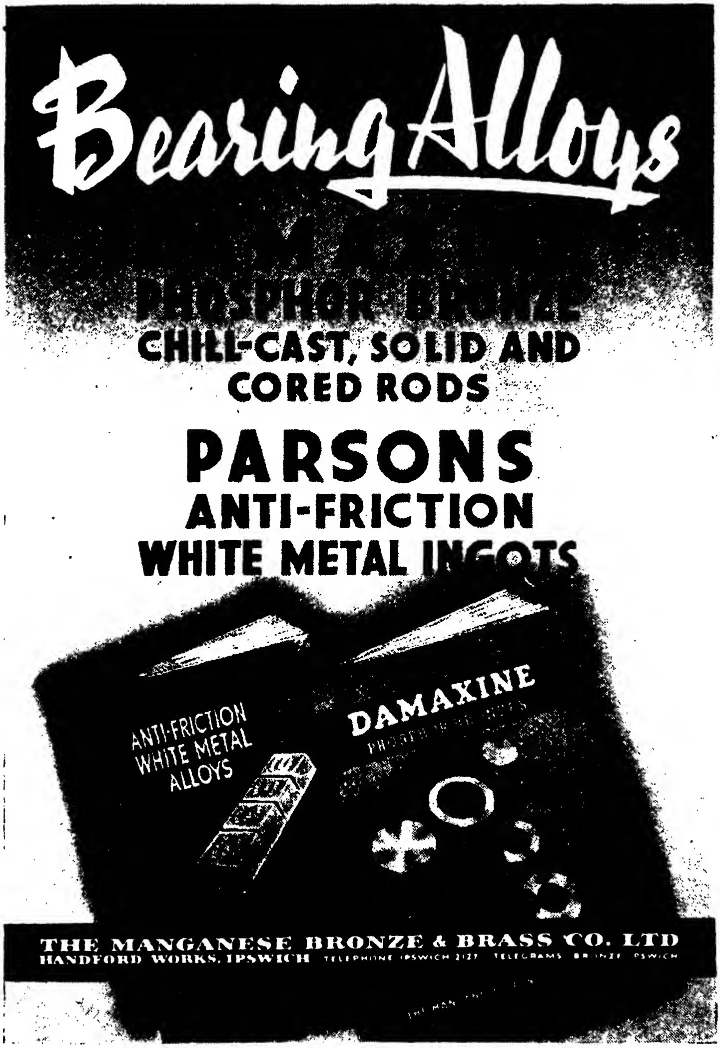
(*b*) Bearing alloys of spongy character; the network of the sponge being moderately hard and tough, the meshwork being soft and plastic.

These types can be grouped according to composition as follows :—

(i) Brasses and bronzes containing compounds of copper with tin, phosphorus, etc.
(ii) Bronzes containing appreciable quantities of lead.
(iii) Alloys having a tin base and containing compounds of tin with antimony, copper, etc.
(iv) Alloys having a lead base and containing compounds of tin with antimony, etc.
(v) General alloys possessed of the characteristics required of a bearing metal.

Of the above, types (i) and (ii) have been considered under the heading of ' Brass ' and ' Bronze ' (*q.v.*), types (iii) and (iv) are known as ' white ' metals, while the majority of the alloys of type

* See also Bearing Pressures, p. 866.


Bearing Alloys
PHOSPHOR BRONZE
CHILL-CAST, SOLID AND
CORED RODS
PARSONS
ANTI-FRICTION
WHITE METAL INGOTS
ANTI-FRICTION
WHITE METAL
ALLOYS
DAMAXINE
THE MANGANESE BRONZE & BRASS CO. LTD
HANDFORD WORKS, IPSWICH

(v) can be termed 'white metals.' Of types (iii), (iv) and (v) it has been observed that metals whose atomic weight divided by their specific gravity gives a high number are good anti-friction metals, while the reverse is the case for metals which have a low quotient. Hence the use of lead and antimony for bearing alloys. In using white metal for bearings the best practice is to use the bearings wholly of white metal, discarding the cast-iron shells and using up the old bearings with the addition of a further small quantity of the more volatile constituents. White metal must be melted in a clean vessel and at as low a temperature as possible. Dross and scum in the metal tends to rapid heating in service. For the metals to be properly alloyed the high as well as the low melting points of these must be reached, but care must be taken not to overheat the alloy. For a thorough alloying a temperature of about 500° C. (930° F.) and less has been found to be sufficient. The molten metal should be stirred with a stick of pine, and a layer of charcoal always kept on the surface to prevent oxidation. The rate of cooling affects the size of the crystals and also the wearing quality. As a general rule cooling too slowly and pouring too cold produces a coarse crystal formation which is liable to heat quickly in use. In the case of the tin-antimony alloys it has been found that the best size for the tin-antimony compound cubes is 0·25 mm. edge. Pouring too hot on the other hand results in a soft metal. The shell should be heated to a temperature of 100° C. to 150° C. (212° F. to 300° F.) before pouring the metal into it, as this not only prevents blow-holes, but reduces the tendency of the lining shrinking away from the shell. The bearing should not be disturbed while the lining is solidifying, since vibration tends to enlarge the crystals and produce brittleness. The matrix of the lining should be just stiff enough to support the crystals, and the latter should be as numerous as is consistent with these, lying clear of each other and thereby causing brittleness. A Brinell hardness of 23·5 for an alloy with a lead base and 30 for a tin base has been found to give excellent results.

ALLOYS HAVING A TIN BASE.

Tin-Antimony Alloys.

These alloys, which are of type (*a*), are the basis of Britannia metal.

Binary tin-antimony alloys are not so satisfactory as ternary alloys containing copper and nickel. The function of either of these elements is to inhibit flotation of the hard crystals which constitute the actual bearing surface. These hard crystals have a low density and otherwise tend to segregate in the upper part of the casting. Typical compositions are :—

Tin.	Copper.	Antimony.	Nickel.
88	4	8–10	—
92	4	3·5–4·0	0·5

Babbitt 'metal' is of similar type.

Babbitt 'Metal.'

This alloy, whose composition is as follows: Tin, 83·3; Antimony, 11·1; Copper, 5·6; is used on important bearings under heavy loads, such as crank-shaft, crank-pin, and cross-head pin journals.

In cast-iron boxes, for light work: 50 parts tin, 5 parts antimony, 1 part copper; for heavy work: 46 parts tin, 8 parts antimony, 4 parts copper. In brass boxes: 64 parts copper, 8 parts tin, 1 part zinc. The bearing to be lined with the metal is first tinned with the Babbitt metal mixed with twice its weight of tin.

Babbitt metal should be stirred thoroughly from the bottom of the pot at frequent intervals when being used, to prevent segregation of the constituents. If the pot is used only occasionally, the Babbitt should be covered with sawdust to prevent the formation of dross.

Admiralty White 'Metal.'

Tin, 86·0; Antimony, 8·5; Copper, 5·5.

Cadmium improves the strength and other properties of tin-base bearing and up to 1·0 per cent. may be present. It is essential for the alloy to be practically free from lead, however, since otherwise its melting point may be too low.

ALLOYS HAVING A LEAD BASE.

Lead-Antimony Alloys.

These alloys should not contain more than 10 per cent. of antimony if they are not to be brittle. Such as do contain upwards of 18 per cent. of antimony must be given suitable support. As with the tin-base alloys, best results are obtained from ternary and quaternary lead alloys.

'Magnolia.'

Lead, 80; Antimony, 14; Tin, 6.
Lead, 79; Antimony, 20; Tin, Copper, etc., 1.

Bearing Alloys with Alkali Hardeners.

Name.	Composition.
Frary	Pb 97·5, Ba 1·5, Ca 0·75, Hg 0·25.
Satco	Pb 97·25, K 0·1, Ca 0·15, Sn 2·5.
Lurgi	Pb 96·5, Ba 2·8, Ca 0·4, Na 0·3.
Bahnmetal	Pb 98·63, Ca 0·69, Na 0·62, Al 0·02, Li 0·04.
Can	Pb 94·9, Ba 1·0, Ca 1·75, Cu 1·35, St 1·0.

These alloys have high compressive strength, hardness and melting point. They will not carry the oads, or operate at as high speeds as will the best tin-base alloys. They are also subject to excessive loss of hardening elements on remelting.

Cadmium-nickel alloys containing from 0·5–3·0 per cent. of nickel and the rest cadmium have been used for bearing liners with excellent results.

(*Swartz.*)

Anti-Friction Alloys.

Delta Anti-Friction Alloys.

(*The Delta Metal Co., Ltd.*)

Delta White Anti-Friction Metals, Nos. IX and IXa, are used for lining bearings, bushes, etc., and are particularly suitable for heavy bearings such as those of marine engines, for railway trucks, dynamos, etc., as they do not wear, heat, or cut the journals; they are also cheap.

The application of the metals presents no difficulties whatever; the parts required to be lined should be thoroughly cleaned, and when the metal is melted (which may be done in a ladle, over a slow fire, taking care not to overheat it) it is applied in the usual way. The melting point is about 450° C.

Fry's Anti-Friction Alloys.

(*Fry's Metal Foundries, Ltd.*)

These anti-friction alloys cover the whole range of white metals employed in engineering practice; commencing with lead-free alloys of high tin content with sufficient antimony and copper to provide necessary hardness and wearing qualities combined with great toughness (as in the O1 brand) which are used for special service, as in the big end bearings of engines for aircraft or high-powered cars, and ending with the finest quality of lead-base alloys of the Magnolia class.

The series includes bearing metals suitable for every class of service, heavy and light loads. high speeds and slow-running machinery, and plastic metal for emergency and special services. The high-tin alloys are self-tinning. All but the first alloy in the series have a structure consisting of a matrix of tough and durable alloy with cuboid antimony-tin crystals regularly embedded throughout. The first is a tin-antimony solid solution with crystals of copper-tin and copper-antimony constituents interspersed in dendritic or needle-shaped formation. The constituent of dendritic crystal form is actually present in all well-balanced alloys of this character, adding greatly to their toughness and durability, and also serving to ensure the even distribution of the cuboid antimony-tin constituent. Finally, hard and highly resistant zinc-tin rock crusher bearing metal of great endurance can be furnished.

The following table shows a selection of bearing alloys for particular purposes and for design work:—

Constituents, per cent. A = Tin; B = Antimony; C = Copper; D = Lead.					Tensile Test.		Brinell Hardness Number. 10-mm. Ball. 500 Kilos.	Compression Test. Tons per sq. in.	
No.	A	B	C	D	Tons per sq. in.	Elongation per cent. in 2".		Yield Pt. 0·001 in.	Compressed to half length.
1	93	3·5	3·5	—	5·12	11·6	24·9	3·569	14·732
2	86	10·5	3·5	—	6·65	7·1	33·3	4·372	17·232
3	83	10·5	2·5	4·0	5·60	—	34·5	4·284	17·640
4	80	11·0	3·0	6·0	5·70	—	32·1	4·640	17·500
5	60	10·0	1·5	28·5	5·04	—	27·1	3·696	12·856
6	40	10·0	1·5	48·5	4·58	—	21·8	3·660	11·284
7	20	15·0	1·5	63·5	5·48	—	31·3	4·016	12·212
8	78	11·0	11·0	—	6·36	—	37·0	4·550	17·856
9	5	15·0	—	80·0	4·69	2·8	24·9	3·590	13·356

Miscellaneous Metals for Bearings and Engine Parts.

The following special mixtures are stated to be satisfactory for the duty mentioned :—

For Lining.	Tin.	Lead.	Zinc.	Antimony.	Copper.	Bismuth.
Dynamos—high speed . .	88	—	—	8	3·5	0·5
Marine engines	77	17	—	3	3	—
Eccentrics	5	78	—	15	2	0·25
Submerged bearings . .	40	48	—	10	2	—
Main bearings	34	44	—	16	6	—
Slides, thrusts	63	—	30	2·5	2·5	—
Railway trucks	42	—	56	—	2	—
Axle-boxes (by analysis) .	74·22	13·50	1·80	6·55	3·60	—
Loco. and carriage bearings (Italian Railways) . .	38	37	—	25	—	—
Carriage bearings (German Railroad)	20	60	—	20	—	—
Bearings for rock-breaking machinery	10	—	80	10	—	—
Loco. bearings (A. Allan & Sons)	5·5	35	—	—	59·5	—
Cross-head slides and rod bearings (Eastern Railway, France)	0	65	—	25	10	—
Axle bearings and connecting rods for heavy engines (Eastern Railway, France) .	83·3	—	—	11·12	5·5	—
Eccentric straps and slide valves (Eastern Railway, France)	10	70	—	20	—	—
Passenger and freight car bearings (A. Allen & Sons)	4·5	45	—	—	50·5	—
Loco. bearings (Dewrance) .	33	—	—	45	22	—
Loco. and wagon axle boxes (Fenton's alloy) . .	14·5	—	80	—	5·5	—
Bearings, sleeves, slides and guide gibs (U.S. Navy)	12·5 to 14·5	1	2·5to4·5	—	82 to 84	—
Pivot bearings (Hoyle's alloy)	46	42	—	12	—	—
Lining, bearings and crank pin bearings (moderate speed) (Parsons' white brass) .	68	0·5	30·5	—	1	—
Metallic rod packing (Eastern Railway, France) . .	12	80	—	8	—	—
Piston packing rings .	—	50	—	—	50	—
	13	1	1	—	85	—
Pump and pump chambers and gun-metal valves .	10	1·5	1·5	—	87	—
Cog wheels	9	25	—	—	56	—
	10	10	—	—	80	—
Anti-acid metal (by analysis) .	78·84	14·75	—	Trace	3·70	—
Plastic metal . . .	80	10	—	1	8	1
Genuine Babbitt (hard) . .	80	—	—	10	10	—
,, ,, (No. 2) .	83	—	—	9	8	—
Universal bearing metal. .	6	78	—	16	—	0·25
Anti-friction castings . .	24	—	80	—	4	—
Bearings for high-speed machines	—	—	14 to 18	—	82 to 86	—
,, ,, ,, . .	17	—	—	77	6	—
,, ,, ,, . .	18·5	—	—	76	5·5	—

ZINC-TIN ALLOYS.

Zinc and tin form a series of eutectiferous alloys, some of which are of value as bearing metals. The alloys, in common with all bearing metals, are very susceptible to differences in casting temperature, as may be gathered from the following table, which gives in detail the effect of varying the temperature of casting an alloy of 70 per cent. of zinc and 30 per cent. of tin :—

Casting Temperature.	U.T.S. *or* Tensile Strength. Tons per sq. in.	Elongation. Per cent.	Stress to compress to half length. Tons per sq. in.
525° C.	5·09	3·0	7·7
500° C.	6·17	3·0	13·6
475° C.	6·56	8·0	15·4
450° C.	7·37	17·5	14·8
425° C.	7·37	35·5	15·0
400° C.	7·24	20·0	15·8

The above figures are for chill castings.

The alloys of zinc and tin containing from 25 to 75 per cent. of zinc possess, when in the chill cast condition, tenacities varying from 6·0 to 6·5 tons per square inch, percentage elongations varying from 30 to 45, and are compressible to half-length under stresses (calculated in the original areas of the test-pieces) varying from 13 to 17 tons per square inch. The eutectic melts at about 200° C. and contains 16 atomic per cent. of zinc.

In order that these alloys may approximate more closely to type (*a*) (*q.v.*) bearing metal, the zinc-tin series containing no relatively hard constituent, copper and antimony are frequently added. The addition of one or both of these constituents results in the formation of a bearing metal in which the hard compounds of antimony and zinc, etc., are indiscriminately distributed throughout the eutectiferous matrix. The following are analyses of certain alloys of this type :—

Zinc.	Tin.	Copper.	Antimony.	Use of Alloy.
80	14·5	5·5	—	Fenton's alloy for locomotive axle-boxes
69	26·0	5·0	—	Propeller bush.
88	2·0	8·0	2·0	Bearing metal for general use.

In this connection it may be noted that ternary alloys generally are far preferable to complicated mixtures and usually fufill all the requirements of good bearing metals.

FUSIBLE METALS.

With the following combinations of metals, which form 'eutectiferous' alloys, it is possible to prepare an alloy having a melting point lower than that of any member of the series.

Series.	Alloy with Lowest Melting Point (Eutectic).	Melting Point. °C.
Antimony-lead	12·5 per cent. Antimony	247
Bismuth-lead	57·4 „ Bismuth	124·5
Bismuth-tin	56·7 „ „	137
Bismuth-cadmium	60 „ „	146
Cadmium-tin	52 „ Cadmium	177
Lead-tin	36·5 „ Lead	180
Cadmium-lead	17·4 „ Cadmium	247·3
Zinc-lead	2·3 „ Zinc	318

By the addition of one or both of the residual metals to any of the above combinations, systems of ternary and quaternary alloys, having individual members with melting points lower than those of all other members of these systems and less than those of the eutectics given above, may be obtained. These ternary and quaternary alloys are the true 'fusible metals.'

The alloys of lead, tin and bismuth are of value for the following purposes:—

(i) The manufacture of casts of delicate objects that would be damaged if subjected to high temperatures. The following alloys are recommended for this purpose:—

Bismuth	59·2	50·0	50·0	50·0
Lead	27·3	31·2	30·0	25.0
Tin	13·5	18·8	20·0	25.0

These alloys expand considerably on cooling and are valuable also for the taking of impressions from dies, medals, etc.

(ii) Fusion pyrometers. Parkes and Martin recommend the following fusible metals as pyrometers for the tempering of steel tools. The article to be treated is placed on a plate of the alloy, which is heated till it melts. The tool, being then in the required condition, is quenched in water.

TABLE OF MELTING POINTS OF BISMUTH-LEAD-TIN ALLOYS.

Bismuth.	Lead.	Tin.	Melting Point.	Bismuth.	Lead.	Tin.	Melting Point.
47·0	35·5	17·7	98° C.	13·7	44·8	41·5	160° C.
40·0	40·0	20·0	113° C.	12·8	49·0	38·2	172° C.
33·3	33·3	33·3	123° C.	12·5	50·0	37·5	178° C.
30·8	38·4	30·8	130° C.	11·4	45·6	43·0	165° C.
20·0	40·0	40·0	145° C.	10·0	40·0	50·0	162° C.
16·0	36·0	48·0	155° C.				

(iii) Soft solders. For the union of metal to glass, *e.g.* in the manufacture of paraffin lamps, the following alloy has been suggested:—

Lead, 40·0; Bismuth, 33·4; Tin, 26·6

(iv) Fusible plugs.

The above alloys are nearly all of a brittle character. For the manufacture of ductile fusible alloys the use of cadmium in place of bismuth is recommended.

The following table includes the best known of the other fusible metals :—

Alloy.	Cadmium.	Lead.	Tin.	Bismuth.	Melting Pt.
Fusible alloy . .	12·5	25·0	12·5	50·0	55·5° C.
Lipowitz's alloy .	10·0	26·6	13·3	50·0	60–68° C.
Woods' alloy . .	15·4	30·8	15·4	38·4	71° C.
Fusible alloy . .	34·5	27·5	10	27·5	75° C.
,, ,, . .	6·2	34·5	9·3	50·0	77° C.
,, ,, . .	25	25	50	—	86° C.
,, ,, . .	16·6	—	33·3	50·1	95° C.
,, ,, . .	11·1	—	33·3	55·6	95° C.
,, ,, . .	25	—	25	50	95° C.

Spence's Metal.

This compound is obtained by melting the three sulphides of iron, zinc and lead with sulphur. The product is a dark grey mass of great tenacity, small power of conducting heat, a specific gravity of 3·4, a melting point of 160° C. In congealing it expands like bismuth and type metal, and resists in a remarkable degree the action of atmospheric influences, alkalies and acids, even of aquaregia, its surface scarcely being affected after having been exposed to the action of the latter for four weeks. Its property of expanding in congealing, and therefore filling completely all depressions of the mould, makes it particularly available for castings. If the compound is poured on a plate on which the impression of the hand has been made, the cast will show all the lines and the pores of the palm.

The material has been used for jointing gas and water pipes.

Type-Metal.

Metal.	Tin.	Antimony.	Lead.
Monotype	5% to 10%	15% to 19%	Remainder
Founders' Type for Hand Composition	10% ,, 18%	20% ,, 30%	,,
Slug Casting Machines—			
Linotype, Intertype, and Ludlow .	2·5% ,, 5%	10% ,, 13%	,,
Stereotype—			
Flat Plates	5% ,, 10%	15% ,, 18%	,,
Rotary Plates	6% ,, 10%	15%	,,

The addition of copper to type metals as an extra hardener is generally discontinued. Small quantities creep in during repeated remelting. Quantities above 0·12 per cent. are apt to give trouble in machine composition work.

Non-Magnetic Alloys.

These alloys are used as substitutes for steel wherever it is employed in the parts of watches. The composition of the alloys varies from 45 to 75 parts of palladium, 15 to 30 parts of copper, 20 to 25 parts of silver, with sometimes the addition of small proportions of iron, nickel, gold and platinum. These alloys are claimed to be unoxidisable in moist air, to preserve their elasticity indefinitely, not to vary sensibly with changes of temperature, and to remain uninfluenced by proximity to dynamos.

Another non-magnetic alloy is made of from 30 to 40 parts of gold, 30 to 40 parts of palladium, 0·1 to 5 parts of rhodium, 10 to 20 parts of copper, 0·1 to 5 parts of manganese, and the same proportions of silver and platinum. The copper and manganese are first mixed, after which the other metals are added ; or all the metals may be put into a crucible at the same time, the manganese being used for the bottom layer.

A non-magnetic variety of 'Bull's Metal' is manufactured and sold as No. III, and is being adopted for funnels and other erections of warships where the material must possess the mechanical properties of steel without its magnetic influence.

NON-FERROUS ALLOYS.

The following tables * give representative compositions of alloys met with in commercial engineering. In a number of cases the composition of an alloy may vary over a relatively large range in practice. The analyses given are of alloys actually used for the purpose specified.

Copper Base Alloys—Brasses.

	Cu.	Zn.	Sn.	Pb.	Fe.	
Brass.						
Aluminium	63	33·3	—	—	—	Al 3·3, Si 0·3
Bell	64·25	35	0·75			
Bismuth	47	21	1	—	—	Ni 30·9, Bi 0·1
Brazing	75	25				
Bullet	90	9	—	1		
Burr	62	38				
Cartridge	70	30				
Clock	62·5	35·75	—	1·75		
Diaphragm	95	2	3			
Dipping	66·7	33·3				
Deep drawing	70	30				
Electrical castings	84	13	3			
Engravers	66	33				
Free cutting	88·5	10	—	1·5		
Eyelet	65	35				
Hot pressings	60	38	—	2		
Red	87	13				
,,	86	11·1	2·9			
Reed	69	30	1			
Rolled	74·3	22·3	3·4			
Rule	62·5	35	—	2·5		
Screw	78	16	4·5	1·5		
Spring	72	28				
Tube	65	35				
Vanadium	70	29·5	—	—	—	V 0·5
White	69·8	25·8	4·4			
Wire	67	33				
Yellow	66	34				
Various.						
Aitch metal	60	38·2	—	—	1·8	
Barronia (tubes)	83	14·5	2·0	0·5		
Battery copper	94	6				
Birmingham platina	46·6	53·15	—	—	0·25	
Bronze powder	84	16				
Burr metal	90	10				
Solder for steel	82·6	17·4				
Gold-leaf metal	84	16				
Gold leaf	66–80	34–20				
Hercules metal	54	36	—	—	7·5	Al 2·5
Jewellers' metal	90	10				
,, ,,	91	7·5	1·5			
Leaded	88·5	10	—	1·5		
Manganese	80	5	—	—	—	Mn 15
Mn–Ni	65	5	—	—	—	Mn 20, Ni 10
Matrix	62	36·5	—	1·5		
Manganin	53·4	39	2·7	—	—	Ni 2·5, Mn 1·7 Al 0·2
Non-tarnishable	63·6	31	3·25	2		
Pen metal	85	13	2			
Pin wire	61	39				
Seamless tubing	60	40				
Solder, hard	57	43				
,, white	40	60				

* W. Campbell (*Proc. Amer. Soc. Test. Mat.*) and other authorities.

Copper-Tin Alloys—Bronzes.

	Cu.	Zn.	Sn.	Pb.	Fe.	
Bronzes.						
Acid	82	2	8	8		
Aluminium . . .	88	—	—	—	1·2	Al 10, Mn 0·8
Bearing, railway . .	82	—	18			
„ auto . .	80	—	10	8		
„ machinery .	87·5	—	12·5			
„ hard . .	79·3	6·4	14·3			
Manganese . . .	82	5	8	3	—	Mn 2
Silicon	91	—	9	—	—	Si 0·05
Telegraph . . .	80	7·5	5	7·5		
Tobin. . . .	58·2	2·3	39·5			
Screw. . . .	93·5	5·0	1			
Plastic . . .	64	—	5	30	—	Ni 1
Various.						
Argental . . .	85	—	10	—	—	Co 5
Speculum (Coopers) .	57·8	3·6	27·3	—	—	As 1·2, Pt 10
„ (English) .	66·6	—	33·4			
„ „ .	50	29	21			
Hairsprings. . .	94·3	—	5·5	—	—	P 0·2
Hercules . . .	85·5	2	10	—	—	Al 2·5
High temperature .	90·7	6·3	2·7	1·3		
Instrument . .	92·2	—	7·8			
„ . .	82	5	13			
Gun-metal, small . .	93	—	7			
„ large . .	90	—	10			
Piston rings . .	84	8·3	2·9	4·3		
Steam fittings . .	88	2	8	2		
Medals . . .	93	—	7			
Rivets . . .	64	--	36			

Tin Base Alloys.

	Sn.	Sb.	Pb.	Cu.	
Britannia metal . .	91·5	7·1	1·4		
„ „ . . .	25	50	—	—	Bi 25
„ „ spoons	84·7	5	3·7		
Hoyt's No. 11 . .	91·4	3·5	0·18	4·3	Ni 0·55
Pewter . . .	81·2	5·7	1·5	1·6	
„ best . .	86	14			
„ common . .	80	—	20		
Plastic metal . .	80·5	8·6	—	9·5	Fe 1·4
Richard's babbitt .	82·43	9·8	—	8·1	
Silver solder. . .	75	—	—	3	Ag 10, Zn 2
Ship's nail alloy . .	50	17	33		
Silver foil . . .	90	—	—	—	Zn 10
„ „ . . .	97·5	—	—	2·5	
Tin foil . . .	87·5	0·5	8	4	
White metal . .	28·4	56·8	—	7·4	Zn 7·4

Lead Base Alloys.

	Pb.	Sb.	Sn.	Cu.	
Accumulator metal .	90	0·75	9·25		
Anode metal . .	94	—	6		
Antifriction. . .	77	14	8	1	
Capsule metal . .	92	—	8		
Electrotype metal. .	93	4	3		
Linotype metal . .	83	12	5		
Stereotype metal . .	82·5	13	4·5		
„ „ . .	69	15·5	—	—	Bi 15·5
Printing type . .	80	20	—		
Shot lead . . .	99·8	—	—	—	As 0·2
Tea lead . . .	80	—	20		
Terne metal. . .	80·25	1·75	18		
Alloy that expands on cooling . .	75	16·7	—	—	Bi 8·3

Copper-Nickel Alloys.

	Cu.	Ni.	
Constantan	60	40	
„	54	45	Mn 1·3, Fe 0·1
Cupro nickel sheet	80	20	
Mumetal	5·3	74	Fe 20, Mn 0·7
Turbine blading	82·1	14·7	Al 2·5, Zn 0·7, Si 0·04
Flat spring strip	57	16	Zn 27
Valves for superheated steam	33	54	Sn 13
German silver	33·3	33·3	Zn 33·4
Watch alloy	50	47·2	Cd 2·8
White copper	40·4	31·6	Zn 25·4, Sn 2·6

Copper-Nickel-Zinc Alloys.

	Cu.	Zn.	Ni.	
Argentan sheet	40–65	17–32	15–20	
Brazing solder	45	45	10	
Electrum	51·6	22·6	25·8	
German silver	46	34	20	
„ „	50	29	21	
Knife bolsters	56	28	16	
Nickelin	75–55	10–20	18–32	
Optical wire	54	28	18	
Platinoid	62	22	15	
Resistance wire	56·5	25·5	18	
Rheotan	52	18	25	Fe 5
Spoons, forks	60	22	18	
Watch-case bezels	60	24	16	
Watch-case metal	55–65	30–16	10–28	Pb 0·1
Toucas	35·75	7·14	28·56	Fe 7·14, Sn 7·14 Pb 7·14, Sb 7·14

Copper-Manganese Alloys.

	Cu.	Mn.	Ni.	
Magnetic alloy (Heusler)	68	18	—	Al 10, Pb 4
Rheotan	84	12	—	Zn 4
Manganin	86–84	12	2–4	
„	84	4	12	
Resistance metal	85	12	3	
Everdur	96	—	—	Si 2·75–3·25, Mn 0·75–1·25, Fe 0·5 max.

Aluminium Alloys.

	Al.	Cu.	Mn.	
Mg 7	93	—	—	Mg 7
Curtiss	95·2	2·5	—	Mg 1·5
Aerolite	86	12	2	
Magnalite	94	2·5	—	Zn 0·5, Mg 1·5, Ni 1·5
Magnalium alloy	95	—	—	Mg 5
Y alloy	92·5	4	—	Mg 1·5, Ni 2·0
Lynite	90	8	—	Mg 0·5, Fe 1·5

ALUMINIUM ALLOYS—*continued.*

	Al.	Cu.	Mn.	
Aerolite	96·9	1·2	—	Mg 0·4, Fe 1, Si 0·5
Alpax / Silumin	86·6	0·15	0·25	Zn 0·2, Fe 0·8, Si 12–13
Duralumin	Bal.	3·5–5·5	0·5–0·8	Mg 0·5
„	Bal.	4·2	0·49	Mg 0·76, Fe 0·67, Si 0·3
Die casting	92–82	8–18		
Magnalium (cast)	85	—	—	Mg 15
„ (sheet)	95	—	—	Mg 5
Nickeloy	94	4·2	—	Ni 1·4
Dental alloy (Pages)	93	2·25	—	Au 4·75
Platinum substitute	23·6	—	—	Ni 72, Bi 3·7, Au 0·7
Sibley Alloy	67	—	—	Zn 33

EUTECTIC ALLOYS.

Alloy.	Temp. of Fusion °C.
Bi 49·5, Sn 13·1, Pb 27·3, Cd 10·1	70–74
Bi 51·6, Pb 40·2, Cd 8·1	91·5
Bi 52·5, Sn 15·5, Pb 32	96
Bi 54, Sn 26, Cd 20	103
Bi 56·5, Pb 43·5	125
Bi 58, Sn 42	137
Bi 61·5, Cd 38·5	144
Sn 50, Pb 32, Cd 18	145
Sn 73·5, Cd 24·5, Zn 2	163
Sn 68·5, Cd 31·5	178
Sn 63, Pb 37	182
Sn 91·5, Zn 8·5	197
Pb 87, Sb 13	247
Pb 77, Cd 23	249

HEAT-RESISTING ALLOYS.

Alumel	Ni 94, Fe 0·5, Mn 2·5, Al 2·0, Si 1·0.
Chromel A	Ni 80, Cr 20.
„ B	Ni 85, Cr 15.
„ C	Ni 64, Cr 11, Fe 25.
Corronil	Ni 70, Mn 4, Cu 26.
Crenit	Ni 60, Cr 40.
„ Heating units (up to 850° C.)	Ni 60–65, Cr 12–15, Fe balance.
Eureka or Manganin	Ni 4, Mn 12, Cu 84.
„ „	Ni 12, Mn 4, Cu 84.
„ „	Ni 5, Mn 25, Cu 70.
Kanthal	Cr 25, Al 5, Co 3, Fe 67.
Nickelin	Ni 32, Cu 68.
Nichrome I	Ni 60, Cr 11, Fe 25, Mn 4.
„ II	Ni 75, Cr 11, Fe 12, Mn 2.
„ III	Ni 85, Cr 15.
„ IV	Ni 80, Cr 20.
Non-magnetic, high resistance	Ni 30, Cr 70.
Permalloy	Ni 78, Fe 22.
Perinvar	Ni 45, Fe 30, Co 25.
Platinoid	Ni 14, Cu 60, Zn 24, W 1–2.
Rheotan	Fe 12, Cu 84, Zn 4.
„	Fe 45, Cu 50·4, Zn 16·9, Ni 25·3.
Smith's No. 10 Alloy	Cr 37·5, Al 7·5, Fe 55.

Heat-resisting Alloys.

Flame-resisting	Ni 9·7, Cr 14, Fe 75·1, Si 0·2, Mn 0·77, C 0·23.
Carburising boxes	Ni 30–50, Cr 16–22, Fe balance.
Non-scaling	Ni 24, Cr 24, Si 3, Fe balance.
Electrodes	Ni 93–97, Si 2–4, Mn 1–3.
"	Ni 82–86, Cr 10–12, Si 2–4, Mn 1–3.

Cutting-tool Alloys.

Carboloy I	Tungsten carbide + 8 per cent. Co.
" II	" " + 13 per cent. Co.
Widia	W 86·4, Co 6·1, C 5·68 (tungsten carbide + 6 per cent. Co.)

Physical Properties of Metals and Alloys.

Copper and Copper Alloys.

Alloy.	Condition.	Yield Point. Tons per sq. in.	Ultimate Strength. Tons per sq. in.	Elongation. %	Reduction of Area. %	Brinell No.	Remarks.
Copper, 99·9%	(a) Electrolytic, annealed at 200° C.	3·8	16·9	50	50	40	Shearing strength, cast copper, 3,000 lbs. per sq. in.
	(b) Not annealed, 96% reduction	—	30	0·8	64·5	—	Modulus of elasticity (Copper).
	(c) Hot-drawn, 64% reduction	—	20·9	4·3	70·5	—	Lbs. per sq. in. (a) Electrolytic 17·4 × 10⁶ (b) Cast 11·0 × 10⁶ (c) Hard-drawn 17·6 × 10⁶
Copper, 99·6% Commercial	Cast	4·5	11·3	20	60	80	
	Rolled hard (40% reduction)	9·0	22·5	5	8	94	
	Annealed at 500°C.	—	15·6	50	60	42	
	Cold-drawn (50% reduction)	16·5	22·5	9	—	—	
Brass (red) Cu 90, Zn 10	Sand-cast	—	13·0	22	—	—	Compressive strength (Brasses).
	Hard-rolled	—	24·6	5	—	60	
	Soft-rolled	—	16·5	40	70	47	Cu / Zn / Lbs. per sq. in.: 90 / 10 / 30,000; 80 / 20 / 39,000; 70 / 30 / 60,000; 60 / 40 / 75,000; 50 / 50 / 110,000
Brass (yellow) Cu 60, Zn 40	Sand-cast	9·8	20·5	15	22	—	
	Hard-rolled	20·0	31·2	30	50	—	Modulus of elasticity. Average, 13 × 10⁶ lbs. per sq. in.
Bronze Cu 97·7 Sn 2·3	Cast	3·8	12·5	20	—	—	Compressive strength (Bronzes).
	Rolled	4·8	21·4	55	75	—	Cu / Sn / Lbs. per sq. in.: 97·7 / 2·3 / 34,000; 90·0 / 10·0 / 56,000; 80·0 / 20·0 / 118,000; 70·0 / 30·0 / 150,000. Modulus of elasticity 10–15·5 × 10⁶ lbs. per sq. in.
Bell metal Cu 90, Sn 10	Cast	4·5	14·3	1·5	—	—	

Properties of Non-Ferrous Alloys at Elevated Temperatures.

Alloy.	Analysis.
A. Muntz Metal	Cu 58·96, Zn 39·77, Sn 0·56, Fe 0·14, Pb 0·67.
B. Extruded Sceptre Brass	Cu 61·7, Zn 35·6, Fe 1·44, Al 1·05, Pb 0·17.
C. Drawn Manganese-Bronze	Cu 56·9, Zn 40·28, Sn 0·75, Ni 0·21, Fe 0·82, Al 0·19, Mn 0·19, Pb 0·66.
D. Sand-Cast Manganese-Bronze	Cu 58·6, Zn 36·9, Sn 0·08, Ni trace, Fe 1·46, Al 0·01, Mn 1·88, Pb 0·06.
E. Sand-Cast Phosphor-Bronze	Cu 85·38, Zn 1·01, Sn 12·55, Ni 0·11, Fe 0·02, Pb 0·61, P 0·24.
F. Chill-Cast Phosphor-Bronze	Cu 90·25, Sn 9·47, Pb 0·17, P 0·17.
G. Nickel Copper	Cu 97·79, Ni 2·02, Fe 0·08.
H. Naval Brass	Cu 60·95, Zn 35·04, Sn 1·3, Fe 0·11, Pb 0·20.
J. Sand-Cast Gun-Metal.	Cu 85·5, Zn 3·29, Sn 10·2, Pb 0·12.

Brinell Hardness at Different Temperatures.

Alloy.	Temperature. °C.			
	18.	250.	350.	450.
A	126	107	63	18
B	133	111	71	26
C	114	99	61	15
D	132	101	62	19
E	98	82	77	52
F	77	75	64	42
G	171	—	142	141
H	96	88	61	17
J	86	69	63	58

Tensile Strengths at Different Temperatures.
(Tons per sq. in.)

Alloy.	Temperature. °C.			
	20.	250.	350.	450.
A	31·5	23·0	12·6	3·9
B	—	—	—	—
C	34·4	22·8	11·7	2·7
D	32·5	23·0	12·6	4·1
E	13·5	13·5	9·7	9·4
F	—	—	—	—
G	38·8	35·8	36·0	31·0
H	—	—	—	—
J	15·8	16·1	10·3	5·2

(*Aitcheson.*)

Physical Properties of Metals and Alloys.*

Metal or Alloy.	Condition.	Yield Point. Tons per sq. in.	Ultimate Strength, Tons per sq. in.	Elong. %	Reduction of Area. %	Brinell No.	Remarks.
Gold, 100%	Cast	—	11·3	25·0	—	—	
	Hard-drawn	—	16·5	—	—	—	
Gold-copper Au 90, Cu 10	Hard drawn	—	29·0	—	—	73	
Lead, Commercial	Cast	—	0·8	—	—	—	
	Hard-rolled	—	1·5	—	—	—	
	Soft-rolled	—	1·1	—	—	—	
	Hard-drawn	—	1·4	—	—	—	
Nickel, 98·5%	Cast	10·6	17·0	5·7	6·1	76	
	Wrought	8·0	19·0	11·0	—	83	
	Hard-rolled	—	41·0	11·0	—	—	
	Hard-drawn	—	69·4	—	—	—	
Palladium	Hard-drawn	—	17·4	—	—	—	
Platinum	Hard-drawn	—	23·6	18·0	—	—	
	Drawn, annealed	—	15·6	50·0	—	—	
Silver, 100%	Cast	—	17·8	—	—	—	
	Hard-drawn	—	22·9	—	—	59	
Ag 75, Cu 25	Hard-drawn	—	49·0	—	—	—	
Tantalum	Hard-drawn	—	58·0	—	—	—	
Tin, 99·8%	Cast	—	1·8	35·0	—	14	
	Rolled	0·72	2·4	—	—	—	
	Hard-drawn	—	4·5	—	—	—	
Tungsten, 99·2%	Swaged rod	—	96·0	4·0	28·0	—	
	Hard-drawn	—	263·0	—	65·0	—	
Zinc	Cast, fine	—	5·4	—	48·0	—	
	Rolled	1·3	12·0	—	—	—	
	Hard-drawn	—	7·1	—	—	—	
Aich's metal, Cu 60, Zn 38·2, Fe 1·8	—	25·5	—	—	—	—	
Sterro metal Cu 55, Zn 42·4, Fe 1·8, Sn 0·8	Cast	—	27·0	—	—	—	
	Forged	—	34·0	—	—	—	
	Hard-drawn	—	37·0	—	—	—	
Leaded brass Cu 60, Zn 35, Pb 5·3	Cast	—	14·7–17·4	30–26	35–30	—	
	Annealed sheet	—	18·7	50	—	—	
	Hard sheet	—	27·2	30	—	—	
Manganese-bronze	Sand-cast	13·4–15·6	31·2–33·5	30–22	32–25	109–119	
Cu 58, Zn 39, Mn 0·05	Chill-cast	14·3–16·5	33·5–35·7	32–25	34–28	119–130	
Nickel silver Cu 60·4, Zn 31·8, Ni 7·7	Cast	6·9	16·0	40·0	42	46	
German silver Cu 61·6, Zn 17·2, Ni 21·1	Cast	8·4	18·3	28·5	25	80	
Naval brass	Sand-cast	7·0	19·0	30·0	32	—	
	Rolled and annealed	16·5	27·6	25·0	37	—	
Tobin bronze Cu 58·2, Zn 39·5	Cast	11·2	26·8	—	—	—	
Sn 2·3	Rolled	24·1	35·2	35·0	40	—	
Victor bronze Cu 58·6, Zn 38·5, Al 1·5, Fe 1·0, V 0·03	Cold-drawn	35·7	41·0	11·5	29	—	

* Bureau of Standards, Washington.

PHYSICAL PROPERTIES OF METALS AND ALLOYS—*continued.*

Metal or Alloy.	Condition.	Yield Point. Tons per sq. in.	Ultimate Strength. Tons per sq. in.	Elong. %	Reduction of Area. %	Brinell No.	Remarks.
Cobalt, 99·7%	Cast	12·4	14·7	—	—	—	
	Annealed	14·1	16·5	—	—	—	
	Wrought wire	—	39·7	3·0	—	—	
Stellite . .	—	—	—	—	—	512	Co 59·5, Mo 22·5, Cr 10·8 Fe 3·1, Mn 2·0, C. 0·9, Si 0·8

Mechanical Treatment of Metals and Alloys.

Method of Treatment.	Details of Method to be avoided.	Consequences of Details of Method of Work.
Hot working . .	1. Excess 'reduction' or rate of 'reduction.'	Cracking of metal.
	2. Deficient 'reduction' or rate of 'reduction.'	Coarse, weak metal.
	3. Excess temperature at commencement.	Weak, brittle metal and formation of fissures therein.
	4. Deficient temperature at conclusion.	Internal strain and consequent hardening. Possible 'season cracking.'
	5. Retention of scale on metal during operations.	Rough and pitted surface and possible cracking.
Cold working	1. Excess 'reduction' or rate of 'reduction,' without annealing.	Surface cracking and hollow drawing. Strain-hardening.
	2. Deficient 'reduction or rate of 'reduction.'	Insufficient hardening — where hardening is expected.
	3. Unequal 'reduction' . . .	'Season cracking' or cracking during annealing.

Heat Treatment of Metals.

Method of Treatment.	Details of Method to be avoided.	Consequences of Details of Method of Work.
Annealing . .	1. At too high a temperature .	'Burning' and 'over-heating' in steel and brass. Brittleness and sponginess generally, owing to abnormal grain size.
	2. At too low a temperature. .	Insufficient structure change.
	3. For too long a period -- at correct temperature.	Diffusion and aggregation of micro-constituents of metal or alloy.
	4. Careless charging in or firing of furnace.	Sudden access of heat may lead to cracking of cold metal.
	5. In an injurious atmosphere or in contact with injurious substances.	Copper is injured by reducing gases, steel by oxidising gases. Water and oil have injurious effects in many cases.
Quenching . .	1. From too high a temperature .	Cracking, warping and brittleness.
	2. From too low a temperature .	The object of the operation is defeated.
	3. Unequal	Warping.
Tempering . .	1. At too high a temperature .	Removes the effects of quenching.
	2. At too low a temperature. .	Does not remove the effects of quenching sufficiently.
	3. Unequal	Lack of uniformity and consequent failure.

SOLDERS AND SOLDERING.

The following notes are published with the kind permission of the Editor of 'The Metal Industry.' (See 'The Metal Industry,' vol. 9, No. 4.)

Points to be Observed for Successful Soldering.

Attention should be paid to the following points if success is to be attained in soldering or brazing:—

1. In the preparation of the solder, uniformity of composition should be aimed at. Readily volatile or oxidisable constituents should be retained.

2. In the choice of the solder, consideration should be taken of the fusibility of the solder. The melting point of the solder should be less than that of the metal to be soldered. It should be noted, however, that the more nearly the melting point of the solder approaches that of the metal to be soldered the better is the union obtained, although the more difficult is the operation of soldering. It should be considered whether the solder will mingle perfectly with the surfaces to be united. One having mechanical and physical properties as nearly as possible approximating to those of the metals to be united should be used. One whose mechanical and physical properties will not be subsequently modified by any treatment—thermal or otherwise—should be employed.

3. In the operation of soldering, thorough mechanical and chemical cleanliness of the surfaces to be united must be obtained, e.g. by scraping, filing, grinding, etc., and by dipping in suitable chemical reagents. A source of heat both smokeless and sufficiently hot to raise the temperature of the sections to be united above the melting point of the solder is necessary. A suitable flux must be used.

Fluxes.

Fluxes are used to protect the surfaces of the metal to be united from the oxidising action of the atmosphere during the process of heating. The presence of any non-metallic materials, such as oxides or carbonates, on the surfaces to be joined, effectually prohibits perfect union. As the process of heating a metal in the air always leads to the formation of a film of oxide on its surface, means must be employed to protect the surface. Use is made of readily fusible salts and of oils that are non-volatile at the temperature of soldering for this purpose. For hard-soldering borax is practically the only flux employed. It is applied as a paste with a brush to the surfaces of small articles, and as a powder to those of larger articles. For soft-soldering a variety of materials is used, the following list giving the most important of these, together with their uses.

(i) Zinc chloride is used for the soldering of sheet metal, copper alloys and bright iron. It is poisonous and corrosive, and is on these accounts unsuited (*a*) for the soldering of tins destined to contain food, and (*b*) for the production of delicate work, e.g. electric fuses.

Zinc Chloride Solution.

Kill 2 quarts hydrochloric acid with all the zinc it will take up. Add a quart of water or add part until zinc is fairly dissolved. A quart of glycerine which has been mixed with a quart of alcohol is then added to the solution.

This fluid is used for all kinds of soldering, especially with greasy and dirty connections and iron. It is claimed that glycerine prevents rust, which plays havoc with other soldering solutions containing hydrochloric acid. (*Street Railway Journal.*)

Zinc Chloride Paste.

1 lb. vaseline or petrolatum; 1 oz. (fluid) saturated solution of chloride of zinc.

The use of petrolatum instead of vaseline is recommended. While they are identical in composition, the name 'vaseline' is registered as a trade-mark, and commands a higher price on this account. Petrolatum is much cheaper. The chloride of zinc solution is made by dissolving as much zinc in strong muriatic acid as it will take up. An excess of zinc should be present, and all the acid neutralised. This will form a thick, oily solution. The petrolatum and chloride of zinc are mixed and thoroughly incorporated by means of a mortar and pestle or by vigorous stirring. The advantage of this soldering paste lies in the fact that it does not spatter and is not corrosive.

(ii) Ammonium chloride is used as a flux when soldering copper and iron.

(iii) Ammonium phosphate is used when treating with tin, zinc, copper and brass ware. It is non-corrosive and non-poisonous, but is somewhat refractory.

(iv) Hydrochloric acid is employed when dealing with zinc and zinc-coated ware.

(v) Lactic acid is useful in connection with copper and its alloys, but has the disadvantage of producing a tarnish on the surfaces adjacent to the soldered surfaces.

(vi) Venice turpentine and Gallipoli oil are used (separately) when soldering pewter or Britannia metal.

(vii) Russian tallow is employed when soldering heavy lead work.

(viii) Palm oil is employed in the treatment of light lead work.

(ix) Resin is an advantageous flux when subsequent corrosion is likely to be deleterious. It is used on soft alloys and tinware.

Hard Solders.

The hard solders may be divided into two groups—brazing or 'spelter' solders and silver solders.

Brazing or 'Spelter' Solders.

The basis of the brazing or 'spelter' solders ('spelters' they are commonly called) is brass. As the percentage of zinc is increased in the brasses so does their utility as solders for the more readily fusible metals and alloys increase. There is, however, a limit beyond which the addition of zinc to these alloys serves no useful purpose, owing to the fact that the resulting alloys are, on account of their brittleness and weakness, of no practical value. This limiting value is at about 45 per cent. of zinc. The particular alloy which is in most general use contains 50 per cent. of zinc and melts at about 875° C. (1607° F.). It is employed in brazing general brass work, which is possessed of melting points varying from about 890° C. (1634° F.) to about 1,000° C. (1832° F.). For brazing metals, etc., having higher melting points than the above, brasses containing a less percentage of zinc are used. Thus, for the soldering of general iron-work the brass containing 35 per cent. of zinc is used, while for the brazing of copper and light ironwork that having 40 per cent. of zinc is employed. The most malleable solder contains some 30 per cent. of zinc, but its refractory nature somewhat subtracts from its value.

For certain purposes, the above alloys, owing to their yellow colour, are prohibited. Occasionally—e.g. in the brazing of nickel silver and iron—whiter alloys are required. For this purpose alloys containing nickel are employed, of which the following are examples:—

Copper per cent.	Zinc per cent.	Nickel per cent.
37·50	50·00	12·50
35·00	57·00	8·00

Other white solders, for use in the brazing of brass and more fusible than 'spelters,' contain a percentage of tin. Two such solders are given below:—

Copper per cent.	Zinc per cent.	Tin per cent.
43·34	6·66	50·00
57·50	25·00	17·50

The British Standards Institution has issued a Specification (No. 263—1926) covering two grades of brazing solder. The copper contents for the two grades range from 53 to 55 per cent. and from 49 to 51 per cent. respectively, maximum limits being specified for the tin, antimony, arsenic, bismuth, iron and lead permitted in the alloy. A table is included as an appendix to the specification giving a range of grain sizes of granular solder to which it is recommended that the material be ordered.

Silver Solders.

Silver solders are employed for soldering gold, silver, copper, iron, and their alloys, the flux generally used being borax, though with very hard solders—*i.e.* solders having high melting points—powdered glass is sometimes of value.

Such as are suitable for general soldering and brazing, and containing an excess of base metal, are tabulated below:—

Copper per cent.	Silver per cent.	Zinc per cent.	Uses.
90—95	10—5	—	Solder for thin sheets of mild steel.
55·0	45·0	—	Alloy for instrument makers; tough and fluid.
50·0	30·0	20·0	Solder for small brass work.
46·0	38·5	15·5	Solder for bronze and nickel silver.
43·0	9·0	48·0	Solder for general work on copper alloys.

Tin-Lead Solders.

Tin and lead form a series of alloys, of which only those containing upwards of 35 per cent. of lead are employed as solders. These alloys may be divided into two classes, those that solidify at practically one definite temperature and those which are pasty over a considerable range of temperature before they become completely solid. The former are used as tinmen's solders, the latter as plumbers' solders, their value to each of these craftsmen being dependent on the properties associated with their solidification already mentioned.

It will be seen that fine tinmen's solder solidifies at a definite temperature of 180° C. (356° F.) while coarse tinmen's solder commences to solidify at about 209° C. (408° F.), passes through a pasty stage, and finally completely solidifies at 180° C. (356° F.). Soft solder is the name given to the alloy containing 50 per cent. of lead, whilst the fine plumbers' solder, used in making 'wiped' joints, which commences to solidify at about 250°C (482° F.), contains 66·6 per cent. of lead. A solder frequently used for the union of tinplate contains about 90·0 per cent. of lead, and hence has a very marked pasty range of temperature.

The appearance of the tin-lead solders affords some idea as to their value. The surface of good solder should be smooth and even, and rectangular castings of the same should have a distinct furrow along the centre of their upper surface. The presence of zinc, though only traces of this metal be present, causes the free surface of the solder to become minutely crystalline. Bismuth, copper, and silver do not affect the surface condition of solders when present in quantities varying from 0·2 per cent. in the case of the last metal to 1·0 per cent. in the case of the first, while antimony up to 2·0 per cent. improves, rather than otherwise, the surface condition. The presence of any of these elements leads to a reduction of the melting point of the solder.

To Prevent Solder Sticking to Work.

To prevent hot solder sticking to work, mix common whiting or cold-water paint with wood alcohol, and paint the part that is to be protected. The hot solder will not stick, even if the piece is held in the melted alloy. Water will do as well as alcohol to mix the paint; but alcohol is the most convenient, inasmuch as it can be used without waiting for the paint to dry. If water is used the paint must be thoroughly dry, as otherwise the moisture will cause the solder to fly.

Bismuth Solders.

These solders are alloys of bismuth, tin and lead, and are in general more fusible than the tin-lead solders. The useful alloys contain from 20 per cent. to 65 per cent. of lead, from 12 per cent. to 55 per cent. of tin, and from 12 per cent. to 55 per cent. of bismuth. The alloys containing the higher percentages of bismuth are the most fusible of these solders, which are employed for general pewtering. The solder containing about 50 per cent. of bismuth and 20 per cent. of tin has the lowest melting point of the series, its value being 96° C. (205° F.).

Aluminium Soldering.

Success in soldering aluminium depends on the effective removal from the metal of the microscopically thin film of oxide always present on the surface. When measures are taken to deal with this film the main difficulty of soldering is removed. Three different types of soldering are employed, which may be distinguished by the terms hard soldering, soft soldering, and reaction soldering. A proper flux, usually containing fluoride, must be used.

Hard Soldering.

In this process the solder consists of an alloy of aluminium having a melting point between 500 and 600° C. Many such alloys exist but the silicon alloy, containing 10 to 13 per cent. of silicon, appears to be the best. The oxide is removed by means of an alkaline halide flux, such as is used for aluminium welding. At the temperature at which the soldering is carried out the flux is melted and rapidly attacks the oxide, permitting the melted solder to come into contact with clean aluminium and alloy with the surface. In carrying out the process a gas blowpipe is used as heating medium, but apart from this and the high temperature required, the process does not differ from the ordinary soldering of brass. The flux is applied on the end of the solder stick, which is melted up and flows readily, sweating the parts together. It is necessary to see that the edges to be soldered are properly hot, and in some cases it will be found convenient to heat them first until they are hot enough to melt the solder stick when the latter is rubbed on them. Certain manufacturers supply silicon alloy solder in the form of a tube with the flux contained inside.

Hard soldering with a silicon alloy solder is thoroughly to be recommended as regards ease of application, strength, and permanence. Unlike soft soldering, the joint is capable of withstanding the action of boiling water or steam without protection.

Soft Soldering.

In this process the solder melts at a comparatively low temperature, and it is this type of work which has given rise to the widespread view that aluminium is difficult to solder. The reason is that no satisfactory flux has been available until recent times which will attack the oxide at the low temperature of working, so that the oxide had to be removed by mechanical means. After a preliminary cleaning, the metal is heated until solder melts upon it. The molten solder will not adhere but it can be made to do so by scraping through it with an old hacksaw blade or other form of scraper to break up the oxide film. Once the film is broken the oxide cannot reform under the solder, and alloying takes place. When the surface is fairly well covered with molten solder the adhesion is improved by rubbing with a wire scratch brush while still molten, thus breaking up any remaining traces of oxide. After such 'tinning' the parts can be sweated together in the ordinary way. Fluxes are sometimes supplied with these solders but these consist largely of stearin or resin and are of little assistance.

Aluminium Solders (Low-Melting Point).

Al.	Zn.	Sn.	Pb.	Cu.	Cd.	Melting point ° C.
—	50	46	1	3	—	195–350
5	8	87	—	—	—	264–375
—	60	—	—	—	40	270–358
—	20	36	44	—	—	185–369
9	8	78*	—	—	—	200–460
—	15–30	20–30	30–50	—	2–10	—

Reaction Soldering.

The solder is a chemical mixture containing halides and zinc chloride, which is spread on the parts to be jointed and heated by a blowpipe to about 200° C. A chemical reaction takes place which results in the deposition of pure zinc in a molten condition on the aluminium surfaces to be joined. The zinc flows readily between the edges and alloys readily with the aluminium, forming an excellent joint. Such joints are much more permanent than joints made by the ordinary soft soldering process.

Cold Solder (Amalgam Solder).

By shaking finely-divided copper with mercury in a little dilute sulphuric acid an amalgam can be prepared which may be used to unite joints at ordinary temperatures. The amalgam is formed into pellets, which are heated till mercury begins to exude from the surface. These pellets are wiped free from mercury and are then rubbed to a soft paste in a mortar. The surfaces to be united are first treated with sodium amalgam (sodium 2 per cent.), the soft paste of copper amalgam being then applied. The surfaces to be joined are then tightly pressed together, the union being completed in about three hours.

Soldering Aluminium Brass.

To solder aluminium brass with ordinary soft (pewter) solder: Cleanse well the parts to be joined from dirt and grease. Then place the parts to be soldered in a strong solution of sulphate of copper, and place in the bath a rod of soft iron, touching the parts to be joined. After a while a copper-like surface will be seen on the metal. Remove from bath, rinse quite clean, and brighten the surfaces. These surfaces can then be tinned by using a fluid consisting of zinc dissolved in hydrochloric acid in the ordinary way with common soft solder. If the surface is not coppered an aluminium alloy must be used.

Soldering Cast Iron.

(1.)

If the part of iron to be soldered is cast iron that is hard and thin, it should be polished on an emery wheel and made clean and bright. Then dip it in potash water, after which dip it for an instant in clean water, and wash quickly with undiluted hydrochloric acid of the ordinary strength; go over it with powdered resin and solder made from half tin and half lead. This must be done quickly, before the surface has time to dry.

(2.)

File the surface clean, and wash as before; wipe it over with a flux made of sheet zinc dissolved in hydrochloric acid until it is surcharged, or is a saturated solution and has been diluted with its own quantity of water; then sprinkle powdered sal ammoniac on it, and heat it on a charcoal fire until the sal ammoniac smokes. Dip it in melted tin, and then remove, and rap off the surplus tin.

* Plus 2 per cent. SnP.


SPARKLETS

CASTING AND FOUNDING.

Moulding-Sands.

(1.)

93 parts pure quartz sand; 7 parts clay free from lime; the latter acts as a cement for the former.

(2.)

For small castings:—Red sand, 3 parts by measure; coal dust (finely sifted), 1 part by measure.

(3.)

For large castings:—Red sand, 3 parts by measure; coal dust (finely sifted), 1 part by measure; chopped straw, or dried horse manure, ½ part by measure.

The latter makes the sand light and porous, facilitating the release of the gases. As the sand becomes impoverished by being burnt, the coal dust is renewed from time to time. If paraffin is used to damp the mould, black sand will bind and can then be dried as when red sand is used.

Moulding-sand which has partially lost its cementing power may have it restored by mixing with fresh sand.

Green, or floor sand that has 'gone dead' (dehydrated) is frequently revived with Mansfield sand.

When the metal touches the clay bond the clay is undoubtedly killed, but not so the ferruginous bond, and long-life sand is generally rich in the latter.

Ferruginous bond in moulding sand is usually the cause of the production of a smooth, sound surface to the casting, particularly in the case of steel.

The washes used for painting the mould have usually either china clay or a graphite substance as base. Graphite (plumbago) is itself used at other times. 'Scotch blacking,' which is a mixture of charcoal and graphite, is employed.

A suspension of the powder in water may be assisted by the addition of a little borax or molasses.

Cores.

For the making of 'cores' in British foundries a 'compo' moulding sand is made from ordinary sand such as Mansfield and broken fragments of graphite crucibles ground and mixed.

The admixture of the coarser material ensures good venting of cores; at times the sand alone is used.

Oil-bonded cores (where the oil is generally a commercial linseed, such as 'Baltic seed oil'), unlike those used in steel founding, have not met with much success in non-ferrous founding at home, but are adopted extensively in America, where a fairly coarse high-silica sand is the base. Sands from Lumberton and Howell, N.J., are so used.

(*Horobin.*)

Moulding-Sands

Sand.	Moisture Content. %	Green Bonded Strength. Riddled. Ins.	Green Bonded Strength. Milled. Ins.	Transverse Strength. Lbs.	Permeability. Secs.
Green facing sand	7·6	2·3	2·4	—	120
Dry facing sand	8·5	2·5	2·8	3·0	99
Black	7·5	1·8	—	0·75	141
Red	10·1	2·3	—	2·25	86

(*Morgan.*)

Core Sand Mixtures.

Mixture.	Moisture Content. %	Transverse Strength. Lbs.	Green Bonded Strength. Ins.	Permeability. Secs.	Remarks.
Ordinary core sand	8·2	2·25	2·3	106	Not free venting, but can be handled in the green state
Sea-sand 20 parts to 1 part linseed-oil	5	65	1·2	46	Cannot be handled in undried state.
Sea-sand 40 parts to 1 part linseed-oil	5	28	1·1	43	Suitable for most ordinary work
Leighton Buzzard 20 parts to 1 part linseed-oil	5	53	1·1	28	Binds better with treacle than linseed-oil
Leighton Buzzard 40 parts to 1 part linseed-oil	5	22	0·75	23	

Testing Moulding-sands.

A rough-and-ready test for voids and porosity consists in putting 50 c.c. of water in a glass measuring cylinder and 50 c.c. of the sand (previously passed through a 12-mesh sieve) into a graduated glass cylinder holding 100 c.c. The water is added to the sand and the cylinder with its contents is violently shaken until the water and sand are thoroughly mixed. The mixture is allowed to settle and the total volume is then measured. The reduction in volume is a measure of the volume of voids or pores in 50 c.c. of the sand.

By watching carefully, the time taken for the grains of different sizes to settle may also be noted.

(Searle.)

Sealing up Cracks in Castings and Cementing Joints.

Cracks in the walls of pressure vessels may be patched by bolting on a plate with a thickness of sal ammoniac and iron borings beneath it. Many an external firebox sheet has been patched in this way.

The sal ammoniac treatment has also been used in railway shops for stopping cracks in cylinders.

White lead has been used with advantage in making up threaded pipe joints and when pressing the brasses into locomotive driving boxes, placing driving pins in the wheels, etc. In the latter case the white lead acts as a lubricant while the job is being done, and later on, after it has set, it serves to cement the part pressed in. For many uses red lead is preferable to white, and next to litharge, is best for holding anything. If a threaded pipe joint is made up, say a 2 in., with white lead, after a year's time it will still be possible to remove the fitting, but if the same fitting is made up with red lead, and it is desired to take it off after the lead has hardened, the fitting will have to be broken unless sufficient heat can be applied to cause it to loosen from the pipe.

There is no cementing effect when graphite and oil is used on a pipe thread. With graphite and oil on a thread a fitting can be made up tighter than would otherwise be possible. Also, the graphite is squeezed compactly into the interstices in the joint and its very compactness holds it against blowing out under pressure, but it does not anchor the fitting. Therefore, no greater stress is necessary to remove the fitting at any time than was used to put it on.

Litharge will make a threaded pipe joint absolutely leak-proof against any kind of fluid and against any pressure that the pipe will stand.

(Power.)

Pattern Composition.

A composition that has many advantages over wood for making small patterns can be made as follows:—With hot water mix into a thick paste three parts by volume of starch, one part ground glue, two parts fine resinous sawdust. The sawdust should not be added until the starch and glue have been dissolved by the water. After the ingredients are thoroughly mixed, heat the whole to 190° F., and continue the heating until the whole becomes a hard mass, then allow to cool, and remove from the receptacle. The resulting composition is a strong, hard, horn-like substance, that can be machined, sand-papered and varnished just as can wood. The principal advantages of this composition over wood lie in the fact that it has no grain, and therefore turned and complicated patterns made from it do not have to be built up or glued together, and that for the same reason it is easier to turn and machine, and offers a smoother surface when finished. It is also more fireproof than wood and not so readily affected by atmospheric changes.

(American Machinist.)

Shrinkage of Castings.

The allowance for shrinking in casting should be for each foot in length :—

	Parts of an Inch.		Parts of an Inch.
For cast-iron pipes . .	·125 = $\frac{1}{8}$	Brass	·17 = $\frac{3}{16}$
„ beams and girders	·1 = $\frac{1}{10}$	Lead	·31 = $\frac{5}{16}$
„ cylinders, large .	·094 = $\frac{3}{32}$	Zinc	·25 = $\frac{1}{4}$
„ „ small .	·06 = $\frac{1}{16}$	Copper	·17 = $\frac{3}{16}$
In thick breadth . . .	·156 = $\frac{5}{32}$	Tin	·25 = $\frac{1}{4}$
In thin breadth . . .	·156 = $\frac{5}{32}$	Bismuth	·156 = $\frac{5}{32}$

Cleaning Castings.

A large proportion of the loose sand on castings may be removed in the 'tumbling' process but the adhering sand and scale of oxide, which soon destroy the cutting edge of the tools, are best reduced by a pickle ; the following have met with general approval :

Sulphuric Acid Pickle.—Ordinary commercial sulphuric acid and water in the proportion of 1 of the former to 30 of the latter. In this case the castings are required to remain in the pickle for about 10 hours, and are then washed with hot water to remove all traces of the acid and prevent rusting.

Hydrofluoric Acid Pickle.—This acid is slower in its action and not so liable to cause corrosion as the above. The castings, however, should be first tumbled in order to remove loose sand (as the sand neutralises the solution very much more quickly than the iron) and then immersed in a mixture containing 1 of hydrofluoric acid to 15 of water. The bath should be maintained at a temperature of 120° F. by a steam coil. The castings are then washed in clean hot water and sometimes in hot water with a small proportion of lime (about 1 lb. of lime to 8 gallons of water).

The baths are best made of oak, wood pinned, with no exposed metal fastenings and well lined with asphaltum. The attendant should be furnished with wooden clogs, strong leather apron and rubber gloves to protect him from the acid.

The acid should be added *very slowly* to the *water* and well stirred in the process. Never add the water to the acid.

An American factory is using high-pressure water for cleansing castings of sand and cores. A centrifugal pump, driven by a 50-h.p. motor, delivers 250 gallons per minute at a pressure of 250 lb. per sq. in., and the work is carried on in a closed chamber with glass windows. It is said that a casting which previously took a whole day to clean can be finished with the water jet in half an hour.

Dips for Castings.

Special for Electro-galvanising.—For preparing work for electro-galvanising the following can be used with advantage, but for no other solution except, perhaps, copper : Add to the dilute acid $\frac{1}{4}$ lb. of sheet zinc to each 25 gals. of solution, adding additional amount occasionally as it works out.

Bright Dip for Iron.—Work that has been pickled in the ordinary way to remove sand and scale, and then dried, usually has a dull-black appearance. If the work is to be polished, the black will be removed and the colour restored in the polishing process ; but if it is to be plated, or if it is to be left with parts of iron showing, the work should be run through the bright dip. This will bring it out white, improving the appearance and tending to make it take on a white deposit if it is to be nickelled. This dip is composed of :—

Hydrofluoric acid, 24 oz. Nitric acid, 10 oz. Metallic zinc, 2 oz. Water, 2 gals.

Work that is to be dipped should be strung on copper wire, and swirled around in the solution for a few seconds, rinsed in cold and then very hot water, and dried in sawdust. Articles should not be allowed to stay in the dip any time, as they would be spoiled and the solution ruined.

Satin-finish Dips.—Brass work requiring a satin finish may be dipped for a few seconds into a mixture consisting of :—

(1) Hydrofluoric acid, 1 part ; Water, 3 parts ;

then dried and run through the bright dip (4) to give it a lustre. If not satined enough the operation may be repeated.

(2) Hydrofluoric acid, 2 parts ; Nitric acid, 1 part ; hydrochloric acid, $\frac{1}{2}$ part ; Water, 5 parts. After which the work may be rinsed, dried, and run through the bright dip (4) if necessary.

(3) Hydrofluoric acid, 2 parts ; Nitric acid, 1 part ; Water, 10 parts.

The following is a *bright dip* for copper, brass, bronze, or German silver :—

(4) Hydrofluoric acid, 4 quarts ; Nitric acid, 3 quarts ; Common salt, small handful.

Die-Casting.

Die-castings have been defined as castings produced by pumping metal into metallic dies at a pressure higher than that of the atmosphere. This distinguishes them from the class of castings made by the permanent mould process into which the metal is simply poured, also sometimes referred to as die-castings and on other occasions as chill castings. There is, however, a tendency now to make the phrase 'die-casting' embrace both the methods, and to distinguish the sections as 'gravity die-castings' and 'pressure die-casting.'

The equipment for die-casting includes a melting pot (usually of cast iron), gas-heated, placed near the die-casting machine, for supplying the molten metal to the reservoirs of the caster.

The casting machines are of two general classes, those in which the metal is forced into the die or mould by the action of a plunger or pump, and the other in which compressed air is used for this purpose.

The reservoir attached to the machine is heated by gas, the pump being in direct contact with it; the plunger is actuated, by means of a long lever, by the operator, the metal being forced from the pump chamber through a suitable nozzle into the dies. The moulds are attached to a hinged plate which swings away from the reservoir and pump nozzle, after each cast, to allow the workmen to remove the tang and clean the metal passages.

The swinging plate is securely gripped to the reservoir at the moment of casting.

The mould is held to the swinging plate by means of a toggle joint, and varies in form and design according to the particular castings; it is always at least a two-part mould, but usually is also fitted with elaborate cores and drawbacks and mechanism for working them, which make the production of complex castings possible.

The dies are made in alloy steel, and occasionally in cast iron, according to the alloy and character of the work in hand. Sometimes the dies are heated, for other work they are required to be water-cooled, so that no hard and fast line can be set in this respect.

The compressed-air type of die-casting machine consists of an air-tight cast-iron chamber for the metal, suitably heated to keep the metal at the desired temperature. This chamber is connected with the die by one opening, and with the compressed-air line, by another. In the latter opening a valve is fitted with a lever for the sudden release of the air into the metal chamber. This valve also permits of a gradual decrease of the pressure on the metal after it has filled the die. This type of machine may be used on metals cast by the plunger type, but is especially suitable for metals that cannot be cast in the plunger type of machine on account of their high melting-point or corrosive action on the cast-iron plunger.

A die-casting die consists essentially of two parts which are brought together and maintained in perfect register by means of the usual dowel pins.

Internal threads are produced by screwed cores and external ones by removable screwed collars, whilst the holes which are usually drilled in machine-made fittings are in these cases formed by means of retreating cores; the latter may be of almost any shape.

Small cores are operated by levers, large cores by a rack and pinion; they are always locked whilst the casting operation is proceeding. A vital consideration is the proper venting of the die; the position and form of the vents can only be determined by experiment.

Great accuracy is obtainable; it is quite possible to produce castings to a limit of 0·001 inch.

(*A. H. Mundey*)

Alloys used for Die Castings.

The alloys generally used are of considerable variation, according to the special purpose and circumstances; those of the white-brass or zinc-base group may contain approximately:—

86 per cent. zinc. 7 to 10 per cent. tin; 4 to 7 per cent. copper; 0·5 to 1·0 per cent. aluminium.

Increased tin gives greater ductility, whilst increased copper renders the alloy harder.

Mazak alloys, made from 99·99 per cent. zinc, contain 4 per cent. aluminium and 0·03 per cent. magnesium. Some contain copper up to about 1 per cent.

The tin-base alloys are (*a*) bearing alloys of all the usual grades of white anti-friction metal, and (*b*) alloys specially designed to withstand the corrosive effect of chemicals in food-stuffs, certain gases, and even acids.

An alloy which is largely employed is:—

90 per cent. tin; 4 per cent. copper; 6 per cent. antimony.

Of the aluminium group the most commonly used alloy is that known as No. 12, consisting of

92 per cent. aluminium; 8 per cent. copper.

Sometimes a small quantity of magnesium is added and also frequently from 1 to 3 per cent. of nickel.

The finish or skin of the casting is smooth and in a large measure satisfactory, but polishing, plating, and bronzing are frequently employed, for protective as well as decorative purposes.

Shrinkage is an important consideration; this varies greatly with the alloys used. Approximate allowances are:—

White brass or zinc base alloys . .	0·0035
Lead base alloys	0·002
Aluminium alloys	0·007

The alloys generally used for die-castings are of relatively low melting-point and are of such a constitution that they have a period of plasticity during the process of solidification. This adds much to the chance of the production of sound castings.

Temperature of working is an important factor; if too high, splashing results, with production of long fins, oxidation, and discomfort in working. Too low temperature produces bad finish with little folds and creases on the surface, due to premature solidification.

The metals known as yellow metals—that is, high copper alloys—have hitherto been cast almost exclusively by the gravity process. In these cases great accuracy is not obtainable for long runs, owing to the wear of the dies at the high temperatures employed.

A great advance has taken place, however, quite recently in the introduction of several processes for the production of yellow metal die-castings of very high finish, having high mechanical properties, by pressure machines.

The yellow metal alloys usually employed for gravity castings are: copper 90 to 95 per cent., aluminium 10 to 5 per cent.; also copper 60 to 65 per cent.; zinc 40 to 35 per cent. For pressure work the series of alloys of the latter class are mostly employed.

See also Descriptive Section XXIII, Part 1

James Booth & Co., Ltd.
Bull's Metal and Melloid Co., Ltd.
Fry's Metal Foundries, Ltd.
Heenan & Froude, Ltd.
Sparklets Ltd.

SECTION XXIII

PART II

WELDING AND CUTTING.

(Contributed by D. Richardson, Wh. Exh., A.M.I.Mech.E.)

OXY-ACETYLENE WELDING.

The term 'autogenous welding' is a cumbersome and loose description for the process when its real meaning 'self-producing,' does not fully apply. The term was chosen to describe a process in which two sections of metal were melted together by means of a blowpipe in contrast with the then existing methods, in which a joint was made by the addition of a different metal, as in brazing. The term 'fusion' is considered to be much more appropriate than 'autogenous,' and is applied to a weld worked in the fluid or vapour state.

The easy control and intensity of the oxy-acetylene flame, the low cost of operation, the relatively inexpensive equipment required, the range of metals which can be welded, and the adequate supplies of carbide of calcium, oxygen, etc., available, are factors which have led to its many and varied applications.

The industrial development which has taken place can also be attributed to the increased efficiency of the welder and the efforts that have been made to reduce the art of welding to a more scientific basis. The uncertainty of the results, upon which opposition and criticism are usually based, can, in general, be traced to one or more of the following causes: (*a*) inefficiency of the welder; (*b*) use of defective or unsatisfactory apparatus; (*c*) use of defective or unsuitable filling materials; (*d*) lack of proper supervision and maintenance of the generating plant and equipment; (*e*) low weldability of the material.

It is not difficult to eliminate these causes, and a well-trained welder using good apparatus, correct filling materials, etc., can produce welded joints on mild steel, alloy steels, wrought iron, cast iron, copper, brasses, bronzes, aluminium and aluminium alloys, lead, etc., with mechanical and physical properties approaching those of the parent metal, and in which the strength and service of the joint are absolutely reliable.

Equipment.

The apparatus required for an oxy-acetylene installation depends upon several factors, such as (*a*) the class of work to be welded; (*b*) whether a portable or stationary installation meets the case; (*c*) whether production or repair work, or both, are possible applications; (*d*) facilities for installing equipment and for handling the work in prospect. In general, the apparatus and accessories will comprise a generator for generating the acetylene gas, which may be a fixed or portable plant—the most portable type of plant uses compressed or dissolved acetylene in portable cylinders instead of a generator; chemical and mechanical purifying apparatus, when the gas is obtained from a generator; oxygen apparatus for the production of oxygen, or oxygen in cylinders; oxygen regulators for lowering and regulating the pressure of the gas after it leaves the cylinder, to the pressure required for the blowpipe, and, in the case of dissolved acetylene, an acetylene regulator for the same purpose; hydraulic safety valve or other safety device for dealing with explosive mixtures of oxygen and acetylene or air and acetylene; welding and cutting blowpipes of various capacities; accessories, such as flexible tubing, goggles, welding benches, blowpipe support, preheating devices, tools, jigs, fluxes and filling materials, carbon paste and blocks, fire protection, protective equipment for the operator, etc., etc.

Fig. 1 shows in diagrammatic form an elevation and plan of the general layout of an acetylene welding and cutting installation. The letters refer to the same details in both views.

The acetylene generator, shown at A, is usually automatic—that is, the generation of gas is controlled entirely by the gasholder regulating the water supply or the carbide supply; B is

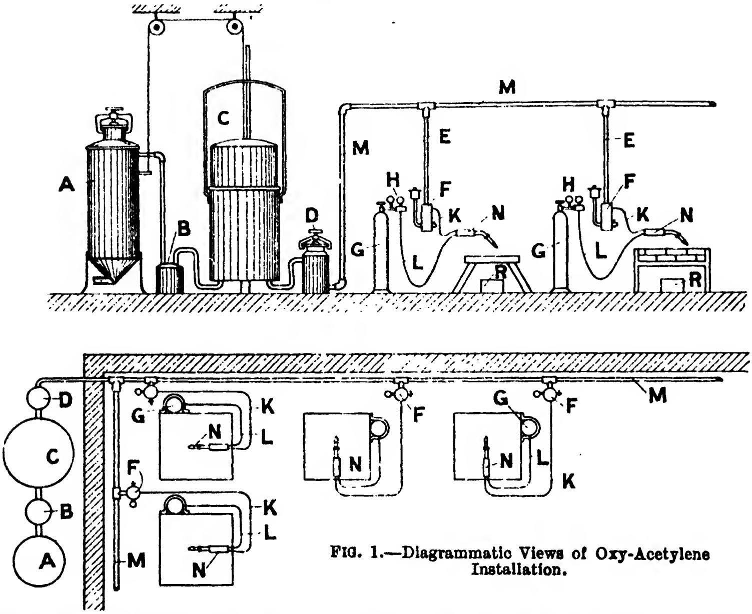


FIG. 1.—Diagrammatic Views of Oxy-Acetylene Installation.

a condenser, washer, and water-seal; C is a gasholder, consisting of a tank to contain the water-seal and a bell to store the gas; D is a purifying chamber and filter. The main piping, M, is connected to the purifying chamber and usually forms a complete circuit of the welding shop; branch pipes, E, are taken from the main for supplying acetylene to each welding station; F is the

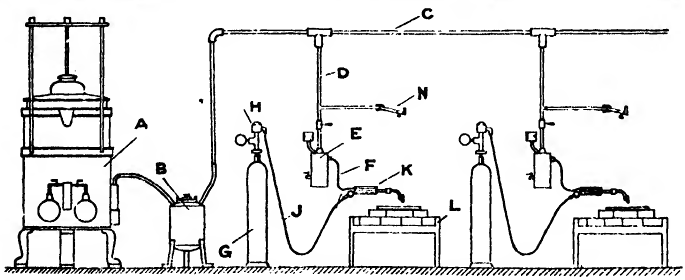


FIG. 2.—Diagrammatic View of Low-Pressure Oxy-Acetylene Installation.

hydraulic safety valve or other safety device; a flexible tube, K, connects the hydraulic safety valve to the welding or cutting blowpipe, N; G is the oxygen cylinder, on to which is screwed the oxygen regulator H; a flexible tube, L, connects the oxygen supply to the blowpipe N; two common types of welding benches, with blowpipe cooling tanks, R, are shown.

For comparatively small installations, the generator, condenser, washer, water-seal, and gas-holder form a single piece of apparatus, as shown at A in fig. 2. The diagram shows at N a pilot light bracket attached to the branch tube D. An acetylene jet burner consuming about 5 litres of acetylene per hour, screwed into the bracket, forms an efficient and economical method of lighting the blowpipe. The burner should be at least 18 inches from the hydraulic safety valve and not more than 6 ft. above floor level. If arrangements are not made for an acetylene or coal-gas blowpipe lighting circuit a friction lighter should be used, not matches. The advantages of a gas economiser support for the blowpipe should not be overlooked, especially in production welding.

An installation of the type described is known as a low-pressure installation, to distinguish it from the type in which the acetylene is obtained from cylinders, known as the high-pressure system.

Fig. 3 shows the equipment for the high-pressure system. After passing through suitable acetylene and oxygen regulators the gases are conveyed to the blowpipe by means of flexible tubes.

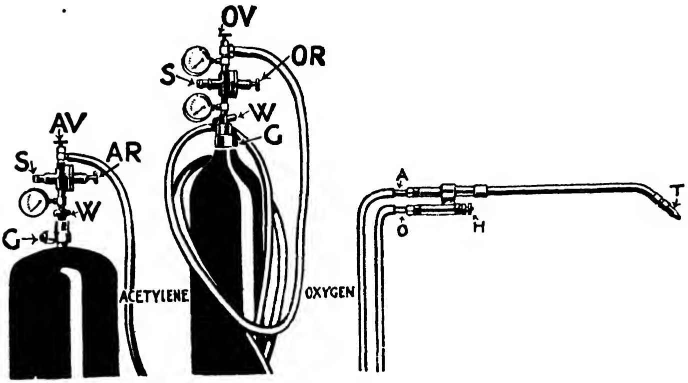


FIG. 3.—High Pressure or Dissolved Acetylene Installation.

OXYGEN SUPPLY.

The oxygen may be obtained in cylinders or may be produced on the premises. There are three methods of production, namely :—(1) chemical processes ; (2) the electrolytic process ; (3) the liquid air process. The main objections to chemical processes, which are now seldom used in welding and cutting processes, are the cost and impurity of the gas. The effect of impure oxygen in welding is to retard the welding operation, and thus increase the cost, and a tendency to produce defective welds. In cutting operations the purity of the oxygen is a most important factor, and tests have shown, for example, that with oxygen of 91 per cent. purity the cutting speed is halved when compared with oxygen of 98 per cent. purity. Simple and cheap apparatus is available for rapidly and accurately testing the purity of the gas.

The electrolytic and liquid-air processes may be used for producing oxygen on the premises. The gas can then be charged into cylinders or piped into the welding shop from large receivers into which it has been compressed, or it may be piped direct from the oxygen-producing plant. The problem of distributing oxygen in a liquid state instead of by the existing method of distributing the gas in steel cylinders at 120 to 150 atmospheres pressure has been solved and the method is in use.

Oxygen produced by the electrolytic or liquid air processes has a purity of the order 99 per cent. Unless the consumption is very large the first cost of an oxygen installation is comparatively high, and the majority of users obtain the gas in cylinders from the manufacturers. The cylinders supplied by the manufacturers for welding and cutting usually contain 200, 175 or 100 cu. ft. of gas at a pressure of 1,800 lbs. per sq. in. (120 atmospheres). The Gas Cylinders Research Committee of the Department of Scientific and Industrial Research have made a series of recommendations for ordinary commercial cylinders used for the storage and transport of gases including oxygen.

They have issued four reports dealing with: (*a*) periodical heat-treatment of gas cylinders; (*b*) alloy steel light cylinders; (*c*) cylinders for liquefiable gases.* An Oxygen Research Committee have also issued a Report dealing with the storage and transport of liquefied gases and a summary of the recommendations relating to compressed oxygen.† The British Standards Institution have issued specifications for cylinder valve fittings, identification colours for cylinders and steels for cylinders.‡

In handling oxygen cylinders precautions must be taken against leakages, and particular care must be exercised not to oil or grease any part of the oxygen apparatus or supply line. Dust and moisture should be expelled by opening the cylinder valve before the cylinder is put into use. Repairs to cylinder valves should not be attempted by the user, and full cylinders should be handled carefully. All oxygen cylinders are fitted with valves having right-hand connections, and are generally painted black. No attempt should be made to transfer oxygen from one cylinder to another by rigging up some form of communicating tube. In the majority of cases a separate cylinder is used for each welding station or blowpipe in use. The arrangement of manifolding oxygen cylinders is being increasingly used as the handling of single cylinders, and wear and tear on the pressure regulators, is largely obviated. Oxygen manifolds should be able to withstand a pressure of at least 3,000 lbs. per sq. in., and each cylinder connection should be equipped with a high-pressure shut-off valve. It is extremely advisable that the necessary manifolds be purchased from a responsible manufacturer or made to specification. A cylinder valve should never be opened suddenly when the oxygen regulator is attached. In the event of the oxygen supply being restricted owing to the freezing of moisture in the cylinder valve, hot water only should be used to put it right. Leakages can be discovered by the hissing noise produced by the escaping oxygen, by the use of soap-suds spread over likely places, or by closing the valves on the blowpipe, cylinder, and regulator, and noting how long the pressure holds up on the gauges. Cylinders should be placed where they cannot be easily knocked over, and never near sources of heat. Spare cylinders should be safely and properly stored, care being taken not to store them near radiators, oil, grease, or combustible materials.

For estimating welding costs, the following table will be found useful. The table gives the number of cubic feet of oxygen remaining in a 100 cu. ft. cylinder corresponding to various readings of the high-pressure gauge on the oxygen regulator. The amount of oxygen indicated by the gauge reading is more or less approximate, as it depends upon the temperature of the oxygen. For practical work the deviations can be ignored, and would not affect the usefulness of the table. In general, the volume of oxygen in a cylinder is nearly proportional to its pressure, and thus, in the absence of the table, the volume of gas remaining in a cylinder can be determined by observing the gauge reading. A regulator provided with a pressure gauge which serves the double purpose of registering the pressure of the gas in the cylinder, and indicating at any period the quantity of gas in the cylinder, can be used.

Gauge Pressure in Atmospheres.	Cubic Feet of Oxygen in Cylinder.	Gauge Pressure in Atmospheres.	Cubic Feet of Oxygen in Cylinder
10	8·3	70	58·4
20	16·7	80	66·7
30	25·0	90	75·0
40	33·4	100	83·4
50	41·7	110	91·7
60	50·0	120	100·0

At atmospheric pressure and temperature oxygen is rated at 12·08 cu. ft. per pound

Oxygen is supplied to the trade at prices which vary according to the size of the cylinder, quantity purchased, and other conditions. Cylinders are loaned, or may be purchased outright. Demurrage charges are made on hired cylinders when retained above a certain period. The cylinder valve should always be closed when the cylinder is not in use and when returned for re-filling. Empty cylinders should be handled with the same care as for full cylinders. The proper care of cylinders and valves and their prompt return to the filling station are important factors in obtaining cheap oxygen. Oxygen cylinders are conveyed by the railway companies under strict regulations and conditions.

* *Four Reports of the Gas Cylinders Research Committee* (1921–1929).

† *Report of the Oxygen Research Committee* (1929).

‡ *British Standards Institution*, Publications Nos. 341, 349, 399–401.

Acetylene Supply.

The acetylene may be obtained compressed in cylinders filled with a porous material, the pores of which hold a film of acetone in suspension, it is then known as dissolved or compressed acetylene, or may be generated on the premises. Before the gas is compressed into cylinders it is thoroughly purified and dried. The precautions in handling acetylene cylinders are much the same as for oxygen cylinders. The gas should not be drawn off at too high a rate—in other words, the greater the demand the larger the cylinder. For heavy continuous work two or more cylinders should be manifolded together by means of special adaptors supplied by the manufacturers. For general work cylinders containing 60 to 275 cu. ft. of gas are advisable. For light work such as thin sheet metal, lead burning, etc., small cylinders can be used with advantage. All acetylene cylinders used for oxy-acetylene welding are provided with valves screwed with left-hand threads, and are painted, or partly painted, a dark reddish colour. Cylinders for welding and cutting can be hired or purchased. Always close the valves of empty cylinders. Stocks of oxygen and acetylene cylinders, whether full or empty, should not be stored in the same place unless there is a fire-resisting partition between them. Cylinders should be kept far enough away from the actual welding or cutting operation so that sparks or flame will not reach them. Acetylene should not be supplied from a battery of cylinders to a system of shop piping without consulting the suppliers of equipment.

The acetylene and oxygen being under high pressure, a perfectly uniform mixture of both gases can be maintained in the blowpipe and the pressure varied according to requirements. The elimination of the injector device, necessary in low-pressure blowpipes, and the hydraulic safety valve are advantages claimed for dissolved acetylene. Dissolved acetylene can be used with any type of oxy-acetylene blowpipe, but it should be recognised that the device for dealing with back-fires and flash-backs does not, as a rule, form part of the details of low-pressure blowpipes.

The amount of acetylene in a cylinder cannot be determined, as in the case of oxygen, from the pressure gauge reading. The only accurate method of calculating the gas used from, or left in, a cylinder is by weighing before and after use. For calculating the volume of acetylene used the following are the equivalents:

1·1 oz. of acetylene = 1 cu. ft. (approx.).
1 lb. ,, ,, = 14·5 cu. ft. ,,

By means of an acetylene generator acetylene is produced by the interaction of calcium carbide and water. There are three general types of acetylene generators which may be classed under the heads of water to carbide, carbide to water, and the dipping or displacement type. They can also be classified as low-pressure and pressure generators. Briefly, low-pressure generators deliver acetylene to the blowpipe at a pressure of about 8 ins. water pressure (0·036 lb. per sq. in.), whilst pressure generators deliver the gas up to a maximum fixed by the Home Office at 250 ins. water pressure (9 lbs. per sq. in.).

The design and layout of a generating plant must be such that the acetylene reaches the blowpipe in a good condition, in sufficient quantity and at sufficient pressure, otherwise the welds may be porous, brittle, and burnt. The acetylene generator should be of good and solid construction, and the gasholder capable of storing all gas generated under working conditions or after stoppage of work. A safety tube, discharging into the open air, should be fitted to gasholders. There should be no signs of polymerisation or overheating of the gas even when working at the maximum delivery. The cross-section of tubes, pipes, and passages through valves should be such that the gas flows normally at the maximum delivery without appreciable loss of pressure. The pressure should be as high as possible, and at least equal to 6 ins. of water pressure in the case of small plants, and 8 to 10 ins. in larger plants. The pressure should remain practically constant during working and not vary by more than ½ in. The charging and cleaning of the plant should not result in any appreciable waste of gas, and be made without introducing sufficient air to interfere with the proper working of the blowpipe. The generator should comply with the Regulations of the British Acetylene Association. The size of carbide suitable for the type of generator should be used. For commercial purposes carbide is broken and packed in the following sizes: 80/120, 50/80, 25/50, 15/25, 7/15, 4/7, 2/4, 1/2 millimetres. The figures refer to the approximate mean thickness of a lump in millimetres. The 50/80 size is in general use for welding and cutting generators. The choice of carbide-to-water, water-to-carbide, etc., generating plants depend on several factors, such as location, class of work, etc.

The Carbic system of generation is based upon the use of a carbic cake, which consists of specially prepared and treated carbide compressed into cylindrical cakes. It is claimed that the process renders every particle of carbide impervious to moisture, so that generation can only take place while water is in actual contact with the cake. Freedom from danger in handling, and from after-generation, the convenience in handling, elimination of waste, and purity of gas are advantages of this system.

Overloading of the plant is a common cause of defective welds. In carbide-to-water generators an allowance of not more than 1 cu ft. of acetylene per hour for each pound of carbide in the rated charge is recommended. In the case of water-to-carbide generators the maximum rate of gas consumption should be half this value. In well-designed generators the yield of acetylene gas from the generator will be 85 to 90 per cent. of the carbide yield. Practically, for every 10 lbs. of carbide used in the generator a yield of 40 cu. ft. of gas at the blowpipe can be assumed.

The proper care, cleaning, charging, and maintenance of generators is of first importance if sound welds are required. Leakages of acetylene should be traced by means of soap-suds and never with naked lights.

The acetylene generating plant should be installed in a well-ventilated shed or building, or in a lean-to having no direct communication with a workshop or adjacent building, and no unauthorised person should be allowed to enter the building or enclosure. Where artificial light is required in a generator house or near a generator, electric incandescent bulbs, encased in outer gas-tight globes, should be used. A conspicuous notice prohibiting lights or smoking should be exhibited on the outside of the generator house. Generators should be cleaned and charged in daylight. Electric switches, fuses, etc., should be placed outside a generator house, or they should be so constructed that risks due to sparking or fusing are avoided. To comply with existing regulations, a licence under the Petroleum Acts must be obtained from the local authority for the storage of carbide. A sliding scale of licence fees, varying from 5*s.* to £5, according to the quantity stored, has been fixed, and severe penalties for contravening any condition printed or endorsed on the licence by the local authority are included. The London County Council have issued a circular setting forth the conditions which acetylene apparatus should fulfil before being approved for use on licensed premises.* A raised platform in the generator house should be provided for the storage of carbide. Drums containing carbide should not be placed in the welding shop or near a flame. Care must be exercised in opening drums to avoid striking sparks; a proper drum opener should be provided for this purpose. When the contents of an opened drum are not entirely used in charging the generator, the remainder should be transferred to a storage bin which can be sealed.

All users and prospective users of oxy-acetylene equipment should consult the memorandums issued by the Factory Department of the Home Office† and the Board of Trade.‡

Purification of Acetylene.

The chief impurities found in acetylene gas are (*a*) those which affect the quality of the weld, such as sulphur, phosphorus, silicon, finely subdivided lime; and (*b*) those which affect the cost of the weld, such as hydrogen and hydrogen compounds, nitrogen and nitrogen compounds, and water vapour. An efficient chemical purifer must be provided capable of removing sulphur and phosphorus compounds, the acetylene must pass through an efficient water-scrubber to remove or partially remove the impurities soluble in water and, finally, the gas should be filtered to remove solid impurities not eliminated by washing or chemical purification. A first-class purifying material will remove chemical impurities, moisture, and lime dust which are a common cause of defective welds.

Purification of the gas must be efficient, and the care and maintenance of the purifying chamber is very important. The cross-sectional area of the purifier should be ample, and the depth of purifying material should not exceed that specified by the makers, otherwise loss of pressure and overheating of the gas will take place. The quality of the gas can be simply and quickly tested by the use of a white absorbent paper, saturated with a 10 per cent. solution of silver nitrate, exposed to the gas. Should the paper turn black after a few seconds' exposure, the gas is unsuitable for welding purposes.

Among the best known purifying materials are Heratol, Catalysol, and Suder. The two last are distinguished by their regenerative properties—that is, they recover their activity by simple exposure to the air, and thus can be used over and over again. In purifying acetylene the cost should be based on the volume of gas purified and not on the first cost of the material.

Preservation of Acetylene Apparatus.

The question of preventing the corrosion or deterioration of the material used in the manufacture of acetylene generators, purifying vessels, and other parts of the generating plant is important in view of the fact that according to the efficiency of the coating the life of the plant may be a question of months or years.

There are five cases to be considered, namely: (*a*) preservation of the external parts of the apparatus; (*b*) preservation of the immersed parts; (*c*) preservation of parts in contact with moist and impure gas; (*d*) preservation of the interior of purifying vessels; and (*e*) preservation of carbide containers. With regard to (*a*), good ironwork paints give good results when applied well to properly prepared surfaces. At least two coats should be applied. If the apparatus is exposed avoid paints used for woodwork, plaster, etc. For (*b*) it is necessary to use a waterproof paint which will resist the prolonged action of water. The best results are obtained when the base consists of tar and graphite. For (*c*) the paints used for immersed parts of acetylene apparatus give the best results, but the work must be done with great care. For (*d*) the majority of industrial preservative paints give satisfactory results, except those containing a zinc or similar base. Several coats are necessary, and great care is required in applying the coats on to well-prepared surfaces. For (*e*) tests have shown that no paint or coating will satisfactorily resist the action of the lime residue at the temperatures usually present, and that the lime residue appears to have a preservative action on the parts with which it is in contact. It is therefore unnecessary to protect these parts.

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* 'Carbide of Calcium,' L.C.C. Public Control Department (Circular 80).

† Dangers from the use of acetylene gas and in oxy-acetylene welding in factories (Form 1704).

‡ Instructions to Surveyors. Repairs to Boilers (Circular 1596).

Oxygen and Acetylene Regulators.

Regulators in common use to reduce the pressure of gases are of the diaphragm type, and may be roughly divided into two classes—first, where the pressure of the gas closes the inlet, and, second, where the inlet is closed by a spring. They are designed to deliver automatically gas to the blowpipe at any uniform pressure to which the regulator is set. They should be designed for working over a wide range of pressures, and should be sensitive even at low pressures. Oxygen regulators are usually equipped with two gauges, a high-pressure gauge, which shows the pressure of gas in the cylinder, and in some types the quantity of gas in the cylinder, and a low-pressure gauge. Pressure gauges should be provided with a check to prevent a sudden inrush of gas. For successful welding the regulator should be provided with a low-pressure gauge, giving the pressure at which the gas is delivered to the blowpipe. The range of the low-pressure gauge should not be too great, and a maximum reading of 75 lbs. per sq. in. will be sufficient for the majority of welding and cutting operations. Regulators must be kept in good order, otherwise defective welds are inevitable. Regulators intended for use when welding should not be used when cutting unless the work is within the range of the regulator. It is dangerous to use oil, grease, or any organic material for lubrication. If it becomes necessary to lubricate the pressure-regulating screw, or to repack a needle valve, a little glycerine should be used. The pressure-adjusting screw should always be released after ceasing work and before changing from one cylinder to another. Creeping regulators or regulators with leaky valve seats should be overhauled, because they are likely to become dangerous and uneconomical. The gradual increase of pressure when the blowpipe is shut off indicates a creeping regulator. Devices in the form of electric heaters, which prevent freezing of regulators and thus interference with the supply of oxygen, can be used and are compact and convenient where electric current is available. They are interposed between the cylinder and the regulator. Regulator gauges should be tested at regular intervals but never tested with an oil-testing pump.

Acetylene regulators are usually painted red and have a left-hand screw thread corresponding to that on the standard valves of acetylene cylinders. Regulators for oxygen are usually painted black and have a right-hand screw thread. Any possible confusion is obviated by this distinction. Before connecting an oxygen regulator to a cylinder open the oxygen cylinder valve for an instant and then close it.

Cylinder valves should be opened very slowly so that pressure does not suddenly come on to the gauges or diaphragm. The proper working of regulators largely depends upon the care given them. They should not be left in places where dirt or dust may get into the interior and be deposited on the seat or inlet nozzle, and they should never be repaired by inexperienced workmen.

Hydraulic Safety Valves.

These are indispensable in low-pressure and medium-pressure installations, i.e. where the acetylene is generated on the premises. Their function is to provide a water-seal and to prevent explosive mixtures entering or forming in the acetylene piping or generating plant. The valve should be chosen so that the maximum supply of gas is obtained without burning the metal. Safety devices based on mechanical principles, such as valves, etc., may more or less prevent the return of the flame, but unless properly designed are not infallible in preventing explosive mixtures forming in or reaching the generating plant.

Hydraulic safety valves should be charged and examined daily, and should be installed so as to facilitate filling to the proper level. The local authority responsible for issuing the carbide licence generally insists on the provision of a metal shield for the valve, to avoid accidents in the event of the valve being fractured by an explosion. In general, the valve should be charged with water whilst the acetylene tap admitting gas to the valve is open. The design should be such that no water is carried over with the gas. An investigation has shown that the water vapour contained in the gas after passing through the hydraulic valve rises with the temperature, and can attain 7 per cent. when the gas is at 97° F. With 2·5 per cent. of water vapour, there is a reduction of over 8 per cent. in the speed of welding. No safety valve should have more than one outlet to which a blowpipe can be connected. Flexible tube should not be used to connect the hydraulic valve to the acetylene generator.

Pressure relief valves or safety valves used on pressure generators should be tested periodically; they do not replace the hydraulic safety valve, which is still required in a modified form.

Piping and Flexible Tubes.

The acetylene piping should be large enough to avoid loss of pressure. The size of piping should be based upon its length and the maximum supply of gas required. All piping should, when possible and convenient, be run overhead. When piping has to be run underground it should be placed in properly ventilated trenches. Piping and connections should be installed by workmen experienced in the installation of acetylene apparatus. Piping should be so arranged that any moisture will drain back to the generator, and where low points cannot be avoided these must be drained through drip cups closed with screw caps or plugs. No pet cocks should be used. A main gas cock should be arranged where it can be conveniently reached and near the point of entrance, so as to allow for shutting off the gas supply from the workshop. Gas cocks should be provided on the branch tube leading to the hydraulic valve so that a valve may be removed without discharging the line. All cocks, valves, unions, and joints should be full bore, and the installation should be thoroughly tested when the system is completed, and at regular intervals. It should not show a loss of pressure exceeding two inches of mercury within twelve hours when subjected to a pressure equal to a column of fifteen inches of mercury. Acetylene piping should be painted a distinctive colour. Galvanised iron pipe with welded or threaded connections should be used for piping acetylene. The pipes should be of uniform diameter, and bends—the number of which should be as small as possible—should be used in preference to square elbows. Compressed oxygen should never be used for testing or cleaning acetylene piping. If compressed air is used for this purpose the acetylene plant should be disconnected.

The following table will serve as a guide in installing acetylene piping for welding purposes. Manufacturers of welding equipment will supply full particulars as to pipe sizes suitable for any particular layout.

Acetylene Main Piping.

Approximate Diameters of Service Piping. For working pressure of 6 *in. to* 10 *in. water column.*

Consumption of Acetylene per hour.	Length of Pipe.	Internal Diam. of Piping.
Cubic feet.	Feet.	Inches.
15	100	½
40	100	¾
80	100	1
140	100	1¼
225	100	1½
400	200	2
1000	200	3
2500	200	4

Ordinary black steel piping may be used for low-pressure oxygen, and extra heavy steel pipe for high-pressure lines. The suitability of steel piping for this purpose is doubtful, and some manufacturers recommend copper piping from the point of view of safety and convenience. It is advisable for prospective users of oxygen pipe lines, headers, and manifolds to purchase them from, and have them installed by, reliable manufacturers familiar with the correct shop practice with reference to their construction and installation. No oil or grease must be used for the connections; litharge and glycerine should be used. The diameter of oxygen piping will depend on the length of the lines. A ¾-in. pipe is suitable for supplying twenty welding stations, provided the length of the line is not more than 300 ft. and the maximum consumption is not more than 500 cu. ft. per hour.

The flexible tubes conveying the gases to the blowpipe should not be too long, too flexible, or too rigid. The strength should be such that for welding purposes the tube should withstand a pressure of 100 lbs. per sq. in. without bursting, swelling, or leaking, and for use in cutting operations it should withstand 300 lbs. per sq. in. Armoured or wired hose is not necessary or recommended for ordinary welding and cutting operations. Three-ply tubing with a core of pure rubber is satisfactory for ordinary work, and for welding and cutting in confined spaces fire-resisting hose should be used. Avoid greasing the ends of the tube to facilitate fixing to the blowpipe or other connectors; use water. No white lead, oil or grease, or other pipe-fitting compounds, should be used for making tight joints. Tube clips should be used in preference to wire, etc., for securing the tube firmly to the connectors.

Flexible tubing should be examined regularly, and defective tubing should never be used. Avoid tangling or kinking the hose, long lengths—about 10 ft. lengths are suitable for stationary equipment and 15 ft. for portable—dragging the hose along the floor, and exposing it where it is likely to get damaged. Special care should be taken to avoid the interchange of the oxygen and acetylene tubes when connecting up to the blowpipe. Both acetylene and oxygen tubing should be blown through occasionally, so that dirt and dust will not get carried into the blowpipe, and tubing should be examined and tested periodically.

Welding Blowpipes

Welding blowpipes are classified in relation to the pressure of the acetylene. Low-pressure blowpipes are those fitted with an injector to aspirate the acetylene, which is supplied from a generator under a pressure of 6 to 12 ins. of water. Medium-pressure blowpipes, sometimes called positive pressure, use acetylene generated from a pressure generator at a pressure up to 9 lbs. per sq. in. An injector or partial injector forms part of the blowpipe. High-pressure blowpipes use the acetylene supplied from acetylene cylinders, and are designed to mix acetylene and oxygen in equal volumes. No injector is required, and a change of nozzle with a corresponding adjustment of pressures is all that is required when changing over from welding one thickness of material to another.

In the low-pressure system thorough mixing of the gases in the blowpipe is very necessary. To obtain this, well-designed blowpipes carefully adjusted to the oxygen pressure recommended by the maker, are essential. In the multi-flame blowpipe two flames emerge from the nozzle and serve to preheat the material, welding wire and form the molten pool of metal.

The size or power of blowpipe which should be used depends upon the thickness to be welded. By power is meant the amount of acetylene consumed per hour. Too small a power produces the defects of adhesion, inclusion of oxide, overheating, and burning of the welds. Too large a power produces a molten bath which is much too large, holes, and adhesion in the lower part of the weld.

The figure denoting the power of blowpipes is supplied by the maker, and is sometimes marked on the blowpipe. In welding materials other than wrought iron and steel the power of blowpipe to be used is dependent upon the material, its dimensions, and whether the job has been preheated; as a general rule the power of blowpipe for a given thickness of material is greater, because in welding, for example, cast iron, copper, or aluminium, more heat is required to melt a given weight of the metal, and, in addition, the high-temperature part of the flame is held farther from the molten metal.

All blowpipes should be carefully handled and treated as precision tools. The procedure to be followed in lighting and extinguishing the blowpipes varies with the design and type of blowpipe. The manufacturer's instructions should be closely followed. Lighting devices are convenient and safer than matches. A blowpipe should never be put down, unless a blowpipe stand is provided, until the gases are turned off, and should never be hung from regulators or other equipment. Gas economisers which serve the purpose of a stand and automatically cuts off the gases, except sufficient acetylene for a pilot jet, are valuable, especially in repetition work. Blowpipes in which the acetylene valve is entirely independent of a single valve which controls both gases, thus obviating the adjustment of the flame every time the blowpipe is lit, can be obtained. Carefully avoid the use of grease or oil, and never dismantle a blowpipe with a view to overhauling, as this may lead to irregular working after assembling. In the event of obstruction of the nozzle, a soft brass wire should be used for cleaning. Dust in the interior of the blowpipe can be removed by blowing oxygen through from the nozzle end. Equipment should be inspected at frequent intervals by a competent workman, and only standard parts should be used in repairs to apparatus.

In the case of back-firing or flash-back due to overheating, the tip should be cooled in water, taking care to close the acetylene valve and to leave the oxygen valve slightly open. By back-fire is meant the momentary return of the flame into the blowpipe tip, which may relight immediately upon withdrawing the blowpipe away from the work or necessitate the reignition of the gases by means of a lighter. By flash-back is meant a sustained back-fire, the flame burning in the mixing chamber with a squealing sound accompanied by a smoky, sharp-pointed flame, necessitating the immediate cutting off of the gas supply, to prevent severe heating and possible destruction of the blowpipe head, and the subsequent cleaning to remove carbon deposits. If internal ignition occurs frequently, the blowpipe should be sent for repair. An iron or steel wire should never be used for cleaning the nozzle.

Back-firing and flash-backs, which are a common cause of defective and expensive welds, may be due to one of the following causes:—

(*a*) Faulty design and manipulation of the blowpipe.
(*b*) Enlarging the nozzle orifice when removing obstructions
(*c*) Working in confined spaces or with an accumulation of slag on the nozzle.
(*d*) Overhauling and internal cleaning.
(*e*) Restriction and fluctuation in the acetylene supply due to bad lay-out, exhausted purifier, etc.
(*f*) Defective hydraulic safety valves leading to air and moisture being carried with the gas.
(*g*) Defective oxygen regulators.
(*h*) Presence of solid matter in the gas either from the acetylene generator or inferior flexible tubing.

Oxy-Acetylene Flame Characteristics.

The combustion of acetylene in oxygen produces a flame which apparently consists of two parts —an inner cone, incandescent, clear, and well defined; and a non-luminous outer envelope which may be transparent or more or less opaque according to the purity of the acetylene. In a well-regulated and correctly designed blowpipe there is, between the inner white cone and outer envelope, a reducing zone which is distinctly visible when the acetylene is impure. The exact character of the flame will depend on the proportion of the oxygen and acetylene as they emerge from the nozzle. The oxygen and acetylene should mix in approximately equal proportions. Gas-ratio, or the ratio of the oxygen to the acetylene consumption, is an important factor in determining the quality and cost of welds. Varying proportions of the gases produce three characteristic types of flames, commonly known as carbonising, neutral or normal, and oxidising. These are shown in fig. 4. It is important that in the majority of welding operations the flame should be neutral—that is, there should be no excess of oxygen or acetylene. Defective welds may be produced with carbonising and oxidising flames. It is particularly important that there should be no excess

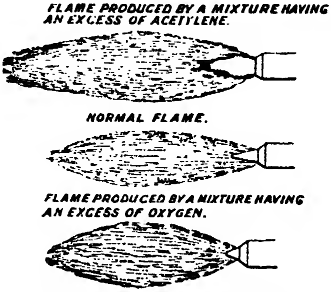

FIG. 4.—Flame Characteristics.

of oxygen, which not only tends to produce weak welds, but also to blow the metal forward and produce adhesion. Regulation of the flame during important welding operations is essential, as very few blowpipes maintain a neutral flame under welding conditions, and further, the operator has a tendency to adjust the flame to oxidising. It is advisable to adjust the flame whilst wearing dark glass goggles.

It is necessary to emphasise that the white cone of the flame should never be brought into contact with the molten metal, and that its distance from the molten metal depends, in general, upon the kind of metal being welded and its thickness.

GOGGLES AND PROTECTIVE EQUIPMENT.

For welding with acetylene it is necessary to protect the eyes with goggles fitted with suitable coloured glasses. There are a great many different kinds of special safety glasses on the market and many combinations of ordinary coloured glass are in common use. The eyes must be protected from injurious rays, the heat and glare from the work, and from particles of hot metal that fly up from the weld. To obtain the clearest definition or visibility with the least amount of glare, the selection of the colour tint in safety glasses should properly be decided by an expert; the depth of the tint may be determined best by the operator himself, owing to the individual difference which will permit one man to see clearly through a glass that would be too dark for another man. Amber, dark-amber, or dark-amber-green glasses give efficient protection. For the frames a tough heat-resisting material like vulcanised fibre is advantageous. Goggles should be light, comfortable, and well ventilated, and should be obtained from a reliable source.*

Welders employed on heavy work should be supplied with fireproof gauntlets and aprons, as a protection against radiated heat, and where welding is being performed in the interior of some structure or confined space all the outer clothing should be fireproof.

Filling Materials.

The necessity for employing the best materials for every welding operation cannot be too strongly emphasised. It is only by using the most suitable filling-in wires, rods, and fluxes, specially manufactured and controlled for each particular metal or alloy, that the most satisfactory work can be done.

Thin material is frequently welded by flanging, bringing the edges into contact and melting them together, but generally a filling material or 'welding rod' is melted into the groove or crack.

Filling materials must, where possible, not only be of the correct chemical requirements to (*a*) replace loss in the molten zone, (*b*) counteract the tendency of the metal to oxidise, (*c*) assist in making the molten metal flow readily, and (*d*) to break down oxides when formed, but they should also act satisfactorily under the blowpipe. As a general rule, the welding wire or rod should have some similarity to the composition of the material to be welded.

* Protective Glass for Welding, B.S.I. No. 679.

The choice of the right size of welding wire or rod is important, and melting of both should take place in the molten pool and not in the flame. The size of welding wire is based on the thickness of the sheet or plate and, in the case of iron and steel, depends upon whether the leftward or rightward method of welding is used. For leftward welding the diameter of the wire should be one-half the thickness of the material welded plus one-sixteenth of an inch. In the case of rightward welding the diameter should be one-half the thickness of the sheet or plate.

Fluxes.

It is not always possible to include deoxidisers and special elements in the welding rod itself. By using a combination of chemicals in the form of a powder, liquid, or paste, known as a flux. the difficulties of producing sound and reliable welds are frequently overcome. A flux is used for (*a*) cleaning the parts to be welded; (*b*) eliminating and preventing the formation of oxide; (*c*) replacing elements burnt out by the flame; (*d*) facilitating the flow of the metal, and, when melted, (*e*) protecting the surface of the molten metal from the action of the flame and preventing blow-holes. The reducing zone of a well-regulated and well-manipulated flame protects, in a measure, the molten metal from oxidation. A flux is necessary in welding steel (except in cases where the steel welding wire contains deoxidising materials), cast iron, copper, aluminium. etc.; for successful welding the proper flux should be used, and as the welding rod and flux are important factors in welding they should only be obtained from reliable manufacturers.

Preparation of the Work.

One of the first principles of sound welding is securing complete fusion and perfect union of the filling material and the edges of the parts to be welded. To ensure this, good preparation is necessary, which includes, as a rule, the cleaning of the edges and the welding wire, the bevelling or chamfering of the edges, the adjustment and maintenance of the parts in position, and allowance for expansion and contraction effects. In general, the preparation of the parts varies according to the thickness and nature of the metal and with the organisation and equipment of the workshop.

Cleaning the parts to be welded and the welding wire or rod is essential. Bevelling is generally recommended for materials ⅛ in. and above in thickness. Bevelling ensures good penetration, increase in ductility and strength, economy, and reduction in the overheating effects. The angle of the bevel depends upon the thickness of material and whether 'leftward' or 'rightward' welding is used. All metals from ⅛ in. up to ⅜ in., and in some cases ½ in., should be bevelled from one side only. Pieces exceeding ½ in. may be bevelled from both sides, but this is not recommended. In leftward welding the 'V' formed by the two bevelled plates should be about 90° for material exceeding ¼ in. in thickness, and 50° to 60° for material ⅛ to ¼ in. In rightward welding for material up to ¼ in. an angle of about 45°, and above ¼ in. an angle of 60°, can be used. The bevel should extend to the bottom of the plate, and thorough penetration and complete fusion should be obtained. In repair work preparation of the work is a very important feature and will save considerable time and keep the cost of welding low. Should it be necessary to re-weld a joint previously welded, it is essential that the weld metal be entirely removed, and on old work the same instruction applies, and corroded metal must be removed. In the case of aluminium repairs, for example, good judgment, as shown by the preparation, means success, and bad judgment a ruined casting.

Expansion and Contraction.

It is impossible to produce satisfactory work unless expansion and contraction effects are properly allowed for. Warpage and residual stresses, followed by failure, are the common result of welders failing to master this subject. The methods commonly employed are, in the case of thin sheets, tacking, or manipulating the sheets so as to take advantage of the effects of expansion and contraction; in the case of heavier material and castings, setting the plates so as to form an angle, heating the entire work, partial heating, heating restraining members, breaking a member so that expansion and contraction can take place freely, bending or springing the material, etc.

If expansion can take place in all directions, it will give the welder no trouble, but he should operate so as to leave practically no residual stresses. If, however, welding is done at a place that is confined by various parts or by the particular construction of a piece, it is necessary to give it due consideration.

The effects of contraction must be considered equally with those of expansion, as failures, cracks, etc., can often be traced to overlooking this property.

Preheating.

This may be partial or total, depending upon the work in hand, and is mainly employed with a view to (*a*) allowing for expansion and contraction, (*b*) decreasing the cost of welding, and (*c*) producing a more homogeneous joint. Preheating is usually carried out with the welding blowpipe, oil or gas burners, charcoal, or other suitable fuel. Where fuels such as charcoal are used for preheating, the combustion gases should, where possible, be carried away. In any case ventilation should be provided to avoid the harmful effects of the gases. The method used should be governed by the particular work in hand.

No general rules can be given for preheating temperatures, as the degree of heat necessary will, of course, depend upon the size and shape of the object to be welded. As a general rule however, cast iron should be preheated to a dull red heat for small and simple castings, and to a blood-red heat for large or complex castings; aluminium and its alloys should be preheated very carefully, as too high a temperature will lead to the collapse or distortion of the article being welded.

Preheating machines burning paraffin and giving a flame of high calorific value, smokeless and transparent, give satisfactory results where portability and a powerful flame are required.

After-Treatment of Welds.

Improvement in the structure of welds with a consequent increase in strength and ductility is obtained by proper after-treatment of welds. Thermal or mechanical treatment, or both, carried out in a correct manner is invariably beneficial, except in the case of defective welds.

Strength of Welds.

With proper equipment and suitable filling materials, the strength of a weld will depend mainly upon the care and skill of the operator. A good operator should be able to obtain welds 95 per cent. the strength of the parent metal on material of good weldability; for the average welder 85 per cent. should be taken as the lower limit of strength.

Recognition of Materials.

The identification of the metal, especially in repair work, is invariably indispensable. Once the metal or alloy is known the next step is to get familiar with all the facts relating to its weldability and, in case of doubt, proceed to the necessary tests for determining these. Some of the means by which the materials of the parts to be welded may be recognised are colour, appearance of surface, contour, grinding spark test, chisel cut, fracture, blowpipe test.

Steel Welding.

The welding of wrought iron and mild steel is apparently simple; in reality it is somewhat difficult to produce strong and reliable welds. Oxidation and inclusions are the most serious defects, and must be reduced to a minimum. Welding wires containing a small percentage of vanadium or nickel are used with highly satisfactory results. To produce the minimum oxidation the blowpipe must be well designed and handled efficiently, and the parts to be welded must be thoroughly cleaned. Expansion and contraction must receive proper consideration. The use of a flux in the form of a powder has not proved satisfactory. It has been found that deoxidising materials in the form of a liquid gives excellent results. The product known as 'Suder' is specially prepared for use with a welding wire which does not contain deoxidising elements. At the high temperature of the flame it slags the oxide, keeps the surfaces clean, and prevents the inclusion of oxide. The weld can be carried out rapidly, thus reducing overheating effects. The liquid is applied by means of a brush to the edges of the plates and the welding wire previous to welding. The welding wire should be a material low in phosphorus and sulphur, and free from oxide. The use of ordinary iron or mild steel is unsatisfactory, and welding wire, such as 'Suder' or 'Ferrox,' made especially for the oxy-acetylene process, should be insisted upon. It will be found that economy is on the side of the controlled, but slightly more costly, and dependable product.

A simple test to indicate the fitness of welding wire is made by laying the wire flat on a clean surface and applying a good neutral oxy-acetylene flame to the wire in such a manner as to heat it for a distance of about 3 or 4 ins., playing the flame backwards and forwards until the wire is red and then slowly melting the wire, moving the flame so that the wire only melts halfway through its diameter. If the flame is withdrawn as soon as the metal begins to melt, the impurities can readily be seen being thrown off in the form of sparks, or a boiling action in the case of inferior metal; in some cases the metal will not flow together but 'runs away.' The metal from gas absorbing wires will, when cold, contain numerous spongy, volcano-like irregularities. In the case of clean metal free from impurities the wire will melt and flow evenly without any disturbing action.

The following table is useful when ordering steel welding wire:

Size. Inches	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$
Lbs. per 100 ft. .	1·04	2·3	4·1	6·5	9·3	16·7	26	37·5
Ft. per 100 lbs. .	9,615	4,273	2,403	1,537	1,067	600	383	266

The flame must be carefully regulated and held so that the tip of the white cone is about $\frac{1}{8}$ in. from the surface of the molten metal.

A good practical rule in welding steel is that the hourly consumption of acetylene in cub. ft. is obtained by multiplying the thickness of the sheet or plate in inches by 90. This general law, based on investigation, holds good for both the rightward and leftward methods of welding.

When using a blowpipe of the power fixed by the above law the speed of welding in ft. per hr. can be obtained by dividing a constant by the thickness of the metal in inches. For leftward welding the value of this constant is 1·6 and for rightward welding 2.

The blowpipe must be given a motion which will cause the flame to describe a series of zigzag or side-to-side movements in the case of light and medium sheets. This is known as 'leftward' welding. A variation of these movements has been introduced which gives more economical and better welds. The welding wire, in this case, follows the flame instead of preceding it, and the flame is directed towards the starting point instead of towards the finishing point. The blowpipe is given a slight gyratory motion, whilst the welding rod is given a more or less elliptical or longitudinal motion. This method of welding is known as 'rightward' welding. Subsequent thermal treatment, which may consist of annealing or normalising, according to the particular requirements, will improve the structure of steel welds, and materially increase their strength and toughness. Hammering may be resorted to with good effect, provided the hammering is not too vigorous or done at too low a temperature. The technique for making welds in a vertical plane has been established and these methods are rapidly developing.

The following table gives the approximate speed at which the welding of mild steel plates can be done under average conditions :—

Thickness of Plate in Inches.	Speed of Welding.	
	Feet per Hour.	Feet per Hour for Average Day's Work.
$\frac{3}{64}$	30 to 33	23 to 26
$\frac{5}{64}$	20 „ 21	16 „ 20
$\frac{1}{8}$	15 „ 16	13
$\frac{5}{32}$	13 „ 15	10
$\frac{13}{64}$	10 „ 12	9
$\frac{1}{4}$	9 „ 10	7½
$\frac{9}{32}$	8 „ 9	6½
$\frac{21}{64}$	7 „ 7½	6
$\frac{23}{64}$	5½ „ 6	5
$\frac{13}{32}$	4¼ „ 4½	3½
$\frac{15}{32}$	3¼ „ 3½	2½ to 3
$\frac{17}{32}$	2¼ „ 2½	1½ „ 2

Owing to metallurgica lchanges that take place when melting medium and high carbon stee with the blowpipe, the welds obtained are invariably unsatisfactory. As a general rule, the lower the carbon content of the steel, the more successful the welding operation. It is possible to get fair results on medium steels, but a flux, rapid welding, and thermal after-treatment, which requires skill to properly carry out, are necessary.

For welding medium and high carbon steel and worn surfaces where great resistance to abrasive wear is desired and where machining is not necessary, a special welding wire such as chrome-vanadium is desirable.

Alloy steels, such as chrome and manganese, are weldable, provided the percentage of carbon does not bring them into the range of medium and hard steels, which is frequently the case. The welding wire should be of the same character as the steel, and a flux must be used. In the case of nickel steel it is necessary to use a flux which will form a varnish over the molten metal and prevent the absorption of the gases from the flame of the blowpipe. It is important to recognise that alloy steels are used largely because of the excellent physical properties that are developed by suitable heat treatment, and that the uneven heat treatment produced by welding temperatures may produce considerable variations in quality. Hence they should be welded with a full knowledge of the alloy and the results likely to be obtained. This also applies to the welding of stainless steel, which has been thoroughly investigated. The principal difficulty is the formation of the oxide of chromium which is dealt with by the use of a first-class flux. The method of applying the flux is an essential factor for success, and consists in coating the welding wire or rod and the underside edges of the sheets to be joined. To obtain an adhesive coating the wire and edges are first painted with a solution of silicate of soda and then sprinkled with the flux. First class results can be obtained on decay proof non-corrodible steels.

Cast-Iron Welding.

The two principal defects in cast-iron welding are the production of hard spots and blow-holes and subsequent cracks or fractures in the weld or adjacent metal. The oxide formed must be removed or destroyed by the use of a flux. The welding rod must be of good material and specially made for cast-iron welding. One great trouble with cast-iron rods in general is that the quality is by no means uniform. This is especially true with the cheaper and uncontrolled products. Rods made from ordinary cast iron produce blowholes and hard spots. Special rods containing titanium, producing sound, soft, and reliable welds, are obtainable. Expansion and contraction should be considered as of great importance. Preheating should be used to some extent in all cast-iron welding. It is advisable to weld in position after preheating. The welder is protected by making use of asbestos sheet or simply plate. Specially designed blowpipes of considerable length are available for this class of work, and two or three welding rods are welded together. The parts must be thoroughly cleaned, and in the case of cracks their extension must be prevented by drilling holes a short distance from the ends in the direction the crack would follow, The weld should be arranged horizontally, and where this cannot be arranged carbon blocks should be used to prevent the metal flowing away. Carbon products, in the form of blocks, powder, or paste, play an important part in modern welding operations. They are far superior to fireclay or fire-bricks, which are liable to melt under the blowpipe and invariably cause blowholes. Carbon compound, in powder form, which can be made into a plastic mass with water, is valuable for making moulds, dams, etc., and for lining up repairs and holding parts in position. To protect machined surfaces during preheating operations, they should be coated with flaked graphite and oil. This can be made into a paste and painted on, or the surfaces can be oiled and the graphite dusted on. The graphite must be coarse, as fine flake is not satisfactory. The flame must be held so that the tip of the white cone is between ⅛ and ⅜ in. from the molten metal, the distance depending on the thickness of the metal. The after-treatment consists in annealing or allowing the metal to cool very slowly. In well-executed welds on cast iron the quality of the weld is generally superior to that of the metal of which the casting is made. The grain is finer and the welded zone is sound, homogeneous, and can be easily worked.

Certain cast irons of great strength and in which the carbon is in the combined state are difficult to weld. Cast iron which has been submitted to the heat of a fire for a considerable time is frequently unweldable. The metal in both cases melts under the blowpipe in a distinctly different manner from ordinary cast iron, and the welder should have no difficulty in recognising this difference and consider whether it is advisable to attempt the repair. The 'bronze welding' of cast iron, iron, steel, etc., is a process in which the oxy-acetylene flame is used in conjunction with a copper-zinc alloy, such as 'Suder Bronze,' 'Sifbronze,' or manganese-bronze, to produce joints which would be difficult or expensive to obtain by the usual method of melting the metal of the casting. The procedure in making joints is very much the same as in actual welding. The 'V' opening need only be about 15°. The heat must be just right—too much will burn the bronze, too little will prevent any alloying—and the surfaces must be clean. It has been found in actual operations that the welding time is greatly reduced when the parts are prepared by sand-blasting.

MALLEABLE CAST IRON.

It is difficult to produce satisfactory joints on malleable cast iron by welding. The welds are usually hard, brittle, and porous. The best results are obtained by bronze welding with the oxy-acetylene blowpipe, using the same methods of preparation as for cast iron, and using 'Suder' bronze or 'Sifbronze' as a filling material. For invisible welds Monel metal welding wire can be used. The two pieces to be joined are brought to a point just below fusion, great care being taken that they do not become fused. When the edges are at the right temperature, the rod is fused into the groove with the aid of a good flux.

The use of acetylene for brazing purposes provides a fuel which has the highest known flame temperature and high calorific value. By using dissolved acetylene with a special brazing blowpipe no air-bellows or oxygen is necessary, as the blowpipe is atmospheric and a clean and relatively small flame of suitable power is obtained.

Aluminium Welding.

Sheet aluminium welding can be handled in very much the same way as steel as regards preparation and allowance for expansion and contraction. The principal difficulty lies in the formation of the oxide, which must be dealt with by an energetic and reliable flux. The filling material is pure or alloy aluminium wire; before starting to weld, the wire and parts to be welded must be thoroughly cleaned by scraping. A delicate touch and considerable practice are necessary, especially on thin material. Material under 22 gauge is best welded after flanging the edges. The flanged edges should practically have no radius, so as to leave the minimum of space between the underside of the sheets. The depth of the flange should be about two and a half times the thickness of the sheet. The white cone of the flame should be level with the surface of the plates and the blowpipe held so that the flame is inclined to the horizontal at about thirty-five degrees. If the methods are correct the upper and lower surfaces of the weld appear identical. The power of the blowpipe to be used is not in direct ratio to the thickness of the metal as in steel welding. For example, on 16 to 19 gauge material a blowpipe consuming 2 cu. ft. of acetylene per hour would be satisfactory, whilst for material ⅛ in. thick 10 cu. ft. is required, and for ¼-in. material 40 cu. ft. per hour. The flame must be carefully regulated, and the tip of the white cone should be from ⅛ to ⅜ in. from the molten metal. After welding, the welds should be thoroughly washed with warm water and, where possible, internal strains should be removed by annealing. The structure and strength of the metal in the welded zone are improved by hammering. Subsequent corrosion of the welded seams may be due to the physical condition of the aluminium sheet and the welds, and in the case of important work preliminary corrosion tests should be made on welded strips cut from the sheets.

Pure aluminium is not usually employed for castings subject to stress. In order to raise the tensile strength and increase the hardness varying percentages of zinc, copper, or nickel are added. There is an endless variety of aluminium alloys, but from a welding point of view they fall into two classes, viz., those containing zinc as their principal hardening medium, and those containing copper. The material expands greatly and is very weak at high temperatures, consequently contraction strains are likely to produce cracks and distortion unless the work is properly handled. In welding castings—crank cases, for example—it is often necessary to preheat them either completely or partially. The preheating must be done carefully, otherwise the casting may collapse. For supporting the casting evenly and lining up, carbon putty is useful. For missing parts, and where the welder is not well skilled in this class of work, it is advisable to back up with sheet copper or galvanised iron. In rigging up jobs for repairs, care must be taken that expansion and contraction effects are not interfered with. The parts should be carefully cleaned and free from oil and grease. The repair of oil saturated castings is frequently one of considerable difficulty. Cleaning, by dipping in caustic soda, followed by washing in scalding hot water, and drying, bevelling to a larger angle than usual and sweating out the oil and scraping it away, indicate the methods which lead to successful repairs. A good grade of filling material of average composition is necessary; pure aluminium wire should not be used. The acetylene should be pure and the blowpipe should give a soft, non-oxidising flame. Two methods of welding are adopted, one known as the flux and the other as the puddle method. In the former a flux is used in the ordinary way and in the latter no flux is used, a steel rod and paddle being used to remove or break down the oxide and to smooth the weld respectively. All traces of flux must be washed off. The casting should be reheated slightly to remove any local strains and then allowed to cool very slowly. For finishing off, a single cut-file or a 'dreadnought' file is most suitable.

The welding of the well-known aluminium alloy known as 'Duralumin,' which contains copper, manganese, magnesium, etc., is satisfactory or not according to the mechanical properties required in the joint. The alloy has important properties which are reduced, and in some cases almost eliminated, by welding operations. The metal should not be welded when the joints are subject to bending action, shocks, or when the full properties of the metal are desired.

The welding of aluminium-silicon alloys in the form of castings presents no particular difficulty. In sheet form, however, the difficulties, in general, are greater than with pure aluminium sheet, and the correct welding procedure must be followed. Bevelling for material slightly exceeding $\frac{1}{16}$ in. is advisable to obtain good penetration. A welding wire of similar composition and a good flux are essential.

Copper Welding.

Copper can be successfully welded when material of good weldability is used and the operator is familiar with the technique. Deoxidised copper in various forms and suitable for welding purposes can be readily obtained. Material which may be satisfactory for brazing may prove quite unsuitable for welding. Where possible, a sample strip 2 or 3 ins. long should be tested by making a weld and applying a bending test. It is necessary to clean and bevel the edges and to support the underside of welds made on thin material, copper tubes, etc., by means of a mandrel consisting of an iron tube covered with asbestos, which must be free from moisture. Owing to its lack of strength when hot, the tacking of copper welds should be done carefully. For the same reason welds should not be started at the edges of sheets, but at a point about one-third the length of the seam. The power of the flame should be practically the same as that used for welding steel of the same thickness and the flame must be carefully regulated. The melting should not be too rapid or too slow. Without a correct welding rod it is impossible to satisfactorily weld copper. A flux is useful. Preheating to avoid adhesion is necessary. The tip of the white cone should be about $\frac{1}{4}$ in. from the surface of the molten metal. The welds should be hammered at a dull-red heat to improve their ductility and strength, and afterwards annealed. The technique for the repair of copper fireboxes and tube plates has been worked out and applied with great success. A good sense of fusion, the addition of the metal in thin layers, and satisfactory equipment are essential for success in this class of work. Welds on copper can be obtained with mechanical properties almost identical with those of the parent metal. Such welds can be planished, stamped, spun and, in general, submitted to the ordinary workshop operations carried out on sheet copper.

Brass and Bronze Welding.

Brasses and bronzes are only welded indifferently without the correct welding wire or rods. The filling material should be practically of the same composition as the metal to be welded, with small additions of phosphorus, silver, aluminium, nickel, etc. Fluxes, similar to those used for copper welding, should be used in both cases, and the power of the flame is less than that used for the same thickness of copper—that is, distinctly less than for steel plates of the same thickness. The regulation of the flame is very important: a slightly oxidising flame will give a surface free from wrinkles, and prevent, or greatly diminish, the volatilisation of the zinc. The position of the flame is the same as for copper. The after-treatment of the welds consists in hammering at definite temperatures. Identification of the particular brass or bronze to be welded is important in view of the importance of choosing the correct welding wire or rod and the subsequent treatment which may be necessary. Reclaiming castings and repairing worn or broken parts, such as gear wheels, bearings, etc., constitute the general run of the work on bronzes. In the case of castings similar methods to those outlined for the repair of cast iron are necessary.

Lead Welding.

The oxy-acetylene blowpipe is rapidly supplanting the older methods of welding lead, or 'lead burning,' as it is generally called. It has proved superior in convenience, quality of joints, and economy. A small type of blowpipe is required, consuming from ½ to 6 cu. ft. per hour. When it is necessary to fill up a hole or to build up any part of a lead object, the work is facilitated by using a mould or a backing of steel plate. When welding patches in repair work avoid rectangular patches on vertical faces, which make two edges of the patch horizontal. The patch should be made lozenge shape, so that the welds are done on the incline. The welding wire or rod should be of the same metal as the sheet. Special outfits have been designed for lead burning, using dissolved or generated acetylene; they are very compact and portable.

Nickel and Nickel Alloys Welding.

The production of sound welds on nickel and high nickel content materials depends upon the deposition of metal which is metallurgically sound, and upon the welder being familiar with the recognised technique. The metallurgical features of the metal and its alloys have a direct bearing on success. The use of the oxy-acetylene blowpipe with an additional reducing flame has proved satisfactory. With a single-flame blowpipe, a well designed blowpipe and a carefully regulated flame are essential. It is advisable to use flux sparingly. Precautions should be taken to prevent stress causing fractures as the metal cools from 850° to 750° C., in which range its ductility is very low. Alloys of nickel such as Monel metal, Cronite and Brightray can be successfully welded by following a technique closely similar to that for nickel. The welding of German silvers, or nickel silvers, as they are now generally known, or nickel brass, as designated by the Institute of Metals, is by no means simple and a preliminary test is advisable to determine the flame regulation, which will give satisfactory welds when used in conjunction with a good flux.

Magnesium and Magnesium Alloys Welding.

The behaviour of magnesium under the blowpipe is very similar to that of aluminium in that the film of oxide which forms on the molten metal prevents coalescence unless an active flux is used. Unless the flame is correctly regulated combustion of the metal may take place. The power of the blowpipe can be based on that used for welding corresponding thicknesses of aluminium. Magnesium alloys, such as electron, are weldable, but established technique should be followed to obtain reliable results.

Machines for Welding.

Acetylene welding machines for the manufacture of tubes have been in use for many years. The development of automatic welding machines for the manufacture of tanks, radiators used with central heating, etc., is more recent. For repetition work machines have been designed which accomplish the work in less time with comparatively unskilled men. The machines consist of two parts, (*a*) holding device for the work, and (*b*) apparatus for moving the blowpipes, with or without oscillatory motion, over the seam.

Recent developments in welding machines have been mainly in the direction of the use of multi-jet blowpipes. In order to obtain speed it was recognised that more than a single jet of flame should be employed, and from two jets in tandem it has become customary to employ a large number of jets extending for a considerable distance along the line of the weld. The defects of this method have been overcome by distributing the heat on both sides of the centre line of the seam. The welded section produced in this way is wider, stronger, and more certain to penetrate than is the case with the narrow weld made by a central line of flame. Using dissolved acetylene, the pressed halves of radiator elements are welded together, fifteen at a time, which means that one workman does equivalent work to eight welders using hand-operated blowpipes, with a consequent economy of 50 per cent. in gases.

Training Welders.

An oxy-acetylene welder has been defined as a person who has the requisite manipulative skill and technical knowledge successfully to operate welding and cutting apparatus, and, at the same time, has skill in the operation of hand tools used in the preparation and finishing of the work in connection with welding, and also has a reasonable amount of ingenuity. He must also have a knowledge of accuracy commensurate with the type of work to be welded. Naturally, a trade in which so much depends on the personal equation has drawn attention to the need of training men to produce sound work. No doubt the best method is that which expounds principles followed by practice under expert supervision. Welding skill grows rapidly under a well-worked-out system of exercises which aim at developing skill in making typical welds on various metals. Testing the welds is also an important requisite of a training course. Classes

for training welders are held in the principal centres, a standardised course for training welders for general work or specialised work, has been laid down, and manufacturing concerns who train their workmen usually follow methods depending upon the size of shop and class of work.

A common error is to attempt work, especially repair work, in a shop where the tools and apparatus for successful all-round welding are not provided, or to endeavour to repair jobs which are beyond the capacity of the welder. It is much more economical, in such cases, to hand over work of this character to firms who have a reputation for repairs of all kinds, especially large castings which frequently must be welded *in situ* and require specialised knowledge and a choice of the most suitable welding process. A firm specialising in general repairs have the necessary equipment for carrying out and testing the work, and welders who are not mere blowpipe operators.

Testing Welds.

The importance of testing welds has, in recent years, become recognised. It is well known that the quality of a weld cannot be assumed from the general appearance, finish, etc., although considerable information can be obtained from the visual inspection of welds. At the present time the usual methods of testing include tensile and impact tests on specimens welded under identical conditions, microscopic examinations of specimens, bending and hammering tests and X-ray examination. In the latter the actual welds can be examined, and flaws, blowholes, and other defects discovered. Electrical and magnetic test for the non-destructive examination of welds are in use. In fact, simple and portable apparatus based on the magnetic properties of iron and steel is now in use for the non-destructive examination of welds on iron and steel of any thickness.

The American Bureau of Welding have drawn up Standards for Testing Welds which are intended primarily to serve as a standard uniform basis for testing and comparing sample welds as distinguished from welds on structures. The testing of the latter must necessarily be by non-destructive methods, which would therefore be, in general, quite different from those employed in examining sample welds made for the purpose of testing a welder's proficiency, comparing welding wires, or comparing welding methods. For the shop standard a bending test is recommended; for the commercial standard, a bending and tensile test on approved lines.*

Miscellaneous Applications.

There are numerous applications of the oxy-acetylene process not included in the term welding, but which have a direct or indirect connection with it and are worth consideration. The oxy-acetylene flame, for example, is a convenient and simple method of applying local heat for numerous workshop operations, such as forging, straightening, riveting, expanding for shrinkage fits, heat-treatments, etc.

Brazing operations are more efficiently carried out with the oxy-acetylene flame than by the older methods. Case-hardening and surface hardening can also be efficiently and economically done with the flame.

OXY-ACETYLENE CUTTING AND FLAME MACHINING.

Wrought iron, steel, and steel castings can be cut with the ordinary cutting blowpipe, and with a special cutting blowpipe or the ordinary cutting blowpipe operated in a special manner, cast iron can be cut.

The cutting blowpipe usually consists of a compound jet made up of one or more heating flames with a pure oxygen jet in the centre. The purity of the oxygen is, in general, of much greater importance when used for cutting than when used for welding. As the purity decreases the oxygen consumption and time increases, and the speed of cutting decreases as the following figures, which represent tests on steel plates 5 feet long by 1 inch thick, show :—

No. of Test.	Purity of Gas. Per cent.	Time in Seconds.	Oxygen Consumption in Cu. Ft.	Remarks on Cuts.
1	99·7	280	8·0	Very smooth and clean.
2	99·4	291	8·8	Smooth.
3	99·1	315	9·8	Satisfactory.
4	98·0	334	11·3	Good but rough
5	96·0	365	13·2	Not smooth.
6	95·0	412	15·4	Very irregular.
7	90·0	644	20·0	Rough, surface bad.
8	87·0	675	23·0	Very rough, surface bad.
9	85·0	696	25·8	Very rough, unsatisfactory.
10	82·5	—	28·4	Very rough & not through.

* B.S.I. Specification No. 709. Bend Test for Welds.

A spot on the metal is first heated to a bright red heat with the heating flame, and then the oxygen jet is turned on. It has been proved that only partial combustion takes place, the phenomenon of cutting being really a disintegration by the current of gases from the blowpipes. The oxy-hydrogen or oxy-coal-gas flame is often used for cutting in place of acetylene. Material up to 36 ins. has been cut. The same acetylene equipment is used for cutting as for welding. For heavy cuts special oxygen regulators are required. There are two kinds of cutting blowpipes, known as the 'central' and 'following jet' types. Hand-cutting blowpipes for cutting steel up to 20 ins. in thickness and cast iron up to 17 ins. thick are available. Where dissolved, acetylene is used a vacuum type of regulator with many advantages can be used. Special equipment, known as oxygen lance equipment, can be obtained for opening up the frozen tap holes of blast furnaces, boring holes in iron and steel blocks, steel or iron, castings, etc. Oxygen regulator heaters, electrical or gas, are frequently used when large quantities of oxygen are being used, to prevent interference with the flow of gas as a result of freezing. Mechanical contrivances have been devised for special forms and shapes and for heavy cutting. Automatic cutting machines using the oxy-acetylene or oxy-coal-gas blowpipe, with or without preheated cutting oxygen, have reached a high degree of perfection. These machines usually have the full range of movements found on a universal machine tool, and have a degree of accuracy such that accurate gear teeth can be cut, and any desired number of thousandths can be left for finishing.

Data for Oxygen Cutting by Hand.

(British Oxygen Co., Ltd.)

Plate Thickness.	Nozzle Size.	Oxygen Pressure at Regulator.	Cutting Speed. Ft./hr.	Gas Consumption. Cub. ft. per hr.	
				Oxygen.	Acetylene.
$\frac{1}{4}$	$\frac{3}{64}$	25/30	180/225	75– 85	30–35
$\frac{1}{2}$	$\frac{3}{64}$	30/35	100/125	85– 95	35–40
$\frac{3}{4}$	$\frac{3}{64}$	35/40	70/100	85–105	40–45
1	$\frac{1}{16}$	35/50	65/90	160–210	40–50
$1\frac{1}{2}$	$\frac{1}{16}$	40/60	60/80	180–250	45–60
2	$\frac{1}{16}$	45/65	40/60	190–270	48–66
3	$\frac{1}{16}$	50/75	25/45	220–300	50–72
4	$\frac{5}{64}$	55/75	18/25	330–430	55–72
5	$\frac{5}{64}$	65/85	16/22	375–485	66–80
6	$\frac{5}{64}$	70/90	12/18	400–500	70–85

The above figures are based on actual cumulative results and the chief variation to be expected is the cutting speed, which will vary according to the nature of the work, experience of the operator, etc.

Heavy Steel Cutting

(British Oxygen Co., Ltd.)

Material Thickness.	Nozzle Size.	Distance of Nozzle from Material.	Oxygen Pressure.	Approximate Speed of Cutting.
Ins.	Ins.	Ins.	Lbs./sq. in.	Ft./hr.
7	$\frac{3}{32}$	[illegible]	70	13
8	$\frac{3}{32}$	[illegible]	75	12
9	$\frac{3}{32}$ or $\frac{7}{64}$	[illegible]	85	11
10	$\frac{7}{64}$	[illegible]	95	10
12	$\frac{7}{64}$ or $\frac{1}{8}$	$\frac{7}{16}$	120	8
14	$\frac{1}{8}$	$\frac{7}{16}$	150	6
16	$\frac{1}{8}$	$\frac{7}{16}$	180	5

Flame machining or cutting is developing under various titles, such as oxygen grooving, scarfing, etc. It utilises the oxy-cutting principle and differs only in the character of the cuts which it produces. Its main distinction is in the use of a tangential positioning of the cutting oxygen stream. Many operations of the usual tool-machining are similar to those obtained with the flame-machining blowpipe, such as planing, milling, turning, drilling boring, etc. The results up to date have shown the inherent possibilities in this application of the oxy-acetylene flame.

Blowpipes can be obtained for cutting steel under water. Under-water cutting is based on the fact that the water must be forced away from the heating flame and cutting jet.

Compressed air has been used for this purpose, and one type of under-water cutting blowpipe makes use of the products of combustion. The cooling action of the water necessitates using a heating flame five to seven times as powerful as that used for cutting the same thickness of material in the ordinary way. With an efficient arrangement an ordinary hand-cutter can be adapted for cutting under water. In salvage work steel plates have been cut at a depth of 40 ft. Various devices for lighting the flame under water when it becomes extinguished have been successfully introduced, being in the form of a pilot light burning continuously and forming part of the blowpipe.

Cast iron can be cut by means of the oxygen lance or by a special oxy-acetylene blowpipe. In the former case an iron tube from 6 to 12 feet long, $\frac{3}{8}$ in. external and $\frac{3}{16}$ in. internal diameter, containing three wires of 12 gauge, is supplied with oxygen at a pressure of from 70 to 150 pounds per square inch, according to the thickness to be cut. The point at which the cut is to be started and the end of the tube are raised to a red heat by means of an oxy-acetylene blowpipe, the oxygen turned on and the heating blowpipe removed. Combustion of the cast iron and the tube proceeds rapidly; it takes 20 seconds to penetrate a piece of cast iron 4 inches thick and less than 2 minutes for material 24 inches thick. In the second method a cutting blowpipe with a flame having a distinct excess of acetylene is used, or a blowpipe in which the cutting jet is modified and not the heating flame. In the latter type a certain quantity of acetylene is introduced into the cutting jet and leads to an increase in the temperature of the oxygen.

The following is an abstract from a manufacturers' table of cutting data:—

APPROXIMATE RESULTS FOR CAST IRON CUTTING.

(British Oxygen Co., Ltd.)

Metal Thickness.	Nozzle.		Oxygen Pressure.	Length of White Cone.	Approx. Speed of Cutting.	Gas Consumption. Cu. ft./ft. run.	
	Size.	Distance from Work.				Oxygen.	Acetylene.
Ins.	Ins.	Ins.	Lbs./sq. in.	Ins.	Ft./hr.		
Up to $1\frac{1}{2}$	$\frac{3}{64}$	[illegible]	110	2	12	30	10
$1\frac{1}{2}$—3	$\frac{1}{16}$	[illegible]	120	3	9	50	15
3—5	$\frac{3}{32}$	[illegible]	130	$3\frac{1}{2}$	7	70	20
5—8	$\frac{3}{32}$	[illegible]	160	4	5	100	25
8—11	$\frac{7}{64}$	[illegible]	150	4	3	250	50
11—14	$\frac{1}{8}$	[illegible]	150	4	2	350	90
14—16	$\frac{1}{8}$	[illegible]	160	4	$1\frac{1}{2}$	500	120

The above pressures, speeds, and consumptions are merely given as a guide to indicate the results obtained on cast-iron cutting and they will vary according to the composition of the iron, etc.

General.

A lack of familiarity with the various welding processes by engineers and bodies controlling regulations is still a contributing factor to the retardation of progress. The primary problems, such as the training of welders, tests for welds, welding wire specifications, technique of welding, investigations on rail joints, pressure vessels, and structural steel, have received attention. There still remains a large number of fundamental research problems in welding which can be classified under the following heads: (*a*) investigations involving physical tests; (*b*) involving considerable metallurgical experience; (*c*) problems in physics; (*d*) problems involving welding procedure; (*e*) problems involving chemistry; (*f*) structural problems; and (*g*) psychological problems. No less than fifty separate problems, arranged in these groups, have been listed and the opportunity for real service to the welding industry by their solution is obvious. The promotion and co-ordination of research in the welding industry is essential in view of its vast expansion.

ELECTRIC WELDING.

Electric welding processes can be placed in two distinct groups, one producing heat by the resistance of the metal being welded to the passage of the current, the other using the heat of the electric arc. They are known as Resistance Welding and Arc Welding, respectively. These two main groups have formed the basis for many processes of welding. The differences rest entirely on methods and special applications. The subdivisions are as follows :—

Arc Welding.

(1) Carbon or graphite electrode.
(2) Metallic electrode.
 (a) Bare electrode.
 (b) Coated electrode.
 (c) Covered electrode.
(3) Electric blowpipe.
(4) Atomic hydrogen.
(5) Shrouded arc.
(6) Gas shielded.

Resistance Welding.

(1) Butt welding.
 (a) Flash welding.
 (b) Upset welding.
 (c) Automatic butt or contact flash.
(2) Seam welding.
 (a) Butt seam welding.
 (b) Lap seam welding.
 (c) Bridge seam welding.
 (d) Flash-seam welding.
(3) Percussive welding.
 (a) Electro-static.
 (b) Electro-magnetic.
(4) Spot welding.
 (a) Lap spot welding.
 (b) Bridge welding.
 (c) Button welding.
 (d) Disc depression welding.
 (e) Projection welding.
 (f) Multiple projection welding.
 (g) Multiple electrode welding.
 (h) Ridge welding.
 (i) T-spot welding.
 (j) Duplex spot welding.
 (k) Mash welding.

In the case of arc welding many of the systems that have been developed from the main groups are associated with the inventor's name or company exploiting the process.

ARC WELDING.

Commercial efficiency of arc welding processes is estimated by the relation that the quality of the work bears to the cost. The principal factors which decide the quality of the work are: (a) electrodes; (b) current (D.C. or A.C.); (c) welding equipment; and (d) efficiency of the welder. In assessing the cost the principal factors are: (a) capital expenditure; (b) depreciation and maintenance charges; (c) wages; (d) cost of electrodes and, (e) cost of electrical energy. The speed of welding enters into the question of cost provided increased speed is not obtained at the expense of welding efficiency. The absolute requisites for good work are: (a) low voltage; (b) short arc; and (c) constant heat per unit area in the weld. An arc welding circuit should possess the following characteristics: (1) ease of arc establishment when the work and electrode are cold; (2) freedom from undue tendency to freezing of electrode or extinguishing of arc with maintenance of short arc streams; (3) stable arc with maintenance of short arc streams; (4) limited current increase with growth of liquid globule; (5) limited increase of current at the instant and during the period the globule short circuits the arc stream; (6) the arc voltage should increase rapidly at the instant of globular detachment or on breaking a momentary short circuit in order to facilitate re-establishment of a stable arc. Welding nomenclature, definitions, and symbols have been standardised.

Carbon Arc Welding.

The characteristic principle of this process, originally known as the Benardos process, is that an electric arc is held between the metals to be joined, which forms one electrode of the circuit, and a carbon rod or pencil manipulated by the welder which forms the other electrode. By placing the carbon electrode in contact with the metal, thus closing the circuit, and instantly withdrawing it again, an electric arc is formed between the two electrodes and, owing to the high temperature of the arc, the metal is melted. Steel up to $\frac{3}{16}$ in. is satisfactorily melted together without using a filling material, but for greater thicknesses a 'melt bar' is melted into place by the heat of the arc. Experience has shown that the metal to be welded should be the positive electrode, and the carbon the negative electrode. If this order is reversed, a shorter arc is obtained and particles of carbon can enter the molten metal and cause brittleness. A long arc is the first essential of good carbon arc welding, the length varying from $1\frac{1}{2}$ to 3 ins.

The current used is direct, the voltage being from 60 to 90, and the amperes vary from 250 to 1,000, depending upon the thickness of the metal. The carbons generally used are $\frac{1}{2}$-in. and $\frac{3}{4}$-in. diameter.

The use of graphite electrodes in place of hard carbon electrodes is increasing as the merits of the graphite electrode are more widely known. The chief advantages are: (a) breakage in handling and in operation is appreciably reduced; (b) breakage and cracking due to sudden changes of temperature is eliminated; (*c*) less carbon is carried into the weld; (*d*) the arc is much steadier; (*e*) the life is from two to three times as long as for an amorphous carbon electrode of the same diameter; (*f*) electrical conductivity is four times that of the carbon electrode of the same cross-section, consequently the heating caused by resistance is negligible and much higher current densities can be carried. Considerable development is being made in the art of carbon arc welding where graphite electrodes are used. The single or double graphite arc has proved useful in the welding of non-ferrous metals. Automatic carbon electrode welding machines are now extensively used owing to the increased speed and lower costs obtained, as compared with hand work, and the reduction in distortion and overheating effects.

In one system of automatic carbon arc welding control of the arc is established by superimposing a strong magnetic field on the arc flame. This control permits the arc to travel through the various fields without disturbance, and the arc is given a gyratory motion of great velocity which also tends to hold it in a straight line. Increased strength, ductility, and high welding speeds have been obtained, and the process can be applied to light sheets or heavy plate.

Metallic Arc Welding.

The use of metallic electrodes for arc welding has proved more satisfactory than the use of carbon or graphite electrodes, which necessitate feeding the new metal into the arc by means of a rod or wire. A very wide variety of metal electrodes are used and can be classified as follows: (*a*) bare wire; (*b*) coated electrodes manufactured by dipping, painting, or spraying with a liquid; (*c*) covered electrodes, which are wrapped with a refractory material or otherwise encased in a much heavier covering than that used for coatings. They can also be classified as: (*a*) heavy slag depositing electrodes; (*b*) light slag depositing electrodes; (*c*) flux-coated electrodes which deposit no slag; and (*d*) bare wire. Sheathed electrodes have a flux between a metal core and sheath, whilst composite electrodes have one or more conducting materials combined mechanically with the flux. The ability of an electrode to fulfil its function depends upon: (*a*) chemical composition, (*b*) freedom from impurities, (*c*) grain structure, (*d*) surface finish. The effect of surface materials on electrodes from the standpoint of some of the scientific laws governing the metallic arc has been investigated and it has been shown that the presence of non-metallic materials on the surface is required. In metal electrode welding a very wide variety of metal electrodes are used and, in general, there is a sphere of utility for each type, and provided care is exercised in their selection, a weld of maximum efficiency should be obtained.*

The use of bare steel welding wires, that is, wires without a coating or covering, is extremely difficult, particularly if the wire contains appreciable quantities of elements such as carbon, manganese, etc., or if it is a question of using alternating current or doing overhead welding. The metal from bare wire is extremely 'wild' in the arc while it is being melted, and the presence of high stabilising factor in the equipment is desirable. The numerous advantages of coated and covered electrodes are now well recognised, and have led to a decreasing use of bare electrodes. Coated and covered electrodes aims at improving the metal in the weld, permitting the use of alloy steels, non-ferrous metals, etc., as electrodes, facilitating electrode manipulation and the use of alternating current. Covered electrodes may be placed in two general classes, the first class being the electrode which has a heavy covering of scientifically prepared material containing a deoxidiser. The second class of covered electrode is of the type which has a very thin coating on the wire and aims at obtaining the same results as the heavy covered electrodes without manipulating the heavy coating of slag deposited during the welding operation. The matching of the deposited metal with the base metal, which cannot be done with any certainty by varying the composition of the electrode metal, can be accomplished with the flux-coated electrode. By applying the correct composition of flux coatings to cores of uniform quality, sound welds in all the ferrous metals in general use can be obtained.

The percentage, by weight, of an electrode which passes over into a weld, sometimes called the deposit efficiency of the electrode, is affected by the following factors: (*a*) current, but not current density; (*b*) polarity; (*c*) thickness of the work; (*d*) chemical composition of the electrode; (*e*) surface materials of electrode; (*f*) occluded gases, slag inclusions, etc.; (*g*) arc length. Experiments so far recorded indicate that a good commercial electrode will show from 80 to 90 per cent. deposit efficiency, whilst a poor rod may show as little as 50 to 60 per cent. The bearing of this on welding costs merits attention.

Arc Welding Equipment.

Electric-arc welding may be done with a direct-current or alternating-current arc, having a reasonable degree of stability and sufficient amount of heat to accomplish fusion. The effect of the 'type' of current on the efficiency of metal deposition by the arc is threefold. It affects the speed of deposition, the amount of penetration or fusing of the metal, and arc stability. The design of the welding generator or transformer affects the cost of welding. The necessity for a special machine rests upon economic factors, and the differences between the various types used are those of power, cost of operation, and difference in speed of operation due to the relative stability of the arc.

* B.S.I. Specification Nos. 639, 640, 782. Electrodes for Steel Welding and Ship's Construction.

The question of direct or alternating current raises many interesting points which have been the subject of much discussion and research. Flux-covered electrodes are essential for certain types of alternating current machines, apparently because the lagging effect of the flux and slag maintains the necessary temperature during the periodical current reversals. Comparative data by independent authorities is missing, and the question of whether direct-current welding is faster or slower, whether flux-covered electrodes enable direct current to take the lead in welding speed, whether the practical losses in motor generator and resistance compare favourably or otherwise with the alternating-current system, etc., etc., are questions which are answered by contradictory claims by both parties to this controversy. However, there are advocates for both systems, and satisfactory equipment can be obtained for obtaining sound, efficient and economic welds with direct or alternating current.*

The successful application of arc welding is largely dependent upon the selection of the proper equipment. With incorrectly or badly designed equipment output tends to be decreased because the arc is difficult to hold, and because the operator is likely to become fatigued after a short period of continuous work. The type of welding equipment which will be most suitable for any particular installation depends upon the power available, the number of operators and the class of work for which it is required. The following cases arise: (*a*) where electricity is not installed; (*b*) where direct current is available; (*c*) where alternating current is available; (*d*) special circumstances, as, for example, where current is available and no transforming or converting equipment is required, or for outside work in exposed positions. Low first cost, low power consumption and negligible maintenance costs are important factors in choosing arc welding equipment and suggestions with regard to the most economical and efficient plant to fulfil any requirements can invariably be obtained without charge or condition from the technical departments of firms specialising in such equipment. There are two systems in general use—(*a*) single operator system, where a separate machine is provided for each operator, each machine receiving its current supply from the supply circuit; (*b*) multiple operator system, where more than one operator receives current direct from the same machine, but each operator is provided with a control panel, or devices which enable him to vary the current without interfering with the other operators.

Amongst the accessories are protective equipment and electrode holders.

Protective Equipment.

The intense glare emitted in the process of arc welding consists of a combination of invisible infra-red rays, visible light rays, and invisible ultra-violet rays, and special safety devices are required to protect the operator from their harmful effects. For light electric welding it is necessary only to protect the eyes with goggles fitted with suitable coloured glasses. Hand shields are made for medium-weight electric welding work which can be done with one hand. They serve the double purpose of protecting the eyes of the operator and also shielding the face from the heat rays and ultra-violet radiations. For heavy electric welding it is common practice to use a helmet or hood which provides protection for the eyes, head, face, and neck, and leaves the hands free. With regard to the tint of the protective glasses, a reference to the note on goggles in the section on oxy-acetylene welding will be useful. The filtering glass used should cut off both ends of the spectrum, the ultra-violet and infra-red, and transmit, in reduced quantity, only rays of high visibility, such as the yellow and green radiations, thus giving the maximum visibility with the minimum reception of energy. For the protection of the hands and wrists leather gloves are used. These devices are subject to modifications to meet the requirements of special work. Neighbouring workmen and operators should be protected from the dangerous rays of the electric arc.

Precautions in Arc Welding.

The precautions that should be observed in arc welding have been set forth in a memorandum issued by the Factory Department of the Home Office (Form 329, May 1925). These deal with: (*a*) hand-screen; (*b*) helmet or mask; (*c*) gloves; (*d*) aprons; (*e*) protection from hot metal; (*f*) chipping of slag. The memorandum also deals with the comparative dangers of direct and alternating currents, use of direct current, injurious effects of radiations, etc. The memorandum on the Electricity Regulations (Form 928, 1925) should also be consulted.

Automatic Metallic Arc Welding.

Depositing metal using an automatic feed for the electrode results in a materially increased rate of deposition. Great strides have been made in the past few years, and machines with completely automatic heads and travel carriages which can be applied to all types of welding have been developed. Machines using covered electrodes are in use, one type of which has a braided covering in which the flux is pressed into the interstices of the weave, thus avoiding the flaking off when passing through the feed rollers. Higher welding speeds, and strong ductile welds are obtained. This method of depositing metal is particularly advantageous on sections of uniform shape upon which a considerable amount of metal is to be deposited. Automatic welding does not eliminate the human element in welding, but does in certain cases greatly increase the speed of the process, and in other cases obtains results

* B.S.I. Specification No. 638. Rating of Arc Welding Plant.

which are beyond the limitations of manual operation. Automatic welding with the bare metallic electrode is in more general use at present than any other method, but, as indicated, the use of a covered electrode has been accomplished with improved results. In one type of machine the electrode is fed to the arc from a reel by means of feed rolls actuated by a 30-volt motor which is shunted across the arc when the arc is struck. Regulation of feed is maintained by devices which change the direction of rotation of the feed motor when the arc voltage increases or decreases above or below a certain amount. Research has proved conclusively that a faster, stronger, and more homogeneous weld can be made on material ½ in. or more in thickness by use of a multiple arc automatic welder, depositing the metal in two or more layers, using about 350 amperes in each arc, rather than by depositing in one layer, using an excessively high amperage. A number of welding applications involve all the requirements of automatic welding except that the size or weight of the pieces to be welded prohibits convenient handling, clamping, and traversing. In such cases a semi-automatic welding machine has proved to be of great advantage. One type of machine feeds the welding wire through a flexible conduit to a welding nozzle in which the control switch for the feed motor is located. The control mechanism operates to start and stop the feed motor. The maintenance of a uniform and steady arc depends on the operator.

Arc Welding Applications.

The application of arc welding to repair and manufacturing, particularly for structural welding and the replacement of castings, is very extensive, and should be made with a full knowledge of the possibilities and limitations of the process. In the welding of steels, with their great variety of physical properties, a familiarity with the wide range of conditions is necessary to meet and solve the problems presented. In all cases the steel to be welded should be identified, and not only should a suitable electrode be used, but also the correct welding technique applied. The operator should know how to prepare the parts, control expansion and contraction, control arc blow, and be familiar with the latest developments relating to the class of work he is engaged on. With regard to the subsequent heat treatment of welds the reports of the Welding Research Committee of the Institution of Mechanical Engineers should be consulted. In welding cast iron, copper, brass, aluminium, etc., the application of arc welding should be looked upon as in the development stage, where welds with physical, chemical, and mechanical properties similar to the base metal are required.

To obtain satisfactory arc welds on cast iron requires special consideration and care on the part of the operator. An essential to success is a suitable electrode, especially if the welded parts are to be machinable and fractures avoided. Such electrodes are available and represent the result of extensive researches in which the following points have been considered :—(1) preparation of the parts ; (2) prevention of slag inclusions ; (3) suitable values of current ; (4) method of deposition and its effect on the strength and hardness of the weld ; (5) reversed polarity versus straight polarity ; (6) effect of long versus short arc ; (7) effect of speed on the softness of the weld ; (8) the hard zone and its removal ; (9) relative weldability of various grades of cast iron ; (10) tests of electrodes.

In arc welding aluminium and its alloys a machine designed to hold a stable arc at low currents and, for light work, the addition of a stabilising resistance are essentials. A flux-coated electrode—the flux satisfying not only the conditions essential in gas welding but in addition decreases the surface tension of the aluminium and aids in stabilising the arc—is necessary. Reversed polarity, that is, with the electrode positive and the work negative, is used regardless of the alloy being welded, or the thickness of the material.

Cost of Arc Welding.

The cost of electric arc welding with the metal electrode process depends upon several variable factors, such as : (1) the price of labour ; (2) the characteristics of the arc, (*a*) arc easy to maintain, (*b*) arc difficult to maintain, (*c*) amount of heat available ; (3) the cost of power ; (4) the cost of electrodes ; (5) supervision required, (*a*) production work, (*b*) work of a varied character ; (6) interest charge on equipment.

Welding speed is fundamentally dependent upon the rate of metal deposition. This rate varies with such factors as, energy required for liquefaction of electrode material, arc stability, current density, and electrical resistance of electrode material. Comparative data showing the relative cost of arc welding and other systems of welding should be carefully studied, because omission to take into account one or more important factors may lead to misleading results. In general, it is conceded that metallic arc welding has economic advantages over other methods on steel plate above ½-in. in thickness, which increases with the thickness, but for the welding of non-ferrous metals the oxy-acetylene process is, at the moment, unrivalled.

Atomic Hydrogen Arc Welding.

The remarkable property of hydrogen to convert electrical energy to thermal energy and to utilise it for welding is the basis of atomic hydrogen welding. By striking an arc between two tungsten electrodes and blowing a stream of hydrogen gas through the arc core an extremely hot flame of a single gas with excellent welding characteristics is obtained. Standard equipments, which can be roughly classified as fixed and portable sets, and give a wide range of application and high welding speeds, are manufactured.

The welding technique has been worked out and although it is an arc method of welding, it has very similar characteristics to gas welding. The process can be used for almost any light gauge steel work, and is very suitable for welding alloy steels, such as stainless, and non-ferrous metals. Recent developments are a new three-phase torch for use with atomic hydrogen sets. This has the advantage of distributing the heat evenly over all three phases and increasing the energy in the arc for any given current.

In view of the success of atomic hydrogen welding the substitution of other gases for hydrogen has led to the introduction of the Arcogen process in which acetylene is used in conjunction with the arc. The process has not the advantages of atomic hydrogen, and is not used extensively.

The Cyc-arc process, named owing to the particular cycle of operations, is confined to the welding of steel or non-ferrous studs, rods, tubes, etc., on to steel plates by automatic apparatus which is portable. Once the apparatus has been set up for a particular size of stud or fitting its operation is automatic and the work can be done by semi-skilled labour.

RESISTANCE WELDING.

There is a marked distinction between the joining of metals by the two electric processes, viz., arc welding and resistance welding. In the former the heat is localised through the action of the electric arc and no pressure is applied; in the latter the heat is localised by the resistance to the flow of the current and pressure is used.

The process of resistance welding is based upon the phenomenon that a poor conductor of electric current will heat if current is forced through it, or that a good conductor will also heat if enough current is passed through it. Since an imperfect joint, such as the abutting ends of two pieces of metal, or the lapping of two pieces of metal which are to be welded, is a poor conductor and offers great resistance to the passage of the current, it will naturally heat and finally cause the metal to fuse and flow sufficiently to weld. In practice the operation is very rapid because comparatively large amounts of current are used.

For this type of welding a single-phase alternating current of any commercial frequency is generally used. The transformer in the machine will take the current from an ordinary shop circuit and reduce it to a suitable welding current.

Skilful development of resistance welding machines has made it possible to turn out welded products of high quality at mass-production speeds, and with a manufacturing economy which recommend them to any line of manufacture. In general, one machine is designed for handling pieces of the same metal and for making joints of similar design. Repair jobs have been done in special cases, but it is only in rare cases that such an adaptation is attempted.

A resistance welding machine is simply a single-phase transformer having a secondary of one or two turns, the voltage required between the terminals being only five to ten volts, and the current being limited only by the size of the pieces to be handled. The transformer is built into a heavy frame which supports the clamping jaws to handle the work and mechanical devices to regulate the heat and pressure. The characteristics of the transformer for welding service are influenced by many factors not encountered in other lines of transformer application. The major factors involved in the design of a transformer are: (*a*) the nature of the welding duty; (*b*) the mechanical features of the machine; (*c*) the load cycle; (*d*) the voltage used on the power lines in the plant where the welder will be in operation, and the cycles of that voltage; (*e*) the range of secondary or welding voltages to give the proper flexibility and the K.V.A. that will be required.

Practically all metals or combinations of metals may be joined by the resistance welding process. Special adaptations for heating rods, bars, billets, rivets, etc., and electric brazing and soldering machines, and special machines for a variety of purposes, such as welding hollow objects, zinc tube welding, etc., can be obtained.

The following factors have to be considered and co-ordinated when resistance-welded joints are contemplated; (*a*) current density per unit area; (*b*) time of current cycle; (*c*) pressure per unit area; (*d*) electrical conductivity of the parts to be welded; (*e*) heat conductivity of the parts being welded; (*f*) specific heat of the parts; (*g*) melting point of the materials; (*h*) medium in which the joint is made; (*i*) cooling stresses; (*j*) heat of fusion; (*k*) contact resistance. The elements usually considered are current, pressure, and time.

Butt Welding.

Electric butt welding consists in joining pieces end to end by securing complete fusion of the entire cross-section of the pieces. The parts to be welded are held in suitable clamps, and when the current raises the ends where they abut to the welding temperature, mechanical pressure is used to produce a weld. Contact is maintained by automatic means or by hand pressure, the current control may be automatic or by hand, and the upsetting gear may be hydraulically operated or controlled by hand. Sections up to 30 sq. ins. in area can be welded together.

Butt welding may be subdivided into three classes, namely, (1) upset welding, (2) flash welding, and (3) contact flash welding or automatic butt welding.

As regards the first subdivision, in which the weld is made by butting the cold ends of the stock to be welded and then applying the electric pressure, it is apparently being superseded for most purposes by flash welding.

Flash welding is made by applying the current before the ends are in actual contact and utilising the flash or minute arcs that play between the ends of the stock. Some advantages of flash over upset welding are (*a*) less power required, (*b*) less extrusion or upsetting, (*c*) burnt metal is thrown off, (*d*) increased production, (*e*) stronger welds. The convex shape of the extruded metal in upset welding is replaced by a thin fin in flash welding, which can be easily removed.

The third subdivision, contact flash welding, is in some respects similar to percussive welding. The weld is made almost instantly by using high continuous contact pressure and heavy current, the current being cut off by automatic methods of control.

Spot Welding.

Electric spot welding consists in pressing the pieces together while passing the current through them, and localising the flow of the current so that the weld is confined to 'spots' between the electrodes, which are used to obtain the pressure. The term automatic spot weld is given to a spot weld made on a welder which applies the pressure, and makes and breaks the circuit automatically.

The material to be welded is placed between die points or electrode tips made of copper or a suitable alloy, supported on the ends of two arms, and the points are brought firmly together on the metal by the use of either foot or motive power.* The current is then turned on. The metal, being a poor conductor, becomes heated, and when heated to a sufficient degree, pressure is applied. In the case of automatic control, the current is turned on and off and the necessary welding pressure applied automatically by controlling mechanism usually driven by a variable-speed motor. A recent development used with spot and seam resistance welding machines is the use of Thyratron tubes for closing and opening the electrical circuit which determines the time in which power is applied to a resistance welder. Plates up to ½ in., combined thickness 1 in., can be spot welded. For certain classes of work portable spot welders can be obtained. The majority of commercial metals can be spot welded, but it is essential that the metal to be welded should offer a considerable resistance to the electric current, otherwise sufficient heat cannot be generated.

Considerable investigation has been carried out with regard to heavy spot welding for ships and constructional work. Large machines capable of supplying a welding current up to 100,000 amperes, and equipped to exert a pressure up to 30,000 lbs. per square inch on each spot, have been constructed. The main difficulties which have been encountered are, briefly, electrode-tip protection and the preparation of the materials to be welded. Two secondary considerations have been the separation of the electrodes and the shape of the head of the spot welder.

The following indicate variations of spot welding:—

Button and Disc-Depression Welding.

Where very thick work is to be spot welded it is often desirable to find means to localise the current and cut down the amount of pressure required. Small metal discs or buttons can be used for this purpose placed between the head of the welder and the metal to be welded. Disc-depression welding differs from button welding in that in the disc-depression method an annular depression is provided in one of the sheets, around the disc. The method makes possible the use of a comparatively light machine for welding heavy steel plate, and also the use of less power and time for making the welds.

Bridge Welding.

This method is used to join thin sheets edge to edge without overlapping and without reducing the dimensions of the pieces. The sheets are brought together in their correct position, and then thin strips or discs are spot welded across the joint.

Multiple Electrode Welding.

Time and power can frequently be saved by making several welds at once, in which case the spot welder is equipped with two or more electrodes. It is advisable to have projections on one of the pieces to be welded to assist in localising the current. Successful multiple welding requires a separate transformer for each weld.

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* B.S.I. Specification No. 807—Electrode Shanks for Spot Welding Machines.

Projection Welding.

In this method the metal to be welded is provided with projections. The projecting parts are brought in line with each other, and the welding effect is to provide a smaller contact point which increases the resistance and allows the weld to be made with the consumption of less time and power. Ridge welding is the same as projection welding, except that, instead of a single projection, a ridge is raised in the metals in such a way that the ridges on the two pieces come together in the form of a cross. Projection welding, in which the two parts to be welded together have raised projections, not produced by shearing the sheets, is developing rapidly, and the machines for doing this work must have their electrical and mechanical details well co-ordinated to obtain desired results.

Duplex Spot Welding.

In this type of spot welding two transformers are used simultaneously on opposite sides of the sheets to be welded and their currents pass through two secondaries, welding two spots simultaneously.

Combinations of various spot welding methods have been made for special jobs. Sometimes it is only necessary to make a change in the construction of the electrodes to make a machine do work of a varied character.

Mash Welding.

Mash welding is a process of welding rods, wires, strips, etc., by crossing or overlapping the pieces and welding between relatively large electrode surfaces under pressure. The process is used in the manufacture of lamp-shade frames, wire waste baskets, etc.

Seam Welding.

This process practically combines the features of butt welding and spot welding, and is used for joining pieces along a continuous line or seam, usually by moving the joint in the path of the current, so that fusion is secured continuously. Excellent welds have been obtained by the step-by-step method in which the roll moves and stops alternately. During the rest period the rolls are energised and the welds made. The construction of all-steel motor bodies has given rise to a development which can be classified as flash seam welding, and consists of flash welding a long seam in a machine which really consists of two butt welding machines mounted on a single base. A 56-in. seam is flash welded in approximately 5 secs. Standard seam welders are usually equipped with two electrodes, which consist of power-driven discs between which the work is passed, or both electrodes may be converging rollers, with the converging edges resting on opposite parts of the work near the edge to be welded. The work is passed beneath the rollers, which carry the current into the metal, and makes a continuous line of welding. The work must be carefully cleaned and correctly positioned to get good results. The width and strength of the weld depend upon the current, pressure, and speed adjustments as well as upon the size and character of the electrodes and the shape, size, and character of the metal being welded.

Considerable development work has been done and seam welding is applied to the manufacture of steel barrels, casks, tanks, ventilators, heating and ventilating pipes, etc., with conspicuous success. The low cost of the joints and their reliability when care is taken to see that the material has good weldability are factors which are recognised. Welds may be butt, lap, or bridge, and may be made continuously or intermittently.

Electro-percussive Welding.

This is a process, sometimes called percussive arc welding, which is chiefly intended for manufacturing applications in which large quantities of duplicate parts are to be produced. The underlying principles utilised in percussive welding were discovered in 1905 by Mr. J. Chubb, of the Westinghouse Company. Following his original experiments, machines were developed for welding metals in wire form of rather small cross-section. Widely different physical properties offered no obstacle, and ductile welds can be made with copper and aluminium, platinum and lead. Electrical devices are used whereby a rapid small separation of the parts to be welded causes an intense arc to melt the surfaces. At the proper moment a forging hammer forges the pieces together. The advantages claimed for the processes are (*a*) power saved; (*b*) time of welding negligible; (*c*) welds of unequal sections are possible; (*d*) welds of unlike metals can be made; (*e*) uniform welds; (*f*) cost of finishing welds reduced. Recent results have shown that percussive welding has many valuable applications, especially in non-ferrous and alloy-steel welding.

Experimental work has been directed toward extension of the process to larger cross-sections. A large impulse transformer has been built capable of storing 20,000 joules. Sections up to ¾ in. diameter have been successfully welded and no apparent limitations are anticipated from a process point of view.

ELECTRIC CUTTING.

Although cutting with a blowpipe has many advantages over electric arc cutting, yet there are many places where the electric arc can be used advantageously. The groove is not so narrow as that cut with a blowpipe, nor is the surface cut as clean. One advantage of cutting with the arc is that scale, either on the surface or in pockets, cold shuts, sand pockets, and blowholes have no appreciable affect on the cut and will not extinguish the flame or arc as is the case in oxy-acetylene cutting.

The electric arc will cut any metal, ferrous or non-ferrous, and has been found particularly useful for the cutting of cast iron. The thickness that can be economically cut by the arc varies from 4 to 6 inches, for straight cutting. The current necessary for cutting operations on general and heavy cutting varies from 800 to 1,200 amperes. The voltage of the cutting arc varies between 25 and 45, a good average being 37. The best arc for cutting is a reasonably long one or a reasonably short one, whichever is preferred. The chief point is to have the liberation of energy near the work to be cut.

It will be seen that heavy values of current are required, and special apparatus has been designed so that the arc could be controlled with these heavy currents. An important part of these developments is the design of a handle to protect the operator, hold the electrode, and to carry the high amperage. A small item like the use of pure graphite instead of carbon for electrodes means the difference between success and failure.

Increase of current means larger electrodes and hence more metal to be cut out. One development for this work is a low resistance graphite electrode, and the approximate sizes for various current densities are as follows :—200 amperes, $\frac{1}{4}$ to $\frac{3}{8}$ in. electrodes; 300 to 400 amperes $\frac{3}{8}$ to $\frac{1}{2}$ in. electrodes; 400 to 600 amperes, $\frac{1}{2}$ to $\frac{5}{8}$ in.; 600 to 800 amperes, $\frac{5}{8}$ to $\frac{3}{4}$ in.; 800 to 1,200 amperes, $\frac{3}{4}$ to 1 in. electrodes.

Cutting on heavy sections is generally best accomplished from the underneath side or from a vertical face tipped slightly greater than 90 degrees so that the molten metal can flow away, although the action is not entirely simply melting metal. In cutting with the arc a reasonably smooth surface can be obtained by paying attention to one side only, or the cut may be made rough and trimmed up smooth by a second application of the arc.

If covered metallic electrodes are used, the electrode should be soaked in water previous to cutting and, during the cutting operation, should be kept cool by dipping in water at intervals. A long arc should be used and a current of about 200 amps. in conjunction with a mild steel electrode of about 8 gauge.

A special electrode for cutting with the arc, and for such work as piercing holes has been designed. This special electrode, covered with a coating giving off free oxygen under the temperature of the arc, gasifies the entire metal melted, so that the boring action is uninterrupted after once started. This electrode is used by starting the arc and immediately pushing the electrode into the puddle. After the hole is through the piece it can be widened to any size or shape by drawing the arc around its edges. This type of electrode only takes from 250 to 350 amperes. A development in under-water cutting is an adaptation of the carbon electrode arc cutting process, in which the carbon electrode is pierced by two small brass tubes, through which a special cutting gas is supplied from the surface. The gas oxidises and blows away the metal melted by the arc, and also converts the surrounding water into steam, thus making a gaseous envelope for the arc. The combined action serves to greatly speed up the operation of the cutting electrode, as the metal is blown aside before it has a chance to solidify. This device has been used successfully for cutting metal between three and four inches thick at a depth of about 70 ft.

COPPER HYDROGEN-ELECTRIC WELDING.

This is the title of a new automatic continuous welding process which welds without flame or arc, and produces welds on a quantity production basis with mechanical properties equal and in some cases, superior to the base metal. The parts to be welded are assembled by a snug fit, either by inserting one part into the other or by spot welding, or pinning the parts together. Copper, either in the form of copper wire or copper paste, is applied to the joint and the assemblies are placed in a hydrogen-filled heating zone of a special furnace, and gradually brought up to a maximum temperature of 1500° C. Some of the copper goes into solid solution in the steel, and the rest produces a copper-iron alloy. Under normal conditions no free copper remains between the parts. The hydrogen atmosphere reduces all oxides, scale, etc., on the surface of the parts and provides chemically clean surfaces for the welds. The welds are hardly perceptible, and emerge from the furnace cooled to room temperature, clean and free from scale or oxide. The time occupied in welding is 4 mins., and no assembly seems too intricate for the process which has been in use for a considerable time.

THERMIT WELDING.

Thermit is a trade name for a mixture of finely divided aluminium and iron oxide, which when ignited reacts to produce a superheated liquid steel at 5000° F. The underlying principle of the process is the high chemical affinity of aluminium for oxygen. Up to a temperature of 2800° F. thermit is an inert mixture. At that temperature the aluminium unites with the oxygen of the iron oxide and comes down as liquid steel at about twice the temperature of ordinary molten steel. If steel at this high temperature is poured around the sections to be united, especially if the sections have previously been preheated, it will melt them and amalgamate with them so as to form a fusion weld.

Several varieties of thermit are produced and used for welding ferrous material, *i.e.* plain, cast iron, railroad, and wobbler thermit.

In making a thermit weld, the parts to be united are first lined up with a space between the ends, the extent of which depends upon the size of the sections to be welded. The ends are thoroughly cleaned and a wax pattern is formed around them, the exact shape of the reinforcement of thermit steel which is to be cast at that point to make the weld. A sand box is next rammed around the pattern and inside a sheet iron box, provision being made for pouring gates, heating gates, and risers. The parts to be united are brought to a good red heat by suitable means, such as compressed air liquid fuel. The charge of thermit is placed in a conical shaped crucible supported over the pouring gate of the mould and ignited by means of an ignition powder. In 25 to 35 seconds the reaction is completed and the crucible tapped. The slag does not enter the mould.

The process is extensively employed for locomotive repairs, marine repairs, crankshaft repairs, steel mill repairs, pipe welding, rail welding, and miscellaneous heavy welding. Detailed instructions are available for carrying out thermit welds in the fields mentioned. The process is adapted to all sizes and weights of pipe up to 6 in. diam., and an important feature possessed by the thermit pipe welding process is the ease with which it can be applied to the welding of coils. The joints can be welded either before or after the pipes are bent.

General Information.

A lack of familiarity with the various welding processes and the established technique is still a contributing factor to the retardation of progress. The primary problems, such as the training of welders, tests for welds, welding wire specifications, technique of welding, investigations on rail joints, pressure vessels, and structural steel, have received attention. There still remains a large number of fundamental research problems in welding which can be classified under the following heads: (*a*) investigations involving physical tests; (*b*) involving considerable metallurgical experience; (*c*) problems in physics; (*d*) problems involving welding procedure; (*e*) problems involving chemistry; (*f*) structural problems; and (*g*) psychological problems. No less than fifty separate problems, arranged in these groups, have been listed and the opportunity for real service to the welding industry by their solution is obvious. The promotion and co-ordination of research in the welding industry is essential in view of its vast expansion.

Co-operation in the advancement of welding, both by the users concerned with its efficient application or as manufacturers of welding equipment and supplies, is reflected in the rapid growth of welding. Welding is becoming regarded as the standard method of joining metal parts in general manufacturing, construction, production and maintenance work. Some of the recent advances in welding have made possible light-weight railway equipment, aircraft structures and automotive construction of all kinds. This saving in weight is materially affecting the design of structures, ships, machinery construction, etc. In spite of all these advances in welding new opportunities are almost limitless in their extent.

Progress is illustrated by the issue of B.S.I. specifications for: bare rod or wire electrodes and covered electrodes for arc welding wrought iron and mild steel; oxy-acetylene welding applied to steel structures; cold bend test on welded joints; protective glass for welding; rating of arc welding plant, equipment and accessories; electrodes for shipbuilding purposes; electrodes for spot welding machines; impact tests for welds. In addition the following are in course of preparation: co-ordination of welding tests; steel sections for welding; welding rods for oxy-acetylene welding. The publication, in six volumes, of the Proceedings of the Twelfth International Congress on Welding and Allied Industries adds a rich store of information to the rapidly growing literature on welding.

The maintenance of quality standards and welding procedure, as embodied in welding specifications, will certainly lead to further expansion since the elimination of doubt as to quality of workmanship will bring industry in general to look upon welding with greater confidence. The British Standard Specifications for 'Fusion Welded Steel Air Receivers'; 'Metal Arc Welding as

Applied to Steel Structures'; 'Oxy-acetylene Welding as applied to Steel Structures,' 'Tentative Requirements for Fusion Welded Pressure Vessels intended for Land Purposes' issued by Lloyds Register of Shipping, and the issue of provisional rules for welding in ships by both Lloyds and the British Corporation Register of Shipping and Aircraft, indicate the growing importance of welding.

The British Standards Specification No. 499, dealing with nomenclature, definitions and symbols for welding and cutting has been revised. The revisions of the symbols and the method of indicating welds on detail drawings should lead to their general application, particularly as this specification has been frequently referred to in other specifications.

A Welding Industry Committee has been constituted by the B.S.I. In addition to the standards already mentioned the following specifications have been issued by the B.S.I.:

Methods of testing fusion welds, welded joints and weld metal. (Applicable to the electric arc welding of steel.) 709—1940.
Metal arc welding as applied to tubular steel structural members 938—1941.
Welded joints in copper vessels. 1077—1942.
Gas welding of aluminium. 1126—1943.
Test pieces for production control of aluminium alloy spot welds. 1138—1943.
Spot welding for light assemblies in mild steel. 1140—1943.
Symbols for shipyard drawings. 1303—1946.
Tests for use in the training of welders. 1295—1946.

With regard to the tests the standard lays down the general requirements. There are eleven tests, covering butt welds in plates and piping, fillet welds in plates and sheets, pipe branch welds, building up of surfaces and test pieces for radiographic examination. The preparing of etched specimens and notes for the guidance of instructors are given in appendices.

The London County Council has modified the London Building Act so as to permit the use of oxy-acetylene welding instead of riveting, bolting or lapping.

Amongst the fundamental researches which have been completed, are the following:—

'Welded Beam-Column Connections.'
'The Effect of Low Temperature on the Tensile Impact Resistance of Iron, Steel and Welded Joints.'
'Strain Measurements in Welded Joints.'
'Welded Structural Brackets.'
'X-Ray Methods of Studying Stress Relief in Welds.'
'Peening and its Effects on Arc Welds.'
'Investigation of Plug and Slot Welds.'
'Alternating Current—Non-destructive Test for Welded Seams.'
'Fatigue Tests of Butt Welds in Structural Plates.'
'Static and Impact Tensile Properties of Stainless Steel Welds at Ordinary and Low Temperatures.'
'Electric Heating by the Proximity Effect.'
'Effect of Welded Top Angles on Beam-Column Connections.'
'Characteristics of a Universal Welding Generator.'
'Copper Welding.'
Welded Girders with Inclined Stiffeners.'

With regard to acetylene welding equipment, plants have been installed equipped with a compressor for boosting the acetylene obtained from a generator up to a pressure permitted by law (9 lb. per sq. in.) and distributing the gas at this pressure to the welding shop. There is a growing tendency to distribute the oxygen by a pipe line supplied from a battery of cylinders. At the moment it is advisable to hand over this work to the firms specialising in this class of work, in view of the hazards involved.

A complete revolution in the welding technique as applied to the welding of steel by the oxy-acetylene process has taken place. Instead of the so-called neutral flame, in which the oxygen and acetylene are consumed in almost equal volumes, an excess acetylene flame is used which serves to carburise the surfaces of the bevelled edges and the deposited metal wets the surfaces and produces complete fusion. With this type of welding larger flames are used in conjunction with a smaller V. The welding speed is increased and greater amounts of alloying constituents are, as a result, retained in the deposited metal. A distinctive feature of this method, which has been thoroughly investigated, is that a perfect bond can be obtained without melting the parent metal. The process is known as Lindewelding.

The welding of stainless steels and high chromium high nickel steels have been investigated. The former can be welded by all the welding processes but in order to avoid 'weld decay' and subsequent corrosion the carbon content should be very low; the latter, in the form of heat-resisting alloys, are easy to weld without a flux provided the flame of the blowpipe is well regulated, and a welding rod of similar characteristics used with the correct power of flame.

The welding of medium carbon steels (0·6 per cent.), which is usually looked upon as difficult, has been investigated and excellent results are obtained with a welding rod possessing similar properties to the parent metal but with a lower carbon content. The manganese-chrome steel used for building up worn rails gives the best results when used as a welding rod on this class of work.

In the field of electric arc welding the progress is so considerable, particularly in the structural field, that more and more attention is being paid to the properties of the electrode to be used. The fact that the specification Committees are issuing arc welding specifications in which protection against atmospheric contamination during welding is provided for, shows the necessity for scientifically designing electrodes for various classes of work. The leading manufacturers are coping with this aspect of arc welding and are producing electrodes which will satisfy any requirement or specification.

In the testing of welds considerable development has taken place in non-destructive methods. The X-ray examination is becoming a recognised test and, in Lloyds requirements for fusion welded pressure vessels, X-ray photographs are to be taken of the entire length of each welded seam. The magnetic test is now represented by portable equipment which reveals the presence of defects in a satisfactory manner. The Schmuckler test which mills out a portion of a weld and allows the weld to be etched and examined at any particular point, is also being used, since the portion removed can be re-welded without difficulty or appreciable cost.

For the specifications relating to welded structures destructive tests, which include tensile and bend tests, are required.

The Symposium on the Welding of Iron and Steel organised in 1936 by the Iron and Steel Institute, with the co-operation of fifteen other Institutions or Societies, had for its objects: (*a*) the examination of the art of welding as it existed at present; (*b*) consideration of the difficulties encountered; (*c*) the scientific work endeavouring to solve the difficulties; and, the most important object, (*d*) to determine the directions in which further work was necessary and the factors of importance which had not emerged and should be examined.

The Report on the Symposium contains sections, two of which deal with the account of proceedings subsequent to the Symposium and proposals for the future organisation of research on the welding of iron and steel. The endorsement of the Report by the Societies and Technical Institutions enumerated in the Second Schedule constitutes the authority for the Institute of Welding to carry out, on behalf of the engineering industries, the programme of research referred to in the Report.

The British Welding Research Association, which has taken over the programme of the Welding Research Council, has for its objects and activities the following :—

To ensure that there shall be continuous progress in the methods and materials used for the welding of all types of metals, and to investigate the production of weldable materials in all metals; to provide codes of good practice for the control of the various arc, gas and resistance welding processes. The publications of the Association include :—

'Arc Welded Structural Steelwork,' 1946.

'Investigation into the Pressure Welding of Welded Rigid Frame Structures, 4th Interim Report,' 1946.

'Pressure Welding of Light Alloys without Fusion,' 1946.

'Technique for the Gas Welding of Magnesium Alloys,' 1946.

In view of the large number of problems which require solution, or the provision of additional information, the Association has drawn up a programme of work, based on priorities, which has been adopted by the Council.

The Welding Research Council was set up in 1936 by the Institute of Welding to investigate welding problems.

The formation of an international welding organisation which will be open to technical institutes, research bodies and similar associations, not engaged in commercial or trade activities, is well under way, as a result of a conference held at the Institute of Welding. Welding societies of fourteen countries were represented. It will cover all welding processes and nothing but welding.

A large number of films covering the technique and application of various processes industrially have been made and can be obtained on loan for lecture purposes.

The Associated Offices Technical Committee, which represents five of the leading insurance companies, has issued a schedule of examinations and tests during construction and upon completion of metallic arc welded drums for steam generating units and steam receivers

See also Descriptive Section XXIII, Part II.:

Barimar, Ltd.
Murex Welding Processes, Ltd.

SECTION XXIV

THERMODYNAMICS

(pp. 1299–1305)

(By R. H. Parsons, M.I.Mech.E.)

SECTION XXIV

THERMODYNAMICS.

By R. H. Parsons, M.I.Mech.E.

Below will be found the thermodynamical formulae of most general use to engineers, with explanations of their application where necessary. For fuller information the reader is recommended to consult 'Thermodynamics for Engineers,' by Sir J. A. Ewing, and 'The Properties of Steam,' by Prof. H. L. Callendar.

Symbols used in this Section.

P = Absolute pressure in lb. per sq. in.
V = Volume of 1 lb. weight of gas in cubic ft.
T = Absolute temperature in deg. Fahr. = thermometer reading plus 459·6°.
R = The gas constant in ft.-lb. It is the work done by the expansion of 1 lb. weight of gas under constant pressure when the temperature is raised by 1° F. The value of R for any gas is very approximately equal to 1543 ÷ molecular weight of gas. For air R = 53·4. Also R = 778 ($C_p - C_v$) = 778 C_v ($\gamma - 1$).
C_p = Specific heat at constant pressure, or the B.Th.U. required to raise the temperature of 1 lb. weight of gas by 1° F. at constant pressure. For air C_p = 0·24.
C_v = Specific heat at constant volume, or the B.Th.U. required to raise the temperature of 1 lb. weight of gas by 1° F., the volume being kept constant. For air C_v = 0·1716.
γ = Ratio of specific heats = $C_p \div C_v$. For air γ = 1·4.
W = Work in ft.-lb. done by 1 lb. weight of gas.
= Entropy. (See later.)
= Quantity of heat in B.Th.U.
= Total heat in 1 lb. weight of gas in B.Th.U.

N.B.—When logarithms are used in the formulae, they are always ordinary logarithms to the base 10.

Formulae relating to Perfect Gases

All gases which are far from their liquefying points, including air, flue gases, and products of combustion, may for most practical purposes be considered as perfect gases. The specific heats of perfect gases are constant at all pressures and temperatures.

The temperature, pressure, or volume of a perfect gas may be found when the appropriate value of R and of the other two variables are known, by means of the Equation of State

$$144PV = RT \qquad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

If the temperature after any change of volume or pressure is the same as it was before the change, we have the relationship

$$P_1V_1 = P_2V_2.$$

Isothermal Expansion or Compression.

The work in ft.-lb. done per lb. weight of a perfect gas during its expansion at constant temperature is given by

$$W = 144 \times 2{\cdot}3026\, P_1V_1 (\log V_2 - \log V_1) \qquad . \quad (2)$$

or by the equivalent expressions

$$W = 2{\cdot}3026\, RT (\log V_2 - \log V_1)$$

$$W = 144 \times 2{\cdot}3026\, P_1V_1 (\log P_1 - \log P_2).$$

It should be noted that the above formulae give either the total work during expansion alone, as measured by the area between the expansion curve and the line of zero pressure, or the total work as measured by the complete theoretical indicator diagram.

The same formulae also give the negative work required to compress the gas at constant temperature, the initial state in either case being represented by P_1V_1.

The equation to the curve of isothermal expansion or compression is PV = constant, the curve being a hyperbola.

The heat in B.Th.U. which must enter the gas during expansion or leave it during compression, in order that the temperature may remain constant, is equal to $W \div 778$.

The change in entropy caused by isothermal expansion or compression is found by dividing the heat added or withdrawn, by the absolute temperature.

Adiabatic Expansion or Compression.

During an adiabatic expansion or compression, no heat enters or leaves the gas, nor is any heat generated in it by friction or eddies.

The work in ft.-lb. done per lb. weight of perfect gas expanding adiabatically is given by—

$$W = \frac{144\,(P_1V_1 - P_2V_2)}{\gamma - 1} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

or by the equivalent expressions

$$W = \frac{R(T_1 - T_2)}{\gamma - 1} \text{ or } W = 778\; C_v(T_1 - T_2)$$

These expressions also apply to adiabatic compression, though in this case the work is given as a negative quantity if $P_1V_1T_1$ represent the initial conditions.

It is important to observe that the above formulae give the work done during expansion or compression alone, and take no account of the work done by the entrance of the gas, or of the work required to discharge it. They give, therefore, only the area of a diagram lying between the expansion curve and the line of zero pressure.

To get the total work done in the expansion or compression of a lb. weight of gas, as represented by the area of the complete indicator diagram, the following formula must be used,

$$W = \frac{144\,\gamma}{\gamma - 1}(P_1V_1 - P_2V_2) \quad . \quad . \quad . \quad . \quad (3A)$$

or its equivalents

$$W = \frac{\gamma}{\gamma - 1} R(T_1 - T_2) \text{ or } W = 778\; C_p(T_1 - T_2)$$

Before formulae (3) or (3A) or their equivalents can be applied in practice, it is usually necessary to determine first the values of P_2, V_2 or T_2. This can be done by means of the following relationships.

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^{\gamma} \text{ or } \log P_2 = \log P_1 + \gamma\,(\log V_1 - \log V_2) \quad . \quad . \quad . \quad (4)$$

$$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}} \text{ or } \log V_2 = \log V_1 + \frac{1}{\gamma}(\log P_1 - \log P_2) \quad . \quad . \quad . \quad (5)$$

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{\gamma - 1} \text{ or } \log T_2 = \log T_1 + (\gamma - 1)\,(\log V_1 - \log V_2) \quad . \quad (6)$$

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \text{ or } \log T_2 = \log T_1 + \frac{\gamma - 1}{\gamma}(\log P_2 - \log P_1) \quad . \quad (7)$$

The equation to the adiabatic curve is $PV^{\gamma} = \text{constant}$.

For air $\gamma = 1{\cdot}4$, $\frac{1}{\gamma} = \frac{5}{7}$ or $0{\cdot}714$, $\frac{\gamma - 1}{\gamma} = \frac{2}{7}$ or $0{\cdot}286$.

In adiabatic expansion or compression the entropy of the gas is unchanged.

The General Case of Expansion or Compression.

Isothermal and adiabatic processes represent ideals with which actual processes may usefully be compared, though they can never be attained in practice. In every actual case of expansion or compression some heat enters or escapes from the gas, or is generated within it by friction or eddies. The expansion curve of a dry gas may, however, always be represented by $PV^n = \text{constant}$, if a suitable value be assigned to n. Hence by substituting n for γ in the adiabatic equations (3), (3A) and (4) to (7) these equations enable the area of the theoretical indicator diagram to be obtained, as well as the changes in pressure, volume and temperature.

In the case of a water-jacketed air-compressor the value of n will be between $1{\cdot}25$ and $1{\cdot}35$, according to the effectiveness of the cooling arrangements. For the expansion of superheated steam without gain or loss of heat, $n = 1{\cdot}3$. For wet steam the value of n may be taken as $1{\cdot}035 + 0{\cdot}1x$, in which x is the dryness fraction at the beginning of expansion, but this is not very accurate, and for all calculations concerning steam it is preferable to make use of steam tables or diagrams.

Although, in accordance with what has been said above, when a gas expands or is compressed according to the law PV^n = constant, the area of the theoretical indicator diagram is always given by the expression

$$\frac{144\,n}{n-1}(P_1V_1 - P_2V_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

it is important to notice that the area of the diagram does not represent the work when n is greater than γ. This is the case when any heat is entering the gas from outside or is generated within it by friction or turbulence. The work is then given by the equation

$$W = 778\, C_p\,(T_1 - T_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

The final temperature T_2 being first obtained from whichever is the more convenient of the formulae :—

$$\log T_2 = \log T_1 + (n-1)(\log V_1 - \log V_2) \quad . \quad . \quad . \quad (10)$$

or

$$\log T_2 = \log T_1 + \frac{n-1}{n}(\log P_2 - \log P_1) \quad . \quad . \quad (10\text{A})$$

The quantity of heat, measured in B.Th.U. which enters or leaves a pound of gas when the latter changes its state according to the law PV^n = constant, is given by the equation

$$Q = \frac{\gamma - n}{n-1} C_v(T_1 - T_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

or its equivalent

$$Q = \frac{\gamma - n}{\gamma - 1} \times \frac{P_1V_1 - P_2V_2}{n-1} \times \frac{144}{778} \quad . \quad . \quad . \quad . \quad (11\text{A})$$

When the entering heat is generated in the gas itself by friction or eddies, as in the case of an uncooled turbo-compressor, or is received from the cylinder walls, as in an I.C. engine, additional work equal to this heat has to be done by the compressor over and above the work shown on the theoretical indicator diagram. Similarly when the gas expands with frictional losses, as in a turbine or a nozzle, the area of the theoretical indicator diagram is greater than the work actually done by the gas, by the equivalent of the frictional heat.

Equations (11) and (11A) enable the heat carried away by the cooling water of a compressor to be calculated. They also give the heat equivalent of the work in excess of the indicator diagram which has to be done by an uncooled compressor or blower. The value of n for such a machine can be calculated from the formula

$$\frac{n-1}{n} = \frac{1}{\epsilon}\left(\frac{\gamma - 1}{\gamma}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

where ϵ represents the efficiency of the compressor.

When a gas expands or is compressed according to the law PV^n = constant, the change of entropy caused by the process is

$$\phi_1 - \phi_2 = C_v\left(\frac{\gamma - n}{n-1}\right) \times 2{\cdot}3026\,(\log T_1 - \log T_2) \quad . \quad . \quad (13)$$

the value of T_2 being obtained from formula (10) or (10A).

The Efficiency of a Heat Engine.

It is proved in the textbooks on Thermodynamics that even an ideal engine working between the temperatures T_1 and T_2 cannot turn into work more of the heat supplied to it then the proportion represented by the fraction

$$\frac{T_1 - T_2}{T_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

This fraction therefore represents the maximum possible efficiency of any heat engine. There are only three cycles of operation by which such an efficiency could theoretically be obtained, the best known being that of the Carnot engine, hence the efficiency as computed above is called the 'Carnot efficiency.' The other cycles giving the same theoretical efficiency are the Stirling and Ericsson regenerative cycles.

The Efficiency of a Steam Engine—Rankine Cycle.

The efficiency of a steam engine or turbine is generally compared with that of an ideal engine working on the Rankine cycle. The maximum quantity of heat that could be turned into work by an ideal engine working on the Rankine cycle is given by the expression

$$U = H_1 - H_2 + T_2\,(\phi_2 - \phi_1) \quad . \quad . \quad . \quad (15)$$

In the case when the condition of the steam after its adiabatic expansion is dry or superheated, the expression takes the simpler form

$$U = H_1 - H_2.$$

The values of H, T and ϕ are to be taken from steam tables H_1 and ϕ_1, referring to 1 lb. of steam in its initial dry or superheated condition, and H_2 and ϕ_2 to 1 lb. of dry steam at the temperatures T_2.

The value of U may also be read directly from a Mollier diagram (q.v.) without any calculation. It is known as the 'adiabatic heat drop.'

The theoretical steam consumption of an ideal engine working on the Rankine cycle, is

$$\frac{2545}{U} \text{ lb. per h.p.-hour, or } \frac{3412}{U} \text{ lb. per kW. hour.}$$

The 'efficiency ratio' of an engine or turbine is the ratio between the theoretical steam consumption as calculated above, and the actual steam consumption.

Owing to the departures from the Rankine cycle which characterise modern steam practice, the use of the 'efficiency ratio' as a figure of merit is falling into disuse, and steam turbines are now usually compared on the basis of their total heat consumption per kW. hour.

The Efficiency of an Internal Combustion Engine.

The efficiency of any internal combustion engine is usually expressed by comparing it with that of an ideal engine working on the 'air-cycle.' In the air-cycle, the working substance is supposed to have constant specific heat, which is not true for the actual gases. Hence the air-cycle standard is mainly serviceable as a convenient criterion for the comparison of different engines.

The efficiency of an ideal engine working on the air-cycle is given by the expression

$$1 - r^{0.4}$$

in which

$$r = \frac{\text{clearance volume}}{\text{piston displacement and clearance volume}}$$

For formula more exactly representing the conditions of working of actual engines of various types, reference should be made to the 'Report of the Heat Engine Trials Committee,' 1927.

The Efficiency of a Compressor.

Isothermal Efficiency.—The efficiency of an air compressor is usually expressed as the ratio of the work theoretically necessary for isothermal compression to the work actually required by the machine. The work necessary for isothermal compression may be calculated from formula (2) or its equivalents.

Volumetric Efficiency.—This is the ratio of the volume of free air delivered, reduced to normal temperature and pressure to the volume swept out by the piston of a reciprocating compressor. The volumetric efficiency of a well-designed compressor is from 85 to 90 per cent.

The Efficiency of a Refrigerating Machine.

The efficiency of a refrigerating machine is stated in terms of its 'Coefficient of Performance,' which is the ratio of the heat extracted to the work spent, both being of course measured in the same units. If heat be extracted from a substance at T_2 and delivered at T_1 the maximum possible coefficient of performance of an ideal refrigerating machine is

$$\frac{T_2}{T_1 - T_2} \qquad \text{(16)}$$

If an ideal refrigerating machine take in heat at T_2, and be driven by an ideal heat engine taking in heat at T, both machines rejecting heat at the temperature T_1, then the maximum possible ratio of Q_2, the heat extracted from the cold body, to Q, the heat supplied to the engine, is

$$\frac{Q_2}{Q} = \frac{T_2 \,(T - T_1)}{T\,(T_1 - T_2)} \qquad \text{(17)}$$

Entropy.

For the practical purposes of the engineer, entropy may be regarded as a pure number related to the thermal condition of a substance. Its physical significance is that every increase in entropy implies that heat has become less available for doing work. Entropy changes whenever heat enters or leaves a body, or is generated within it by frictional processes. Entropy is the one quality of a substance which does not change during adiabatic expansion or compression, although temperature, pressure, total heat, and internal energy are all changing.

Entropy is measured from some arbitrary zero, as its changes are alone of importance. The entropy of steam and water is always reckoned from the freezing point of water, at which the entropy is considered to be zero. The values of entropy for steam and water, per lb., are given in steam tables. Any area on a diagram which has absolute temperature and entropy for ordinates represents work measured in heat units. The ways in which the entropy of steam and water can be calculated are explained in connection with the description of the temperature-entropy diagram below.

In isothermal expansion or compression, and in evaporation or condensation at constant temperature, the change of entropy is equal to the heat added or withdrawn, divided by the absolute temperature. In adiabatic processes there is no change in entropy.

In the general case of a gas changing its state from $P_1V_1T_1$ to $P_2V_2T_2$, the change of entropy is calculated by the following formula—

$$\phi_2 - \phi_1 = 2{\cdot}3026 \left\{ C_v(\log T_2 - \log T_1) + R(\log V_2 - \log V_1) \right\} \quad . \quad . \quad (18)$$

or its equivalents—

$$\phi_2 - \phi_1 = 2{\cdot}3026 \left\{ C_v(\log P_2 - \log P_1) + C_p(\log V_2 - \log V_1) \right\}$$

or

$$\phi_2 - \phi_1 = 2{\cdot}3026 \left\{ C_p(\log T_2 - \log T_1) - R(\log P_2 - \log P_1) \right\}$$

The Temperature Entropy Diagram.

If a diagram be prepared with absolute temperatures and entropies as the ordinates the area of any closed curve on this diagram represents work expressed in heat units. A temperature-entropy diagram for steam is shown in fig. 1.

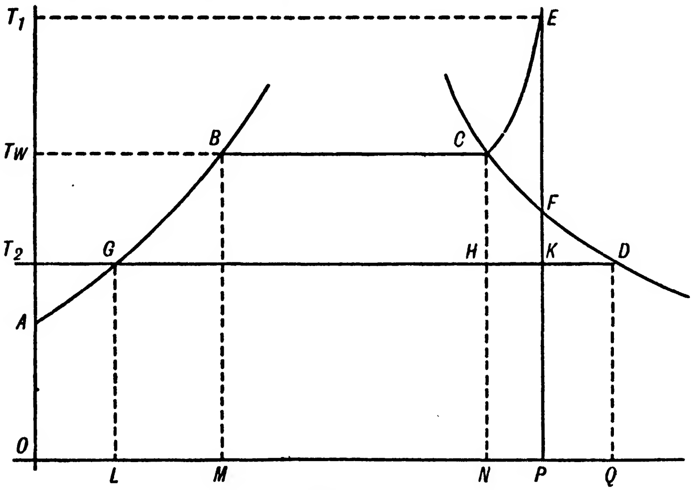


Fig. 1.

The lines AB and CD are the foundation of the diagram. AB is the water-entropy line, the point A for which the entropy is zero, being at the freezing temperature of water, or 491·6° F. abs. The line may be plotted by taking the entropy of water at various temperatures from steam tables, or the value of the entropy of water at any temperature may be calculated (on the supposition that the specific heat of water is unity at all temperatures) from the formula

$$\phi w = 2{\cdot}3026 \log T - 6{\cdot}1976 \quad . \quad . \quad . \quad . \quad . \quad (19)$$

OD is the saturated steam line. It may be plotted from the entropy of saturated steam as given in the steam tables, or from the formula

$$\phi s = \phi w + \frac{L}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

in which L is the latent heat of steam at the temperature in question.

The use of the diagram may be shown by considering its application to an engine working on the Rankine cycle, taking superheated steam at a temperature T_1, expanding it adiabatically to a temperature T_2, and discharging it in a wet condition to a condenser at this temperature.

The heating of the water is represented by the line AB, the quantity of heat required to raise it from the condensate temperature T_2 to the evaporation T_w being the area LGBM. The line BC represents the evaporation of the water, the heat taken in during this process being MBCN. The superheating of the steam is shown by the line CE, the heat taken in during superheating being NCEP. The line CE may best be plotted from the values of the entropy of superheated steam at the appropriate pressure but at various temperatures, given in the steam tables. Alternatively, the value of entropy may be approximately obtained from the equation

$$\phi h = 1{\cdot}1052\,(\log T - \log T_w) \quad . \quad . \quad . \quad . \quad . \quad (21)$$

In this equation the assumption is made that the specific heat of superheated steam is constant and equal to 0·48, which is only roughly correct.

The line EK shows the temperature drop due to the adiabatic expansion of the steam from T_1 to T_2, and condensation at constant temperature T_2 is represented by the closing line KG of the diagram. The heat rejected to the condenser is the area LGKP.

The dryness-fraction of the steam after expansion is the ratio $\frac{GK}{GD}$. The expanding steam loses all its initial superheat when at a temperature and therefore at a pressure corresponding to the point F. Had the steam been dry, but not superheated, before expansion, the adiabatic expansion line would have been CH, the dryness fraction $\frac{GH}{GD}$, and the heat rejected to the condenser LGHN.

The 'available heat' or the heat which would be turned into work by a perfect engine is the area GBCEK, and the efficiency of the cycle is the ratio of this area to the whole of the heat taken in, namely the area LGBCEP. Had the steam had no initial superheat, the available heat would have been GBCH and the efficiency would have been the ratio of GBCH to LGBCN.

The available heat and the dryness fraction are two of the most important facts to be deduced.

To compute the available heat from the data given in the steam tables, we have

$$\text{Available heat} = H_1 - H_2 + T_2(\phi_2 - \phi_1) \quad . \quad . \quad . \quad . \quad . \quad (15)$$

in which H_1 is the total heat of a lb. of steam at pressure and temperature P_1T_1, while H_2 is the corresponding total heat of a lb. of dry saturated steam at P_2T_2, and ϕ_1 and ϕ_2 are the corresponding values of the entropies.

$$\text{The dryness fraction} = \frac{\phi_2 - \phi_w}{\phi_1 - \phi_w} \quad . \quad . \quad . \quad . \quad . \quad (22)$$

in which ϕ_w is the water entropy at T_2. In some steam tables the water entropy is not tabulated, and in this case the following equation may be used :

$$\text{Dryness Fraction} = \frac{H_2 - h_2 - T_2(\phi_2 - \phi_1)}{H_2 - h_2} \quad . \quad . \quad . \quad . \quad (23)$$

in which h_2 is the heat of the water at T_2. For all ordinary purposes h_2 may be taken as equal to $T_2 - 491{\cdot}6$.

The Mollier Diagram.

In the Mollier diagram the properties of a substance are plotted with reference to total heat and entropy as the ordinates, whereas the chief value of the temperature-entropy diagram is to represent heat changes graphically as an aid to calculation, the Mollier diagram does away in many cases with the necessity for any calculation at all. Mollier diagrams are now supplied with all steam tables, or may be purchased separately from Messrs. Edward Arnold & Co.

In fig. 2 is reproduced, in skeleton form, a part of a Mollier diagram for steam, sufficient to explain the manner of its employment.

The diagram it will be observed is divided into two fields by the saturated steam line SS. Above this line is the region of superheat, and below it the region of wet steam. The saturated steam line is crossed by lines of constant pressure sloping upwards from left to right. These lines are straight in the wet region but curve slightly upwards when the boundary line has been crossed. The constant pressure lines are crossed by more or less horizontal lines of constant temperature in the superheat region, and by lines of constant dryness fraction in the wet region.

Every point on the diagram represents steam in some definite condition. Steam at 500 lb. pressure, superheated to a total temperature of 750° F., for example, specified by the point A. We will assume that this steam is expanded adiabatically to a pressure of 1 lb. absolute. The process of expansion is represented by the vertical line AB, and the work done in this expansion, measured in heat units, is given by the length of the line. Reading from the scale of total heat at the side of the diagram, we see that the work done is 1390 — 916 = 474 B.Th.U. This is the same as the 'available heat' calculated by formula (15). The diagram also shows that after expansion to the lower pressure the dryness fraction of the steam is about 82 per cent.

Now let us suppose that the expansion, instead of being truly adiabatic, is carried out with frictional losses as in a turbine. The steam will be partially dried by the frictional heat, arriving

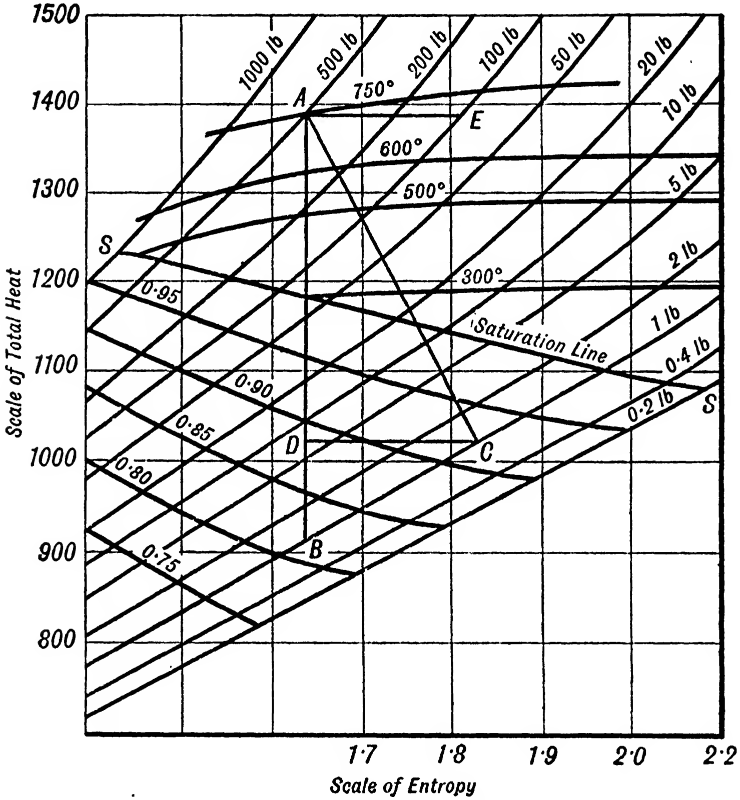


FIG. 2.

at the final pressure in some such state as that indicated by the point C, where the dryness fraction is about 92 per cent. The total heat at C being 1020 B.Th.U., the work done by the steam will be 1390 — 1020 or 370 B.Th.U., and the efficiency of the turbine will be $\frac{AD}{AB}$, or about 78 per cent. Conversely, knowing the efficiency beforehand, we can find the state C of the steam after expansion.

The throttling of steam to a lower pressure is shown by a horizontal line on the diagram, because during throttling the total heat remains unchanged. Thus if steam in the condition A were throttled down to a pressure of 100 lb., the process would be shown by the line AE, from which it is seen that the temperature would fall to about 710°, although the superheat would be higher.

SECTION XXV

FUELS: SOLID, LIQUID, AND GASEOUS

(pp. 1309–1373.)

SECTION XXV

FUELS: SOLID, LIQUID, AND GASEOUS.

Commercial fuels owe their heating value to two main combustible elements, carbon and hydrogen. Carbon is capable of forming two oxides, the monoxide CO formed when the oxygen supply is deficient and itself a gaseous fuel and the Dioxide CO_2; Sulphur, which is also contained in many fuels is itself combustible, but its effect on the total fuel value is relatively very small and it has the disadvantage that its oxide SO_2 has strongly acid properties and is thus liable to set up corrosion in metallic parts of the fuel consuming plant.

Provided that a fuel is completely burned, the carbon to CO_2, the hydrogen to steam, and the sulphur to SO_2, the amount of heat generated per unit weight of the fuel is perfectly definite and independent of the speed or method of combustion and is known as the calorific value, usually expressed in calories per gram or B.Th.U. per 1 lb. of the fuel.

(1 cal. per grm. = 1·8 B.Th.U. per 1 lb.).

If the steam produced from the combustion of the hydrogen is condensed, a further definite amount of heat is given out and there are thus two calorific values for any fuel containing hydrogen: the gross or higher cal. val. (H.C.V.) when the produced water is condensed and cooled and the nett or lower cal. val. (L.C.V.) when the produced water is not condensed.

The difference between the two values is

$$\text{H.C.V.} - \text{L.C.V.} = 10{\cdot}55\,(9H_2 + m)\ \text{B.Th.U. per 1 lb.}$$
$$= 5{\cdot}86\,(9H_2 + m)\ \text{Cal. per gm.}$$

where H_2 is the percentage of hydrogen and m the percentage moisture in the fuel.

In most commerical fuels there is generally some oxygen, and the carbon and hydrogen are combined in a more or less complicated manner. Such combinations may have resulted in the generation or absorbtion of heat and thus the actual calorific value of a fuel is generally different from the sum of the calorific values of its constituent elements.

Certain empirical formulæ have been evolved applicable to particular classes of fuel which give fairly satisfactory results *for those classes*. It is, however, safest to secure an experimental determination preferably by means of a bomb type of calorimeter of the C.V. of actual fuel to be used.

C.V. of Bituminous Coal is given roughly by the following—

Net calorific value $= [8{,}137\ C + 29{,}100\ (H_2 - \frac{O_2}{8}) + 2{,}500\ S - 600\ H_2O] \div 100$, where C, H_2, and O_2 are the percentages of carbon, hydrogen, and oxygen respectively present, and H_2O the percentage of hygroscopic water present.

When the ultimate analysis is not known and it is desired to obtain a rough estimate of the calorific power from the proximate analysis, this can be obtained by the formula :

$$\text{Calorific value} = 1\cdot 8\,(82C + aV)\ \text{B.Th.U.}$$

in which C = per cent. fixed carbon, V = per cent. volatile matter, and a is a constant computed from the following : $a_1 = 100V\,(C + V)$.

a_1	5	10	15	20	25	30	35	38	40
a	145	130	117	109	103	98	94	85	80

COMBUSTION.

When a fuel is raised to its ignition temperature in contact with oxygen, combustion commences. It consists essentially of the chemical combination of the combustible elements with the oxygen and is accompanied with the generation of considerable quantities of heat, raising the whole mass to incandescence. The chemical processes may be very complicated, but the final results, if combustion is complete, are represented by the following equations :

(*a*) If oxygen supply is insufficient

$$2C + O_2 = 2CO.$$

(*b*) If oxygen supply is adequate

$$2CO + O_2 = 2CO_2$$
$$C + O_2 = CO_2$$
$$2H_2 + O_2 = 2H_2O.$$

The amounts of heat generated and the volumes of the products of combustion are given in Table I., p. 1311.

When a fuel contains oxygen this is, of course, available, and a correspondingly decreased quantity of external oxygen is required. The amount of external oxygen necessary for the combustion of 1 lb. of a fuel whose ultimate composition is known would thus be

$$\text{Oxygen required} = 8H_2 + \frac{32}{12}C - O_2 \quad \text{or} \quad 2\cdot 66\left\{C + 3\left(H_2 - \frac{O_2}{8}\right)\right\}$$

where H_2, C and O_2 are respectively the proportions of hydrogen, carbon and oxygen present. As dry atmosphere air contains 21 per cent. by volume of oxygen (23·3 per cent. by weight), the weight of air required is :

$$11\cdot 6\left\{C + 3\left(H_2 - \frac{O_2}{8}\right)\right\}.$$

In accurate calculations it is necessary to make allowance for the moisture contained in the fuel and in the air used for combustion. The latter depends on its temperature and humidity.

The formula assumes that all the carbon is burned to CO_2, no CO being formed, and that combustible constituents other than those named are negligible.

Rich coals require more air than those low in calorific value, and, in fact, for coals such as those used in steam raising, over a wide range of qualities, the 'theoretical' air required may be taken with fair accuracy as proportional to the calorific value, and calculated by the formula

$$\text{Lb. air per lb. coal} = \frac{\text{higher calorific value} \times 7\cdot 2}{10{,}000},$$

the calorific value being in B.Th.U. per lb.

TABLE I.

Reaction.	Oxygen Required. Per Lb. Fuel.		Air Required. Per Lb. Fuel.		Cal. Value. B.Th.U. Gross.		Cal. Value. B.Th.U. Nett.		Products of Combustion. Vol.	Products of Combustion. Wt.	Products of Combustion. Added N_2 if in Air.	
	Vol. Cub. ft.	Wt. Lb.	Vol. Cub. ft.	Wt. Lb.	Per Lb.	Per Cub. ft.	Per Lb.	Per Cub. ft.	Cub. ft. per Lb. Fuel.	Per Lb. Fuel.	Cub. ft.	Lb.
$2C + O_2 = 2CO$ . . .	14·90	1·33	71·0	5·725	4,450	—	4,450	—	CO. 29·80	CO. 2·33	56·1	4·392
$C + O_2 = CO_2$ . . .	29·8	2·66	142·0	11·45	14,500	—	14,500	—	CO_2 4·90	CO_2 3·66	112·2	8·783
$2H_2 + O_2 = 2H_2O$ . . .	89·1	8·0	424·0	34·18	62,100	325·2	52,290	271·9	178·2	9·0	334·9	26·18
$2CO + O_2 = 2CO_2$ . . .	14·90	1·33	71·0	5·725	10,050	323·5*	10,050	323·5*	14·90	3·66	112·2†	8·783†

* Per cub. ft. CO. Other figures per lb. carbon.
† Assume preliminary oxidation in Air.

Excess Air.

Perfect regulation is never attained in practice, and in order to ensure that at least sufficient air is everywhere present to ensure completion of the combustion, it is necessary to provide an average excess over the quantity theoretically necessary. This excess mixes with and dilutes the products of combustion, lowering their average temperature and thus reducing the rate at which they will transmit heat, and when it finally leaves carries away heat corresponding to its temperature, which is lost. It is thus advisable to reduce the quantity of excess air to the lowest amount which will enable combustion to be just completed and avoid the production of smoke. Too little air will, however, lead to the presence of unburned fuel gases and a much greater loss of efficiency than a corresponding excess of air.

The percentage of excess air admitted can be calculated from the analysis of the flue gases. If the oxygen content of the flue gases is known the excess air is given approximately by the equation.

$$\text{Excess air} = \frac{5 \times \text{oxygen percentage}}{100 - 5 \times \text{oxygen percentage}} \times 100.$$

If, as is more usual, only the CO_2 content of the gases is known, the excess air can be calculated from the formula:

$$\text{Excess air (per cent.)} = \left(\frac{A}{CO_2 \text{ per cent. in flue gases}} - 1\right) \times 100.$$

A is the theoretical CO_2 content of the products of combustion of the fuel when no excess air is added, and its value depends upon the ratio of hydrogen to carbon in the fuel. It can easily be calculated if the ultimate analysis of the fuel is known. For average bituminous coal A may be taken as 18·9 per cent., for coke and anthracites as 20·0 per cent., and for petroleum fuel oils as approximately 14·0 per cent.

The amount of excess air necessary to secure complete and satisfactory combustion varies with the fuel used and with the type of furnace and grate. In well-regulated gaseous fuel furnaces very little excess air is used, with direct-fired, solid fuel furnaces the amount varies from 25 per cent. for coke- or anthracite-burning furnaces, 30 per cent. in the case of modern, well-designed mechanically stoked coal furnaces, to 50 per cent. in the case of older designs, and may rise to 100 per cent. or more in small hand-fired installations.

Quantity of Flue Gases.

In order to specify suitable fans for induced and forced draught, it is necessary to know the weight of flue gases and the weight of air actually used per lb. of fuel burned. From this the volume to be dealt with by the fans at the appropriate temperatures can be ascertained.

The quantity includes all excess air present, and can be calculated from first principles of the quantity and composition of the fuel as known.

The air used influences the percentage of CO_2 in the flue gases. For typical coals, the relation between air used and CO_2 may be expressed by the following approximate formula :—

$$\text{Lbs. air per lb. coal} = \frac{132 \times \text{calorific value of coal}}{CO_2 \text{ percentage} \times 10{,}000}.$$

The weight of flue gases per lb. coal may be taken as (lbs. air + 1).

Combustion of Hydrocarbon Fuels.

In the case of fuels containing hydrocarbon compounds many very complicated physical and chemical changes may take place before complete combustion is effected. Upon heating such hydrocarbons, *e.g.* those in bituminous coal, many commence to melt, or to distil or decompose at temperatures well below their ignition point. In such cases there is a primary evolution of combustible vapours and gases, and these when subjected to the radiant heat in a furnace undergo further decomposition. In an ordinary furnace it may not be possible to give such a regulated air supply and to mix it sufficiently intimately with the fuel as to give rapid and complete combustion. The combustion will then be a relatively slow and heterogeneous process taking place in long flames, in which fuel vapours and gases and air are flowing in more or less parallel streams which only mix gradually. It is thus necessary to provide combustion chambers sufficiently large to contain the whole of the flames produced.

In addition to these physical effects it must be remembered that the combustion of a hydrocarbon is a stage by stage process somewhat similar in nature to the low temperature oxidation by chemical means, *e.g.* ethane would be oxidised successively to ethyl alcohol, acetaldehyde and acetic acid, which latter would then suffer thermal decomposition to hydrogen and carbon monoxide before being finally oxidised to steam and carbon dioxide. Heat is given out at each step, resulting in a rise in temperature, and a stage is soon reached when the hydrocarbon molecules suffer thermal decomposition to simpler bodies and free carbon. If there is shortage of oxygen in intimate contact or if the flame is cooled below the ignition point, combustion is stopped and smoke produced. It will be noted also that the final products of the incomplete combustion contain considerable proportions of free hydrogen and carbonic oxide. It will be seen therefore how important it is to secure rapid and intimate mixture of the flame gases with sufficient oxygen if we are to secure complete and smokeless combustion and avoid the loss of fuel gases.

Internal Combustion Engines' Exhaust Gases.

The exhaust gases from internal combustion engines are subject to the same considerations as flue gases. They, however, cannot be regulated so as to produce a definite CO_2 percentage, as to obtain this result it would be necessary to throttle down the incoming air, thus loading the engine by increasing the difference in pressure between exhaust and inlet, in which case a

Theoretical Composition of Exhaust Gas with Varying Amounts of Excess Air when Fuel has the Composition: Carbon 88 %, Hydrogen 12 %.

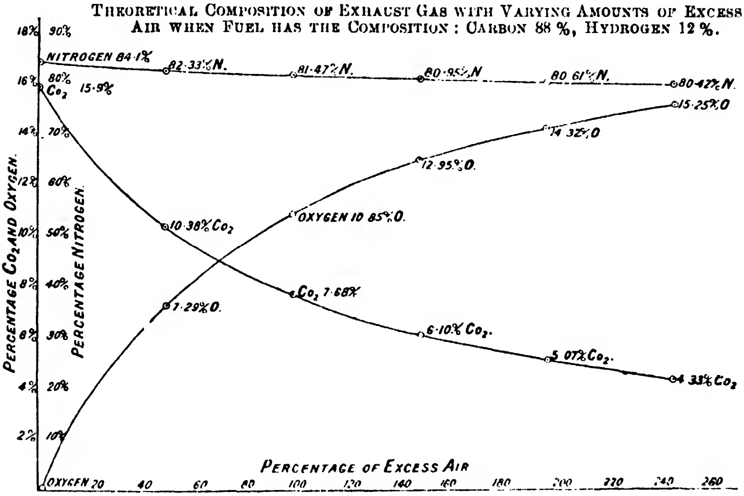


Fig. 1.

loss of efficiency would occur. The graphs in fig. 1 show the composition of the exhaust gases of an engine burning heavy petroleum (about 88 per cent. C and 12 per cent. H) with various quantities of excess air. Diesel oil engines running at full load generally take about 100 per cent. excess air; semi-Diesel engines a somewhat greater percentage excess air.

Heat Losses in Flue Gases.

The heat generated by the combustion of the fuel is communicated to the products of combustion which are thus raised to incandescence. The temperature of the flames thus depends upon the quantity and specific heat of these products and is thus affected greatly by the quantity of excess air admitted to secure complete combustion. As this temperature determines the 'temperature head' available for transmission of the heat, it will be seen what an important effect correct regulation of the air for combustion has upon the capacity of any given furnace plant, as well as upon its thermal efficiency.

When the combustion gases leave the furnace (at some temperature higher than that of the entering air) they carry away a certain proportion of the heat of the fuel and this is wasted.

Its amount may be calculated if we know the leaving temperature of the waste gases and the composition of both fuel and gases.

The result with a typical bituminous coal is shown below :—

Temp. of Exit Gases.		Percentage of CO_2 in Exit Gases.											
		4%	5%	6%	7%	8%	9%	10%	11%	12%	13%	14%	15%
		Corresponding Percentage of Excess Air.											
		372%	278%	215%	170%	136%	110%	89%	72%	57%	45%	35%	26%
F.°	C.°	Total Percentage of Fuel wasted.											
400	204	32·4	26·2	21·8	18·9	16·4	14·9	13·4	12·3	11·4	10·6	9·8	9·2
500	260	40·5	32·7	27·3	23·6	20·8	18·6	16·8	15·4	14·2	13·2	12·3	11·6
600	316	48·6	39·3	32·8	28·3	24·9	22·3	20·2	18·5	17·1	15·9	14·8	13·9
700	371	56·7	45·8	38·2	33·1	29·1	26·0	23·5	21·6	19·9	18·5	17·2	16·2
800	427	64·8	52·4	43·7	37·8	33·2	29·7	26·9	24·7	22·8	21·2	19·7	18·5
		For each 1% Combustible Gases add approximate.											
—	—	11·0	12·0	10·0	8·5	7·0	6·2	5·5	4·8	4·0	3·8	3·5	3·2

COMBUSTION COMPUTATION.

The total heat per pound of water vapour as found in air and in flue gases may be computed, within one B.Th.U., by the formula $H = 1{,}057 + 0{\cdot}46t$, where t is the Fahrenheit temperature. This formula frees the computer from all need for tables so far as computing moisture loss is concerned, while at the same time ensures a degree of accuracy sufficient for all practical purposes.

ANALYSIS OF FLUE GASES.

It is now standard practice to check the regulation of combustion by analysis of the flue gases. The simplest estimation is that of CO_2, but as the 'theoretical' percentage in ideal conditions varies with the hydrogen content of the fuel a knowledge of the O_2 content is really a better guide to the quantity of excess air admitted. For the purpose of checking waste of fuel a knowledge of the amount (if any) of combustible gases is very valuable, especially as there is a likelihood of these being produced when the excess air is being cut down to a minimum.

Most methods for the estimation of carbon dioxide are based on the absorption of this gas by potassium hydroxide (caustic potash) solution, the process consisting in measuring a volume of flue gas at a definite temperature, and then absorbing the CO_2 by means of a 35 per cent. solution of caustic potash, after which the volume of gas is again measured, the reduction in volume indicating the volume of CO_2 originally present. Many simple forms of chemical apparatus are available for this purpose.

It is frequently necessary to determine the oxygen and carbon monoxide contents as well as that of carbon dioxide. These may be absorbed in suitable reagents. In the usual form of Orsat apparatus a 100 ml. graduated burette is used and the reagents, caustic potash (for CO_2), alkaline pyrogallol (for O_2) and acid cuprous chloride (for CO) are contained in absorption vessels. The apparatus is completed by a stopcock manifold and a levelling bottle and is contained in a wooden case which is readily portable.

If other combustible gases such as hydrogen and methane are present they may be determined, together with the carbon monoxide by combustion with air over a heated platinum spiral and measurement of the contraction, CO_2 formed and O_2 used.

As the amounts of combustible gases are usually small it is necessary to use an apparatus of the Haldane type for analyses of this kind.

Sampling of Flue Gases.

In collecting samples of flue gases for analysis it is necessary to observe the following precautions.

(*a*) The composition of the actual flue gas varies with each adjustment of the furnace and alteration in combustion conditions. When 'spot' samples are taken it is, therefore, necessary to note the exact time and to co-relate this with any other records of furnace conditions. When continuous samples are being analysed by one of the many 'automatic' devices a certain length of time is taken by the passage of the gas from the flue and through the conducting tubes before the result of the analysis is recorded. This is known as the 'time lag' of the record, and in some cases the record may be upwards of 20 mins. late. The quantity of gas aspirated by these devices is small, consequently the tubing used to connect the instrument with the point at which the sample is taken should be as short and of as small bore as circumstances will permit. A very satisfactory device is to take a comparatively large bore tube (say 1½ in.) from the sampling point direct to the inlet of the induced draught fan or chimney base. The draught will induce a rapid flow of gas in this tube and the analysis instrument can then be connected to a suitable 'T' thereon by means of a short length of small tubing.

(*b*) Gas in flues, etc., tends to 'striate,' consequently the sampling tube must be arranged to take an average sample.

(*c*) Care must be taken to avoid contamination of the sample by infiltration of air, especially through the hole giving access to the flue for the sampling tube.

(*d*) Where hot gases are being sampled the tube used must be inert to the gases, *e.g.* at temperatures over, say, 800° F. an iron tube must not be used to convey gases containing water vapour and CO_2 as these gases will react with the metal of the tube and cause an alteration in the composition of the gas. Porcelain or fused silica tubes are preferable in such cases. If an iron tube must be used it should be water-jacketed.

(*e*) Liquids used to confine gases during storage or analysis should not be such as to dissolve constituents and so alter the volume or composition of the gas. Mercury is a very satisfactory fluid to use, but is heavy and relatively expensive. Owing to the solubility of the component gases, especially carbon dioxide, in water and the variations of these solubilities with the temperature and pressure, this liquid should not be used to collect the sample if accurate results are desired. If water be used, it will absorb carbon dioxide, for example, until it becomes saturated with this gas with respect to that gas mixture. This results in a loss of some of the carbon dioxide that was in the gas. Then if a gas cleaner in carbon dioxide is collected over the same water, the water will give up carbon dioxide and increase the carbon dioxide content of the gas. If, instead of water, a saturated solution of sodium chloride is used, these errors will be greatly reduced, as the gas is much less soluble in this solution.

Automatic CO_2 Recorders.

In all large installations it is customary to instal apparatus to analyse and record continuously the composition of the flue gases. The majority of these record only the percentage of CO_2, but instruments are available for recording also the percentage of oxygen or combustible gases. They are arranged either to—

(1) record the reduction in volume after absorption of the constituent from a regular series of measured samples; or

(2) to measure some physical property of the gas from which the desired result can be inferred.

SOLID FUELS.

The most important naturally occurring solid fuels are :—

Coal; Lignite; Peat; Wood.

These may be burned in the raw state, or may be subjected to destructive distillation. In the latter case solid, liquid, and gaseous products are evolved. The more important solid fuels which may be prepared by this treatment are coke (of two kinds, metallurgical and gas-works coke), lignite coke (of little value), peat charcoal, and wood charcoal.

Powdered coal is sometimes made into briquettes by the addition of a small percentage of coal-tar pitch or other binding material, and pressing in moulds.

COAL.

Coal consists of the mineralised remains of ancient vegetation and varies considerably in composition according to the particular portions of the paleolithic plants preserved, the age of the deposit, its depth, and the properties of the surrounding strata. The younger formations are more nearly allied to cellulose, as the lignites and brown coals, and contain a relatively large proportion of oxygen. The matter which distils off on heating 'to redness' is known as the volatile matter, and may amount to as little as 7 per cent. or as much as 40 per cent. of the coal according to its nature. (See Table, p. 1316.)

It will be realised that bituminous coal is a very complicated mixture of heavy hydrocarbons, and much work has been done in attempts to determine its constitution. Prof. Phillips Bedson of Armstrong College, Newcastle, uses the selective action of various solvents of which the most useful appear to be chloroform and pyridene, which dissolve out certain constituents.

Workers at Owens College, Manchester, have attempted to separate mineralogical constituents, and recognise four main kinds, viz. Fusain, Vitrain, Clarain, and Durain, but it is doubtful if these constituents are definite compounds, and their composition as well as their proportions vary from one seam to another.

The preparation of microscope sections of coal and the examination thereof by transmitted light has been found to give valuable information as to the nature of the coal and its suitability for particular processes.

The composition of coal from various fields varies considerably according to the conditions to which the coal has been subjected during its formation. Its variation in composition can be seen from the following table of coal analyses :—

Ultimate Composition of Coal (*Brame and King*).*

Type of Coal.	Carbon.	Hydrogen.	Oxygen and Nitrogen.	Fixed Carbon.	Volatile Matter.
Lignites and brown coals	69·5	5·5	25·0	52·0	48·0
Splint coal (Fife)	82·0	5·0	12·8	61·0	39·0
Gas coal (Durham)	85·0	5·5	8·2	66·0	34·0
Coking coal	87·3	5·05	6·9	73·5	26·5
Smokeless steam (Welsh)	91·3	4·05	3·9	85·5	14·5
Anthracite (Scotch)	91·1	3·5	4·65	88 5	11·5
Anthracite (Welsh)	91·0	3·9	4·28	93·0	7·0

Coal possesses a sp. gr. of from 1·2 to 1·5 (water = 1) mainly depending on its composition. It also varies considerably in hardness and texture. The combustible portion of coal usually consists of from 73 to 93 per cent. carbon, 3 to 6 per cent. hydrogen, 2 to 20 per cent. oxygen, 1 to 2·5 per cent. nitrogen, and 0·5 to 5 per cent. sulphur. When completely burned coals yield from 1 to 15 per cent. ash. The moisture content varies within very wide limits. The heat value of coal is generally from 12,000 to 16,000 B.Th.U. per lb. gross. A comparison of the properties of coal with those of other solid fuels may be seen in the following table :—

Calorific Value of Various Fuels.
(Dry ash-free basis.)

Substance.	Calorific Value	
	Calories/g.	B.Th.U. per Lb.
Wood. Ash	4,740	8,530
,, Beech	4,800	8,640
,, Elm	4,750	8,550
,, Oak	4,640	8,350
,, Fir	5,050	9,090
,, Pine	5,105	9,190
Peat	5,555	10,000
Peat Coke	8,480	15,260
Lignite—brown—Devon	6,220	11,200
,, ,, Australia	5,720	10,300
,, ,, Germany	6,280	11,300
Lignite—black—New Zealand	7,010	12,620
,, ,, Burma	6,220	11,200
,, ,, Canada	7,250	13,050
Coal—Scotland—Slightly coking	8,065	14,520
,, Northumberland—Household	8,095	14,570
,, Durham—Coking	8,600	15,480
,, Lancashire—Coking, gas	8,430	15,170
,, Yorkshire—Coking, gas	8,600	15,480
,, Notts and Derby—Household, steam	8,300	14,940
,, S. Wales—Coking, gas	8,210	14,780
,, ,, Steam	8,775	15,800
,, ,, Anthracite	8,665	15,600
,, E. Kent—Coking	8,635	15,540

* 'Fuel: Solid, Liquid, and Gaseous,' p. 62.

Sampling and Analysis of Coal and Coke.

The object of sampling is to obtain a small sample of the fuel for analysis which will be truly representative of the whole consignment. The method of sampling to be employed will depend upon the circumstances prevailing, *e.g.* sampling from wagons, conveyors, chutes, heaps, or barrows. In deciding the weight of total sample to be taken it must be remembered that the particle size and the degree of heterogeneity are the factors which determine the weight of sample to be taken *not* the gross weight of the consignment to be sampled.

With coal the chief variant is the ash content while in coke sampling the variation of the moisture content is the most important factor.

The British Standards Institution has published methods for the sampling of coal and coke to suit a variety of industrial requirements. In 1942 two comprehensive specifications were issued covering the whole field: viz. B.S. No. 1017 'Sampling of Coal and Coke'—covering the sampling down to the stage when the laboratory sample is prepared and B.S. No. 1016 'Analysis and Testing of Coal and Coke' detailing the treatment of the laboratory sample in order to prepare the samples for analysis. This latter specification also contains the standard methods for analysis of the fuel (see below). In general it is particularly important to keep the gross sample free from contamination and from loss of moisture during sampling. In the preparation of the sample for analysis provision must be made for taking special samples for moisture content and to use mechanical means for reduction in particle size to avoid oxidation due to exposure.

ANALYSIS.

Analysis of coal and coke may be either 'proximate' or 'ultimate.' In the former the following determinations are made:—

Moisture.—Determined by loss in weight of a sample on heating at 108° C. to constant weight. Special precautions are necessary to prevent oxidation.

Volatile Matter.—Determined as the loss in weight when a sample is heated for 7 minutes at 925° C. under standardised conditions. This is an empirical test and the conditions laid down must be adhered to strictly.

Ash.—Determined as the residue remaining after the sample has been heated to 800° C. to constant weight in an oxidising atmosphere.

Fixed Carbon.—This is the undetermined matter, *i.e.* 100 − [Moisture + Ash + Volatile Matter].

In ultimate analysis the elementary constituents, carbon, hydrogen, nitrogen and sulphur are determined. Oxygen is obtained by difference.

The results of the analyses may be reported as percentages of air-dried, dry, or dry ash-free material.

In view of the fact that some of the methods are of an empirical character it is highly desirable that the standard methods as laid down by the British Standards Institution (see above), should be followed. Close observance of details is necessary if comparable and reproducible results are to be obtained.

The Commercial Selection of Coals.

In the preparation of coal for the market the run of mine coal is first screened to separate the large (over 1 in. or so) from the smalls, and the former is then passed over a picking belt where visible impurities—stone, shale, pyrites, etc.—are picked out by hand. The large coal may be further screened for certain markets into nuts, cobbles and large-screened coals. The small coal may either be sold without further preparation or further screened into various sizes of smalls and duff, and where reduction in the ash content of smalls is considered desirable these may be subjected to washing or dry cleaning processes by which the particles of high specific gravity are separated from the lighter coal. When purchasing coals, therefore, the sizes of screen to be used and the washing or other purifying processes used in its preparation should be known.

The commercial value of a coal for any specific purpose and for direct combustion purposes will be dependent upon the following factors:—

(i) The lower or net calorific value.

(ii) The amount of volatile matter and nature of coke.

(iii) The amount and nature of the ash content.

(iv) Its physical suitability for the particular conditions under which it is to be burned.

(v) The price per ton delivered on site.

(vi) Local cost of labour for handling, firing, and removal of ashes.

The net calorific value (i) calls for both a test for calorific value and also either an ultimate analysis or calculation in order to arrive at the hydrogen content of the fuel. The latter calculation or estimate is made from the percentage of volatile matter in the coal as disclosed by proximate analysis.

The net commercial calorific value will vary with the conditions obtaining in individual installations, and will depend upon the final temperature of the gases leaving the boilers or furnaces, together with the total hydrogen in the fuel as fired.

The total hydrogen will be the sum of the hydrogen combined in the fuel and that forming one-ninth of the moisture. Lower commercial calorific value for any specific plant will be

$$\mathrm{HCV} - \left\{ 9\mathrm{H}\left([212 - t] + 970\cdot4 + 0\cdot47\ [\mathrm{T} - 212] \right) \right\},$$

where,

HCV = higher or calorimeter value of fuel in B.Th.U. per lb. as fired;
H = hydrogen fraction of the fuel as fired;
t = temperature of the fuel in °F., as fired;
T = final temperature of waste gases.

In the absence of direct ultimate analysis of the coal, the percentage of hydrogen in the fuel may be closely estimated by the following formula (Diederichs'):—

$$\mathrm{H} = \mathrm{V}\left(\frac{7\cdot35}{\mathrm{V} + 10}\right) - 0\cdot013,$$

where,

H = percentage of hydrogen in the combustible; V = percentage of volatile matter in the combustible.

Combustible = coal, less sum of moisture and ash.

(ii) The amount of volatile matter and nature of the coke are found simultaneously in the proximate analysis. This test will give the correct classification of the coal as to its qualifying characteristics for special uses. The ratio of volatile matter to fixed carbon is of the greatest importance in deciding the suitability or otherwise of a coal for use on any particular plant; all other things being equal, the fuel with the lowest volatile content which can be efficiently consumed in the specific furnace conditions will be the most economical in use. This can only be definitely determined by the careful working test of a sample delivery of the fuel, and careful note made of the percentage of CO_2 in the waste gases, together with the final temperature of the gases and the percentage of combustible on the ash heap. The following general classification of coals will be of assistance in selecting suitable fuels for given purposes from the commercial viewpoint of economy in use.

Class of Coal.	Per Cent. Volatile Matter in Combustible.	Economical Application.
Short flame non-caking, semi-bituminous, or Welsh steam coals	12–20	Steam raising with strong draught.
Short flame caking coals	20–25	As above and coke manufacture.
Medium flame, moderately caking coals	25–32	Steam raising, domestic grates.
Long flame caking coals	32–40	Gas manufacture.
Long flame non-caking North Country coals	over 40	Furnaces, gas producers.

Burning a coal in a furnace which has insufficient draught for complete combustion will result in excessive waste of unburned carbon as coke in the ashes. Burning a coal of an unnecessarily high volatile content will produce an unnecessarily high loss in the waste gases, due to the hydrogen content, smoke will be difficult to avoid.

The amount of ash and moisture will not affect the classification given above.

(iii) The amount of the ash in the coal will not only reduce its commercial value by the reduction in actual calorific value, but also by its physical effect will reduce the value at a greater rate than that of the percentage of ash. For ordinary conditions of firing and draught, without steam jets, the rate of fall in practical value with the ash content will gradually increase until at 40 per cent. ash the fuel will be unburnable and valueless. This is for average ash.

The nature of the ash is chiefly affected by the percentage of iron in its composition, the fluxing effect of ferric oxide in the ferruginous ash coals causing the formation of liquid slag at comparatively low temperatures, which closes up the fire-bar spaces, assists in the fusion of firebars, and entraps appreciable quantities of unburned combustible. Ferruginous ash is distinguished by its red colour, the percentage of iron being roughly indicated by the depth of the coloration of the ash in the crucible after proximate analysis for ash.

(iv) The physical suitability of the coal for particular conditions depends upon the construction of the firegrates, type of mechanical stoker, and the amount of draught available. Under practically every condition of use mixed coals as regards size should be avoided. For hand-fired boilers and furnaces screened cobbles or nuts should always be selected. Large steam coal is to be avoided; if fired as received, in lumps, efficient combustion is impossible, while breaking down results in uneven fires and waste due to the formation of a quantity of slack in the process.

Fine slacks are unsuited for hand-firing on plain grates, but can be burned very efficiently on special forced-draught grates fitted with steam jets and closely spaced firebars. High ash coals may be more efficiently burned on these grates than with plain natural or induced draught.

(v) The price per ton of coal, delivered, together with the cost of labour in handling and incidental work, is the last test of the value of the coal to be applied after all the technical deductions already referred to have been made, and is to be utilised to arrive at the net number of B.Th.U. per unit of cost under practical conditions of work.

Wherever possible, the following procedure should be adopted in selecting the most economically suitable coals for contract purposes:—

A sample delivery should be ordered of sufficient size to allow of observation under practical working conditions for a long enough period to ascertain the maximum hourly evaporation under forced and economical conditions of work, probable ash-heap loss, chimney loss, etc. This cannot be carried out without the proper equipment of testing instruments. This working observation will allow of a close estimate of labour and power costs. A low-grade slack of about 10,000 B.Th.U. per lb. will probably call for a power and labour cost quite double that involved in the use of good quality washed nuts of 14,000 B.Th.U. per lb. This shows the uselessness of evaluation on the calorific value alone. Standing losses on the plant, such as conduction and convection, etc., vary directly as the number of boilers, etc., used.

Final evaluation should be made as follows:—

Gross or higher calorific value of the coal, air-dried, then deduct the following:—

Hydrogen loss under practical conditions of waste gas temperature;

Heat loss in waste gases due to excess air as per CO_2 and temperature readings;

Loss due to unburned carbon in ash-heap per lb. of fuel used.

The final figure thus obtained gives the practical working value of the fuel in B.Th.U. per lb.

To the price of the coal must be added the following costs:—

Cost of attendance for firing per ton of coal used;

" power, such as stoker, fan, conveyor, economiser drive, etc., per ton of coal used;

" ash removal per ton of coal used.

From the data thus obtained the ultimate cost of, say, 100,000 B.Th.U. is obtainable for the actual conditions existing with the particular plant upon which the coal is to be used.

The system which has been used with success is to reduce the value of each coal to a figure of merit consisting of its comparative value with pure carbon as a standard, the latter being taken as '100.' The average final temperature of the gases is ascertained in the usual manner. The chimney loss for pure carbon burned perfectly without excess air is then calculated for the conditions of waste gas temperature obtaining on the plant. Taking the calorific value of pure carbon as 14,544 B.Th.U. per lb., the '100' figure of merit for this plant will be 14,544 minus the gas loss per lb. of carbon. This represents theoretical perfection with the plant. The actual results obtained for the coal under examination are expressed as a percentage of the pure carbon theoretical figure, and this is taken as the relative figure of merit for the fuel.

(*C. F. Wade.*)

Economic Effects of Impurities in Coal.

Each per cent. of sulphur reduces the boiler capacity by about 1·5 per cent. In experiments made in America with a chain grate stoker, using Illinois coal mixed with various percentages of ash, the efficiency dropped gradually as the ash increased up to 35 per cent., after which the efficiency dropped to zero rapidly with a fuel containing 40 per cent. of ash.

Temperature of Decomposition of Coal.

The temperatures at which coals begin to decompose are of considerable importance, both for technical purposes and as an aid to the study of the constitution of coal. There is little published information of a precise nature on this point, and it is frequently assumed that coal may be heated to 250° C., for purposes such as drying, without causing any modification in the coal substance. A general investigation of the behaviour of coal on heating is being carried out at H.M. Fuel Research Station, and a report issued by H.M. Stationery Office deals with the results of experiments to determine the temperatures at which products of decomposition of various types of coal are first given off when the coal is heated. It is shown that when bituminous coals are heated, oil fumes may appear at temperatures as low as 215° C. The investigations are continuing and further reports will be issued in due course.

Storage of Coal.

In storage of large quantities of bituminous coal care has to be taken to avoid spontaneous fires. Coal slowly absorbs oxygen from the air (really a very slow form of combustion) with consequent production of heat. The rate at which such reaction occurs is accelerated greatly by rise in temperature, consequently the coal heaps should be arranged to allow of the escape of the heat generated so as to avoid a dangerous rise in temperature which would rapidly be followed by a spontaneous fire. With most British coals this can be effected, provided the heap is not more than 12 ft. 6 ins. deep if flat, or correspondingly higher if peaked.

All coal heaps should be watched for signs of heating. This can conveniently be done by inserting in the heap at intervals small tubes down which thermometers may be dropped and readings taken and logged at regular intervals.

The absorption of oxygen is accompanied by slow deterioration in the quality of the coal. The calorific value is reduced, but still more the free burning qualities of the fuel, and the use of coal which has been stored for long periods will be accompanied by a reduction in the capacity of the furnace.

Loss of Thermal Value of Coal by Storage.

An average of 741 tests, in America, of a particular coal showed, after three years in the open air, a reduction of from 35 to 28·3 per cent. of volatile combustible matter, indicating a thermal value loss of 20 per cent.

Coal in storage should be inspected regularly. If the temperature reaches 150° F. the pile should be carefully watched, and if the temperature rises to 175° or 180° F. the coal should be removed as promptly as possible. The coal should be thoroughly cooled before being replaced in storage.

Stowage Space of Coal.

The stowage space occupied by coal is 40 to 42 cubic feet for one ton of Welsh steam coal. 45 cubic feet for one ton of North country coal, and 45 to 50 cubic feet per ton of Scotch coal.

EVAPORATIVE POWER OF FUELS.

It used to be the custom to express the heating power of fuels as 'Evaporative Power,' *i.e.* the number of pounds of steam which could be made per 1 lb. of the fuel. In order to obtain comparative figures, the results were reduced to 'from and at 212° F.', *i.e.* steam at atmospheric pressure made from feed water at boiling point. Each 1 lb. of steam made under these conditions requires 967 B.Th.U.; consequently, if the calorific value of the fuel in B.Th.U. per 1 lb. is divided by 967, we get the theoretical evaporative power 'from and at 212° F.' Boiler plants never work under these conditions, and their efficiency varies from 50 to 80 per cent. according to care in design and working, consequently such evaporative figures are never realised. The reduction of evaporative factors usually obtained to the 'from and at 212° F.' basis can be effected by using the factors in the following table and such reduction will allow of ready comparison of the merits of different plants.

Evaporation Factors.

To reduce results to 'from and at 212° F.'

Temp. of Feed Water in F.°	Boiler Pressure in Pounds per Square Inch.												
	60	80	100	120	140	160	180	200	220	240	260	280	300
40	1·209	1·214	1·218	1·222	1·226	1·229	1·232	1·234	1·237	1·239	1·241	1·243	1·245
50	1·198	1·204	1·208	1·212	1·215	1·218	1·221	1·224	1·226	1·229	1·231	1·233	1·235
60	1·188	1·193	1·198	1·202	1·205	1·208	1·211	1·214	1·216	1·218	1·220	1·222	1·224
70	1·178	1·183	1·187	1·191	1·194	1·197	1·200	1·203	1·206	1·208	1·210	1·212	1·214
80	1·167	1·173	1·177	1·181	1·184	1·187	1·190	1·193	1·195	1·198	1·200	1·202	1·204
90	1·157	1·162	1·167	1·170	1·174	1·177	1·180	1·183	1·185	1·187	1·189	1·191	1·193
100	1·147	1·152	1·156	1·160	1·164	1·167	1·170	1·172	1·175	1·177	1·179	1·181	1·183
110	1·136	1·142	1·146	1·150	1·153	1·156	1·159	1·162	1·164	1·167	1·169	1·171	1·173
120	1·126	1·131	1·136	1·140	1·143	1·146	1·149	1·151	1·154	1·156	1·158	1·160	1·162
130	1·116	1·121	1·125	1·129	1·132	1·136	1·138	1·141	1·144	1·146	1·148	1·150	1·152
140	1·105	1·110	1·115	1·119	1·122	1·125	1·128	1·131	1·133	1·135	1·137	1·139	1·141
150	1·095	1·100	1·104	1·108	1·111	1·115	1·118	1·120	1·123	1·125	1·127	1·129	1·131
160	1·084	1·090	1·094	1·098	1·101	1·104	1·107	1·110	1·112	1·115	1·117	1·119	1·121
170	1·074	1·079	1·083	1·087	1·091	1·094	1·097	1·099	1·102	1·104	1·106	1·108	1·110
180	1·063	1·069	1·073	1·077	1·080	1·083	1·086	1·089	1·091	1·094	1·096	1·098	1·100
190	1·053	1·058	1·062	1·066	1·070	1·073	1·076	1·078	1·081	1·083	1·085	1·087	1·089
200	1·042	1·048	1·052	1·056	1·059	1·063	1·065	1·068	1·071	1·073	1·075	1·077	1·079
210	1·032	1·037	1·042	1·046	1·049	1·052	1·055	1·057	1·060	1·062	1·064	1·066	1·068

For every 10° superheat, 0·0055 is to be added to the above factors.
See also table on p. 1327.

It is, however, now becoming increasingly common to express the results of boiler tests in terms of overall thermal efficiency, which is a more scientific method.

Coke and Coking.

At the present time nearly all metallurgical coke is made in by-product ovens, where the chief product is still coke, but where other valuable products are recovered. These are : (1) ammonia, marketed mainly as sulphate of ammonia for use as a fertiliser; (2) tar, which is distilled or refined at the coke works, or sold to tar distillers working privately or on a co-operative basis; (3) benzole and its homologues, which either alone or mixed with petrol is sold as motor fuel. (4) (*a*) coke-oven gas, which is essentially the same gas as that produced from gasworks' retorts ; or (*b*) waste heat, which serves for steam generation. The type of oven used determines whether gas or waste heat is the by-product produced.

The largest waste which occurs in almost all coke works is in the sensible heat of the coke when discharged from the ovens, and from which it is possible to recover heat equivalent to 0·4 lb. of steam per lb. of coke. The extra plant necessary for the recovery of this heat is costly and not likely to be generally employed unless it can be simplified and cheapened. Sometimes, owing to the unfavourable situation of a coking plant, the whole of the surplus gas cannot be disposed of with advantage.

In 1947, 14,597,000 tons of coke were made from 19,820,000 tons of coal carbonised. Yields of products per ton of coal are (according to circumstances) from 12 to 16 cwts. of coke : 20 to 26 lb. (and occasionally up to 36 lbs.) of sulphate of ammonia ; 2 to 3½ gals. of 65 per cent. crude benzole ; 85 to 110 lb. of tar ; 10,000 to 13,000 cub. ft. of gas, a large proportion of which is available for external use either as gas or as its equivalent in waste heat.

Coking Coals.

In contrast to gasworks practice, great attention is paid to the condition and properties of the coal used. Coking coals are bituminous in type and pass when heated through a plastic stage between 350° and 450° C., and must possess sufficient coking power to leave a hard and compact residue on further heating.

Usually more than 8 per cent. of oxygen on an ash- and moisture-free basis denotes insufficient coking power. Oxidation or heating destroys coking power, and thus storage of coal, by encouraging this action, often tends to depreciate coking value. The extent of depreciation varies greatly with different classes of coking coals, some such as certain Derbyshire coals becoming practically non-coking after only two or three days in the coke-ovens' service bunkers, whereas certain types of Durham coal have been kept in stock for five or six years without appreciable loss of coking power. Other properties which distinguish coals most suitable for coking are :—

(*a*) Low content of ash and sulphur.
(*b*) Volatile matter between 20 and 30 per cent., since high volatile matter causes the formation of a 'spongy' coke unless special carbonising conditions are adopted.
(*c*) Negligible amount of alkalies, as these have an injurious effect on oven linings.
(*d*) A uniform moisture content not exceeding 8 to 9 per cent. as extra heat above that required for carbonising must be applied to evaporate moisture from the ovens. Thus carbonisation is retarded and the steam produced throws extra work on the gas cooling plant, as it has to be subsequently condensed.

In most coke works the coal is crushed more or less finely before being charged into the ovens and only in a few cases are run-of-mine coking smalls used directly. The effect of crushing is to disseminate shale impurities through the charge, so that their effect as sources of cracks in the coke is reduced. The average size and strength of the coke made is increased, with a corresponding reduction in the production of small coke and breeze. Crushing, however, decreases the packing density of the charge, and so reduces oven capacity. A density variation of from 45 lbs. to 52 lbs. per cub. ft. of coal is typical of varying the degree of crushing within the limits possible in the average coke works crushing plant.

Charging the coal in a loose condition through charge-holes in the oven tops is the usual method of loading. The once common practice of using stamped charges of finely ground moist coal in the form of a solid cake is now confined to a few isolated cases where an exceptionally feebly coking or highly volatile coal is used. Under ordinary conditions this method has nothing to recommend it.

Cleaning processes are used in many coke works to reduce ash and sulphur before crushing the coal. Wet washing plants are common, but require large drainage bunkers to reduce the water taken up by the coal to workable amount ; they also necessitate screening of the coal at some part of the process to secure efficient washing. In some cases dry cleaning plants have been used. In these, coal is separated from its impurities by passing it over oscillating tables through which a strong current of air is blown. This method eliminates drainage bunkers.

Ovens.

'Beehive' ovens, formerly largely used, are now almost superseded by the 'By-Product' type ; in the latter type two systems of oven heating are in use :—

(*a*) The waste-heat system, in which upwards of 80 per cent. of the total gas evolved is burnt in the wall flues with air at atmospheric temperature or at the most only slightly preheated by passage under the hot ovens to the flues. Burnt gases leave the flues at a temperature of about 1100° C., and are used principally for steam-raising, and as in beehive practice, up to 1 lb. of steam is obtained per lb. of coal carbonised. This type of oven is simple in construction and operation.

(*b*) The regenerator system, in which chambers under the ovens, filled with chequer brickwork, as in Siemen's open-hearth steel furnaces, are employed to transfer heat from the burnt gas leaving the flues to air entering them. In this way the waste gas is cooled to 250° C. before it leaves the regenerator chambers, and the air entering the flues is preheated to 900° C. approximately. Only 35 to 50 per cent. of the total gas is required for heating the ovens, the remainder being surplus and available for other uses. The thermal efficiency of the two systems is about the same.

The majority of by-product ovens which were constructed before the war of 1914–1918 had an average length of 35 ft., a width of 20 ins. to 21 ins., and a height varying from 6 ft. to 7½ ft. Their capacity varies from 6 to 9 tons of coal. Working temperatures of the flues are from 1100° to 1200° C., being limited by the fusion point of the semi-silica (75–85 per cent. SiO_2) refractories used for oven linings and flues. On dry coal a carbonising time of 26 to 32 hours is attained, which gives a daily throughput of 5½ to 7½ tons per oven. Heating flues are arranged either horizontally or vertically in the oven walls. In waste-heat batteries combustion occurs continuously in each flue, and the main difference between the various constructions is in the spacing of the flues to obtain uniform heating, and in the methods adopted to regulate air and gas to each flue. In regenerative ovens, the periodic reversal of gas and air flow through the regenerator chambers causes a periodic change in the conditions in the combustion flues. In ovens with horizontal flues this consists of a reversal of the path of travel of the burning gases in the flues with a change in the points at which air is admitted, but necessitates no alteration in the points at which gas is admitted. In vertical flue ovens, which predominate in number, only half the flues are used at a time for combustion, the other half being used for carrying away the burnt gas. Flue designs in existing ovens vary in the arrangement used to eliminate uneven heating from this cause. They differ mainly in the way in which flues are grouped and cross connected to form separate reversal units.

Quenching coke by water in front of the ovens was usual on old batteries either on an inclined bench running the whole length of the battery or in some form of car movable from oven to oven. Approximately 1 ton of water was used per ton of coke quenched, and the coke was very slowly pushed from the ovens to allow sufficient time for quenching. Success of quenching by this method depends largely on the skill of the workmen.

A typical modern coke-oven battery consists of 40 to 60 ovens, each 43 to 44 ft. long, 13 to 14 ft. high, and 16 to 18 ins. wide, heated regeneratively so as to have a maximum yield of surplus gas. In order to withstand the higher flue temperatures used (1200°–1400° C.), flues and oven linings are constructed in 95 per cent. silica material which will withstand temperatures up to 1650° C. without distortion. In some types the heating flues are built vertically and arranged on the 'hairpin' system, in which reversal of air and gas takes place between pairs of adjacent flues. Alternate flues are fed with air and gas, so that combustion takes place up one flue and down the next, at the bottom of which burnt gases are withdrawn. Another method of flue arrangement is by cross connection of groups of flues in opposite walls of the ovens through flues in the oven tops, so that reversal of combustion takes place from one side of each oven to the other. Regenerator chambers filled with firebrick chequerwork, built under each oven, are subdivided, so that each set of flues has its own regenerator system. The use of silica refractories increases very considerably the safety factor when working at high temperatures, with the additional advantage of better thermal conductivity through the oven walls. They are also much more resistant to the action of alkalies than other coke-oven refractories.

Recently erected ovens have a capacity of 14 to 17 tons of coal, which with a carbonising time of 16 hours gives a daily throughput of about 20 to 26 tons per oven.

Subsidiary, but necessary improvements incorporated in modern ovens are :—

(1) High-speed pushing machines by which a 44-ft. oven is discharged in 30 secs., or approximately in one-tenth of the time required to push the shorter pre-war ovens.

(2) Electrically driven coal cars, in which the volume of the coal charged can be accurately gauged.

(3) Self-sealing and plug doors, which are removed and replaced mechanically and which require at most an extremely small amount of luting to render them gas-tight.

(4) Remote coke quenching, by which the coke pushed from an oven at 900° to 1000° C. is discharged into a steel car which carries it rapidly from the front of the battery to a water-tower some distance from the ovens, where a measured volume of water is sprayed on to it. After draining the coke is taken to a coke bench and discharged to steam and cool off before being taken by high-speed rubber belts to the screening plant. This system of quenching makes work in front of the oven tolerable under high-speed conditions and prevents damage to the hot oven walls by contact with steam and water, such as is possible from quenching in front of the ovens. Rather more water is required than with other systems (from 1¾ to 2 tons per ton of coke), but quenching is more uniform and coke moisture more constant.

Narrow ovens are better suited for the treatment of inferior coking coals. In extreme cases of feebly coking or highly volatile coals, 12-in. to 14-in. ovens are used, carbonising in 12 hours and producing a satisfactory coke, but with the average range of coking coals a 16-in. or 18-in. oven, carbonising in 18 hours, seems to give most satisfactory results. The amount of heat supplied to the ovens *via* the combustion flues is usually from 850 to 1100 B.Th.U.'s per lb. of coal carbonised, but varies with the speed and temperature of carbonising and with the efficiency of the system. Under average conditions 60 to 65 per cent. of surplus gas is produced. For maximum yield of gas, compound regenerative ovens are built in which the flues may be heated by combustion of preheated producer, water, or blast-furnace gas, the whole of the coal-gas made being available for other use.

Coke.

This varies in characteristics according to the coal used and carbonising procedure. Size is dependent on the thickness of the charge carbonised. In beehive practice the largest pieces are 30 ins. to 36 ins. long, approximating to the depth of coal in the ovens. With by-product coke the largest pieces are slightly less than half the oven width in length. Typical characteristics are :—

	True Sp. Gr.	Apparent Sp. Gr.	True Porosity. %	Apparent Porosity. %
Beehive coke .	1·88–1·98	0·87–0·90	52–56	45–50
By-product coke .	1·90–2·00	0·95–1·10	42–52	38–45

Crushing strength varies from 1,000 to 3,500 lbs. per sq. in., and volatile matter from 0·5 to 3 or 4 per cent. The volatile matter is greatest in high-speed practice where no over-coking takes place. Most of the coke made is used in metallurgical furnaces, and from 85 to 95 per cent. of the product of a coke works is in pieces large enough for such purposes, the remainder being in the form of coke nuts down to ⅝ in. size and coke breeze below this size.

For smelting use a hard dense coke is required strong enough to resist the severe crushing and abrasion it receives. For many years after the introduction of the by-product oven some operators still preferred bee-hive coke for metallurgical uses and there was some prejudice against by-product coke. Properly sized by-product coke, however, gives equally good results and nowadays only about 0·5 per cent. of the coke produced in coke oven is made in bee-hive ovens. For foundry purposes coke of large size is preferred, while for blast-furnace use smaller evenly-sized coke screened to remove sizes below about 1½ in. is preferred.

Correct sizing and efficient screening is established as a means of obtaining greater efficiency in its use as a fuel. Structure and porosity have a definite bearing on the combustibility, while volatile content governs the ignition temperature, so that there are great possibilities in adapting oven conditions for various types of coke production. Screened coke nuts are to some extent used in domestic applications. A small uniform coke with low ash, up to 6 per cent. volatile matter and an open but strong structure is desirable, so that satisfactory ignition and combustion results under the feeble draught conditions of open ranges and fire grates. A new and increasing use of coke nuts is in the synthetic chemical industry where, as producer fuel, they supply the hydrogen and carbon monoxide required for ammonia and methanol syntheses. Other uses are in lime-kilns and for steam-raising. Coke breeze under ⅝ in. size is usually consumed under boilers by means of mechanical stokers, chiefly to supply steam at the coke works and adjacent plants.

4½ to 5 lbs. of steam can be raised per lb. of coke breeze.

By-Product Recovery.

This follows the main lines of gasworks recovery, with the following exceptions. Individual ascension pipes are not liquid sealed in the collecting main, but are shut off by means of valves. The gas-collecting main is kept clear by a flush of tar or ammoniacal liquor. In some cases, cold liquor is sprayed into the collecting main at the junction with each ascension pipe and gives additional gas cooling.

In coke-oven practice ammonia recovery has reached a higher stage of efficiency than in gasworks practice, and benzole is always recovered. Actual yields depend on the coal, and also on the extent of degradation of volatile products in the ovens caused by contact with the hot walls and coke. Ovens are designed with a view to keeping the tops reasonably cool and to allow the rapid removal of gaseous products in order to reduce the time of contact with heated surfaces.

	Average Gas Temperatures.
Gas leaving ovens	600°–700° C.
„ entering collecting main . . .	300°–400° C.
„ leaving collecting main	150°–200° C.
„ entering by-product coolers . . .	70°– 80° C.

Tar.

This is removed partly by condensation from the gas in various parts of the coolers, and is completely removed by passage through tar extractors before the ammonia recovery plant. Three types of extractors are in common use—(*a*) the impact type, in which tar mist is precipitated by contact with metal screens; (*b*) the cyclone type, in which gas is constrained to follow a rapid spiral course, and tar mist is separated centrifugally; (c) the direct type, in which tar is precipitated by means of high-pressure ammoniacal sprays. In some modern plants complete removal of tar is effected in high-voltage electrostatic precipitators.

Composition of Crude Coke-Oven Tar.

Ammoniacal liquor	2·5 – 5·0%	Free carbon in pitch, 6%–15%.
Naphthas	Trace – 0·5%	
Light oils (to 180° C.)	1·0 – 2·0%	Tar acids in carbolic oils, 1%–2·5%.
Carbolic oils (180°–230° C.)	6·0 – 8·0%	Naphthalene in carbolic and creosote oils, 5%–10%
Creosote oils (230°–270° C.)	8·0 –11·0%	
Anthracene oils (270°–360° C.)	16·0 –22·0%	
Pitch	50·0 –63·0%	
Sp. Gr. at 60° F.	1·150– 1·205	

AMMONIA RECOVERY.

Three systems are used :—

1. *Indirect System.*—Gas is cooled to atmospheric temperatures and sprayed with water. The resulting liquor mixed with liquor condensates from the gas-coolers has a concentration of about 1 per cent. ammonia, of which only 0·1 to 0·2 per cent. is fixed ammonia. The liquor is distilled with live steam and a little milk of lime in a continuous still and passed through a saturator containing hot mother liquor and 5–10 per cent. free sulphuric acid. Ammonium sulphate is formed and salts out, being removed by steam or air blowers at intervals. Draining and centrifuging reduces the moisture in the salt to about 1½ per cent. and the acidity to 0·3–0·5 per cent. Complete drying and neutralisation of the salt is performed in a separate plant. The indirect system is confined to early coke works. It is flexible and can be readily turned over to the manufacture of concentrated ammonia liquor should the sulphate market be unprofitable, but results in the formation of about 9,000 gals. of noxious waste liquor per ton of sulphate, which is often disposed of with difficulty. The steam consumption is about 9 lbs. per lb. of sulphate.

2. *Direct System.*—Gas is cooled to a temperature of 75°–80° C., that is, just above its dew-point, to avoid condensation of ammoniacal liquor. After tar extraction the whole of the gas is passed through a closed saturator for ammonia recovery. Heat of reaction between the ammonia and the acid causes so much evaporation in the saturator that, theoretically, the process uses no steam and makes no waste effluent. Actually a small amount of fixed liquor is condensed from the gas and must be distilled into the saturator, or else evaporated to give crude ammonium chloride. This system followed the indirect system, but it is rather difficult to work for complete tar extraction and elimination of liquor condensation, and is not now built into new plants.

3. *Semi-direct System.*—This embodies most of the advantages of previous systems, and is generally used in modern practice. Gas is cooled to 40°–50° C. with complete tar separation and partial liquor condensation. The remainder of the ammonia is recovered by passage of the gas through the saturator into which the previously condensed liquor is also distilled. Approximately 3–3½ lbs. of steam are required per lb. of sulphate, and the amount of waste liquor formed is only one-quarter of that of the indirect method.

BENZOLE RECOVERY.

Oil scrubbing of the gas at 20° C. is used to remove hydrocarbons of the benzene series, the benzolated oil being stripped by live steam to give a crude benzole before being cooled and returned to the scrubbers. The crude benzole, of which approximately 65 per cent. distils at 120° C., is redistilled to eliminate any wash oil, washed with concentrated sulphuric acid, neutralised and then fractionated into various products, motor benzole, pure benzole, toluole, xylene, naphtha, and so on, as required. Typical analyses of oils used for scrubbing, taken from the 1928 Report of the Research Committee of The National Benzole Association, are :—

Oil.	Absorption Capacity.	Viscosity (20° C. Redwood).	Sp. Gr. 20° C.	Drop Point.	Distillation Range.
Light creosote	4·8–4·5	34·6	1·013	200° C.	95% at 300° C.
Medium creosote	4·5–4·0	37·6	1·031	200° C.	„ „ 350° C.
Washed blast-furnace creosote	4·0	39·8	0·935	210° C.	„ „ 340° C.
Green oil (anthracene)	3·75	69·9	1·088	250° C.	„ „ 385° C.
Gas oil, American petroleum	3·0–2·8	40·0	0·864	210° C.	„ „ 363° C.

(The absorption capacity is the per cent. volume concentration of benzole in an oil solution in equilibrium with coal-gas containing 2·4 gals. of benzole per 12,000 cub. ft.)

A circulation of 80–120 gals. of oil through the scrubbers is required per ton of coal carbonised. Creosote oils are most generally used on account of their high absorption powers, but thicken quickly in use and lose this desirable property. They are also rather volatile, so that stripping results in the loss of up to 6 gals. of oil per 100 gals. of benzole made. Blast-furnace creosote is used in Scotland and retains its fluidity well, but tends to give troublesome sludges in the oil coolers and heat exchangers. Gas oil, though of lower initial absorption power than creosote, is very satisfactory as it retains its fluidity, and on account of its low volatility loses little in stripping. Its low specific gravity makes it easily separable from water in the stripping stills and avoids troublesome emulsions. However, its paraffinoid nature causes contamination of the benzole. This is of little consequence in the manufacture of motor spirit, but it is undesirable when the products are to be used in chemical processes. Other methods tried for benzole recovery are absorption in absorbent charcoal or silica gel.

Coke-Oven Gas.

This is practically identical with coal-gas from gasworks' retorts, a typical analysis after stripping of tar and benzole being :—

CO_2. %	CnHm. %	O_2. %	CO. %	H_2. %	CH_4. %	N_2. %	Gross Calorific Value. (60° F. and 30 ins. pressure.)
1·6	2·4	Nil	5·2	58·4	26·2	6·2	524 B.Th.U. per cu. ft.

A surplus of about 110,000 million cu. ft. is made annually. Most of this is supplied to gas undertakings for domestic and industrial use, and to steelworks. Of the total gas produced in 1947 from coke-ovens, 24·5 and 19·3 per cent. representing about 53,000 and 42,000 million cu. ft. were supplied respectively for the above uses.

Low-Temperature Distillation of Coal.

This consists in the gradual heating of bituminous coal to a comparatively low maximum temperature (500° to 600° C.). The retorting vessels employed are designed in such a way as to ensure the rapid removal of the volatile products. By such a process a low yield of very rich gas is obtained, and a high yield of thin tar. The tar is richer in low boiling-point fractions than the tars produced by the gasworks or coke-oven plants. The residual coke contains approximately 10 per cent. of volatile matter, and may be marketed for use in domestic fires, for which purpose it is claimed to be easily ignitable, and at the same time to give off the major portion of its heat as radiant heat, and to burn without generating smoke. For many years the process did not meet with success, largely on account of mechanical difficulties in the operation of the retorts; these difficulties, however, appear to be now overcome in the production of 'Coalite.'

Coalite.

The following results were obtained in a test by the Government Fuel Research Board *:

The coal used in the test was a good average Yorkshire coal, and one which has been frequently tested in every way at the Fuel Research Station.

The yield of products per ton of coal carbonised was as follows :—

'Coalite'	13·92 cwts.
Gas	5,620 cu. ft., or 39·6 therms, after stripping of motor spirit. (See †.)
'Coalite' crude oil	18·62 gals.
Ammonia liquor	26·00 gals.
Crude motor spirit from scrubbing the gas	1·78 gals.
Equivalent ammonium sulphate	13·55 lbs.

The 'Coalite' produced was of a very suitable size (1 to 3-in. pieces), containing only 4·6 per cent. of breeze, and was quite suitable to withstand the rough handling natural to rail or road transport. Analysis of this fuel showed a slightly lower percentage of volatile matter than usual, but when burnt in a household grate it readily ignited and gave a good hot fire.

The yield of crude oil was high, showing 68 per cent. of that obtained in the standard assay apparatus developed by the Fuel Research Board, and upon examination proved to be a normal low-temperature coal oil.

The yield of gas was greater than that anticipated from this particular coal, and averaged

$$\left(\frac{39\cdot6 \times 100{,}000}{5{,}620}\right)^{\dagger} = 705 \text{ B.Th.U. per cubic foot after being stripped of its crude motor spirit.}$$

The motor spirit obtained by scrubbing the gas was 1·78 gals. per ton, but as the crude oil also contained 1·09 gals., the total yield of this product amounted to 2·87 gals. per ton; a most satisfactory result.

The ammonia was also higher than expected, and, when converted into ammonium sulphate, yielded 13·55 lbs.

* Report published by H.M. Stationery Office, Kingsway, London, W.C. 2, price 9d.

TYPICAL YIELDS FROM ONE TON OF COAL.

	Low-temperature Carbonisation.		High-temperature Carbonisation.	
Temperature of Carbonisation °C.	550–600.		1,000.	
	External Heating.	Internal Heating.	Gasworks.	Coke Ovens.
Coke . . . cwt.	13·5–15·5	8–12	13–14·5	14–15
Gas—Volume . cu. ft.	2,500–4,000	30,000–50,000	13,000–20,000	11,000–11,500
„ Cal. value B.Th.U./ cu. ft.	800–900	180–230	470–560	500–560
Therms	22·5–32	69–90	73–94	58–62
Tar . . . gal.	18–22	16–18	10–14·5	8–10
Sulphate of Ammonia lb.	15	—	25	28

From one ton (American, 2,000 lbs.) of bituminous coal, the Colorado Fuel and Iron Company of Denver, U.S.A., obtains in actual practice the following products, viz.:

Coke	**1,400 lbs.**	**Pure benzol**	**1½ gals.**
Coke breeze	**60 lbs.**	**Toluol**	**½ gal.**
Gas	**7,500 cu. ft.**	**Zylol**	**2/10 gal.**
Ammonium sulphate	**25 lbs.**	**Pitch**	**5 lbs.**
Tar	**10 gals.**	**Naphthalene**	**½ lb.**
Light oil	**3½ gals.**	**Creosote**	**1 qt.**
Motor benzol	**2½ gals.**	**Paint**	**1 qt.**

[Annual capacity of ovens, 540,000 tons (American) coke.]

Many experimental plants on various methods of low temperature distillation of coal have been tried. The Fuel Research Board has carried out many tests on these, and reports of the results obtained are published by H.M. Stationery Office.

Coke as a Boiler Fuel.

With a view to cheapening the cost of steam raising by improving steam-boiler efficiency, and to eliminate the emission of visible smoke, gas coke is now largely used as fue lin lieu of crude coal. In hand-fired boilers, coke can be used under ordinary natural draught conditions. With a draught of 0·25 in. W.G. over the fire, the average rate of combustion is 12 lbs. to 14 lbs. of suitably graded coke per sq. ft. of grate area per hour. Some form of impelled draught is necessary if this rate is insufficient. The forced draught apparatus usually employed is some form of steam-jet blower applied to the enclosed ashpit of the boiler, and used to maintain a 'balanced' draught over the fire. The steam consumption of the jets should not exceed 3 per cent. of the total steam generated when burning graded coke at a rate of 20 lbs. per grate-foot-hour. For hand-fired boilers the minimum allowance of grate area should be 1 sq. ft. per 22 to 25 lbs. of coke consumed per hour. In practice it is found that this rate can be exceeded only at the expense of efficiency.

Unlike coal, the combustion of coke is practically completed on the grate, and a limited quantity of secondary air only is required. This tends to localise the heating effect to the boiler furnace area. The introduction of a circulator, tends to neutralise this effect and set up active circulation of the water in the boiler. The operation of cleaning fires, and the losses incidental thereto, due to inrush of cold air and consequent loss of steam pressure, are minimised by the use of the Gallagher Crompton baffle bridge. During the cleaning operation the live fuel is heaped up on the baffle bridge.

In mechanically-stoked boilers fitted with both forced and induced draught, average rates of combustion exceeding 40 lbs. of graded coke per grate-foot-hour are maintained.

APPROXIMATE FUEL COST OF EVAPORATION FROM AND AT 212° FAHR.

Coal Slack.				Hard Steam Coal.				Smokeless Welsh Coal.				Gas Coke.			
Mechanically Stoked.				Hand Fired: Natural Draught.				Hand Fired: Natural Draught.				Hand Fired: Forced Draught.			
At per Ton.		Evaporative Value. 7·5 lbs. per 1,000 gallons evaporated.		At per Ton.		Evaporative Value. 9·5 lbs. per 1,000 gallons evaporated.		At per Ton.		Evaporative Value. 10·5 lbs. per 1,000 gallons evaporated.		At per Ton.		Evaporative Value. 9 lbs. net. per 1,000 gallons evaporated.	
s.	*d.*	*s.*	*d.*	*s.*	*d.*	*s.*	*d.*	*s.*	*d.*	*s.*	*d.*	*s.*	*d.*	*s.*	*d.*
19	0	11	4	26	0	12	2	28	0	11	10	25	0	12	4
20	0	11	11	27	0	12	8	29	0	12	4	26	0	12	10
21	0	12	6	28	0	13	1	30	0	12	9	27	0	13	5
22	0	13	1	29	0	13	7	31	0	13	2	28	0	13	11
23	0	13	8	30	0	14	0	32	0	13	7	29	0	14	5
24	0	14	4	31	0	14	6	33	0	14	0	30	0	14	10
25	0	14	11	32	0	15	0	34	0	14	5	31	0	15	4
26	0	15	6	33	0	15	6	35	0	14	10	32	0	15	10
27	0	16	1	34	0	16	0	36	0	15	4	33	0	16	4
28	0	16	8	35	0	16	5	37	0	15	9	34	0	16	10
29	0	17	4	36	0	16	11	38	0	16	1	35	0	17	4
30	0	17	11	37	0	17	5	39	0	16	6	36	0	17	10
31	0	18	6	38	0	17	10	40	0	16	11	37	0	18	4
32	0	19	1	39	0	18	4	41	0	17	4	38	0	18	10
33	0	19	8	40	0	18	9	42	0	17	9	39	0	19	5
34	0	20	4	41	0	19	2	43	0	18	2	40	0	19	11
35	0	20	11	42	0	19	8	44	0	18	7	41	0	20	5
36	0	21	6	43	0	20	2	45	0	19	0	42	0	20	10
37	0	22	1	44	0	20	8	46	0	19	5	43	0	21	4
38	0	22	8	45	0	21	1	47	0	19	10	44	0	21	10
39	0	23	4	46	0	21	7	48	0	20	4	45	0	22	4
40	0	23	11	47	0	22	0	49	0	20	9	46	0	22	10
41	0	24	6	48	0	22	6	50	0	21	2	47	0	23	4
42	0	25	1	49	0	23	0	51	0	27	7	48	0	23	10

Anthracite.

The bulk of the anthracite produced in Great Britain is mined in the western portion of the South Wales coalfield, the best qualities in Carmarthenshire and Glamorganshire. It consists virtually of carbon—as much as 94 per cent. in the better qualities—with only a small proportion of the constituents of bitumen. It is jet-black, with a metallic lustre, very hard, slow burning, absolutely smokeless, gives intense heat, and burns with a bright red glow. These qualities fit it for many purposes. It is used largely for smelting; in the burning of lime and brick; for the generation of steam; and in the United States and on the Continent of Europe extensively for domestic heating and cooking. For malting and horticultural purposes, owing to its purity and freedom from arsenic, etc., it is in immense demand; while in Norway, Sweden, and Italy, Welsh anthracite is used for making carbide of calcium.

The American coalfield (Pennsylvania) produces the largest amount of anthracite—about 55 million tons per annum—the Welsh output being only about 3,000,000 tons a year at present, though capable of a much greater yield, the major part of the field being still untouched.

One ton of anthracite used in a pressure or suction-gas plant will give power equal to 3 tons of the best steam coal used under boilers. Some typical figures for a suction-gas plant using anthracite were produced by a test carried out at the Metropolitan Reservoirs, Chingford, under the direction of their chief engineer. A guarantee had been given that the consumption would not exceed 1·1 lb. of anthracite per actual b.h.p. hour, and in the report it was stated that the average fuel consumption for the four main pumps was 0·913 lb. per b.h.p. hour, whilst the small pump gave the figure of 0·8 lb. of anthracite. The general average for the whole of the pumps was 0·904 lb. of anthracite per actual b.h.p. hour, and the guarantee lift of the pumps was exceeded by almost 20 per cent. The producer tests showed that 1 lb. of anthracite yields 82·1 cu. ft. of gas at standard temperature and pressure, the net calorific value being 146·3 B.Th.U. per cu. ft., giving a yield of 12,011 B.Th.U. per lb. of anthracite. The calorific value of the anthracite supplied was 14,600 B.Th.U. The fuel used under the boilers which supplied steam for the producer was taken into account.

A ton of Welsh anthracite will produce from 170,000 to 220,000 cu. ft. of gas for power purposes. The following is a typical analysis of good Welsh anthracite:

Carbon	92·7	per cent.
Hydrogen	3·4	,,
Oxygen and nitrogen	2·7	,,
Sulphur	0·4	,,
Ash	0·8	,,
	100·00	,,

In furnace installations where crude gas can be used bituminous fuels are generally used, as the gas is supplied direct to the furnace along with the tar and soot. In small furnaces, where small burners are used, or where necessary to convey gas a long distance, it is generally not possible to use crude gas, and anthracite is more convenient than bituminous coal. The average consumption in a pressure gas-producer is about 13 lbs. of anthracite per 1,000 cu. ft. of gas, and the overall consumption for producer and engine should be about 1 lb. of anthracite per b.h.p. hour.

The principal anthracite collieries are equipped with electrical and other machinery, the preparation of the coal being very elaborate—breaking, screening, washing, etc. A revolving breaker deals with the large coal; this coal travels to the various 'screens' for sizing. These sizing screens are placed one above another at an angle and worked on a shaking principle. Thus the coal is kept steadily on the move to obviate congestion. The large collieries break and screen their product into at least eleven different sizes. The screens are perforated with round holes, through or over which the coal passes in sizes from 2½ ins. by 4 ins. (the largest cobbles) to ⅛ in. by ¼ in. ('grains'). Similar moving belts, but not perforated, convey the screened coal to the pickers, who remove by hand whatever shale it may contain. Impurities are picked from the cobbles and large coal prior to the coal going to the breaker. These picking belts also load the coal into the railway trucks, but only convey the larger sizes, viz. machine-made cobbles and machine-made 'French' nuts. Sizes below the French nut cannot be conveniently picked and sorted by hand, and are therefore transferred to a washer. The 'washery' belts work with the same shaking movement, carrying the small coals, such as stove nuts, peas, beans, pea-nuts, etc., through tanks of water to remove the small particles of stone. To ensure the maintenance of a maximum standard of quality, a coal examiner usually twice daily takes and tests samples of coal from the screens and washery, and the percentage of waste material detected rarely exceeds 1 per cent.

What is known in the trade as 'rubbly culm' is the rough small coal which passes between longitudinal bars 1½ in. apart. It is used principally for lime-burning and steam-raising.

The 'duff,' or very fine, small anthracite, is used in the manufacture of patent fuel and cement making.

In regard to ordinary boiler furnace utilisation of anthracite not much advance has been made, due to the narrow conception of boiler-house practice, coupled with the question of availability. When anthracite is used for this purpose it must not be disturbed, and should be stoked with a

thin bed, very evenly, and if this minimum attention is given an evaporation of from 8–10 lbs. of water per hour can be easily maintained. Tests made in London in which anthracite peas were used showed an evaporation of 12·2 lbs. of water per lb. of coal in one test, and 11·3 lbs. in another. Sometimes the draught or other conditions render the use of anthracite alone impossible, but its blending with other fuels gives very satisfactory results. Where mechanical stoking is used a larger proportion of anthracite—say 2 to 1—can be used, as distinct from equal quantities where hand-firing is adopted.

An important aspect of anthracite has been the recent utilisation of the large accumulations of duff and washery refuse. Hitherto, anthracite dust could not be satisfactorily burnt on standard furnaces, and considering that this fuel has a calorific value of 12,000 to 13,000, with over 80 per cent. fixed carbon, and about 10 to 12 per cent. of ash, it was obvious that valuable material was lying dormant in the dumps. Therefore a new furnace was introduced, with air spaces subdivided into a large number of small holes, and other modifications. The following results of tests at a Welsh colliery show the great increase made in evaporation :

Test No.	Class of Fuel.	Evaporation in Lbs. Steam per Hour.	Remarks.
1	Duff, good quality . .	3,500	Jet pressure 60 lbs.
2	'Grains'	4,400	,, ,, 30 ,,
3	Duff	3 125	,, ,, 45 ,,
4	Duff	3,000	,, ,, 25–30 ,,
5	'Grains'	5,750	,, ,, 30 ,,
6	Duff	4,250	,, ,, 40–50 ,,
7	Duff	4,875	,, ,, 55–85 ,,

The first four tests were carried out on the old type, and tests 5, 6, and 7 on the newly designed bar-furnace.

Burnt under proper conditions, there is no better coal for steam-raising than Welsh anthracite. At one works using English coal it was necessary to employ two Lancashire boilers 30 ft. by 7 ft. 6 in., but after a patent furnace was fitted and anthracite coal introduced, *one* of these boilers was sufficient to do the work previously done by the two boilers.

Below are some of the typical sizes of Welsh anthracite :--

Cobbles	**2½" × 4"**	**'Beans' . . . . .**	**½" × ¾"**
'French' Nuts . .	**1¾" × 2½"**	**'Peas' . . . . .**	**¼" × ½"**
Stove nuts . . .	**1" × 1¾"**	**'Grains' . . . . .**	**⅛" × ¼"**
'Pea-nuts' . . .	**½" × 1¼"**		

Utilisation of Anthracite Wastes.

A series of tests made in America, consisting of 30, 50, and 70 per cent. run-of-mine coking coal mixed with anthracite culm, showed that the pulverised bituminous coal alone produced better results. except the mixture of 30 per cent. culm with 70 per cent. bituminous, which was equal to that of the bituminous coal alone.

Lignite.

The composition of lignite, and its properties, lie between those of wood and coal.

Lignite occurs in most European countries and is abundantly found in the Colonies, but does not occur in quantities in the British Isles. Lignites vary widely in composition, but generally between the following limits : carbon 55 to 80 per cent., hydrogen 5 to 7 per cent., oxygen 10 to 35 per cent. (in the dry state). The calorific value is from 9,000 to 13,000 B.Th.U. on dry sample. The high ash and water contents of lignite frequently make it unsuitable for direct combustion, but good grades may be used for steam raising. Lignite is frequently subjected to destructive distillation, whereby valuable liquid products are obtained. It may also be used for the manufacture of fuel briquettes, or be utilised in gas producers.

Peat.

Large deposits of peat occur within the British Isles, but little use has been made of them hitherto, the high water content (about 95 per cent. when in bog, and about 30 per cent. after air drying) being a great drawback to their utilisation. Peat is suitable for use in gas producers and is sometimes carbonised, when peat charcoal is the main fuel product.

The ash content of peat is normally low, usually from 2 to 6 per cent. in the air-dried peat. On the dry-ash-free basis the percentage composition of peat usually varies as follows:—Carbon 56 to 63, Hydrogen, 5·7 to 6·3, Nitrogen, 1·3 to 2·7, Sulphur 0·3 to 1·0, and Oxygen 31 to 38. Anhydrous peat usually has a calorific value (gross) between 9,000 and 9,500 B.Th.U. per lb.

Wood.

When freshly cut, wood contains up to 50 per cent. of water, but this can be reduced to about 20 per cent. by air drying. The composition of dry wood is about 49·5 per cent. carbon, 6 per cent. hydrogen, 44 per cent. oxygen, and ·5 per cent. ash, and the heat value is generally a little below 9,000 B.Th.U. It only finds use in countries where large forests exist and where coal is not readily obtainable.

Wood Charcoal.

Wood charcoal consists mainly of carbon and possesses a calorific value of 11,000 to 13,500 B.Th.U. (Brame). Its use as a fuel is very limited.

Briquette Fuel.

The name 'Briquette Fuel' is given to the product obtained by agglomerating coal, in a more or less ground state, by means of a binder or agglomerant which, for general purposes, is coal-tar pitch.

Up to the present time no real substitute for pitch on a large scale has been found to be economic, and among the binders tried with varying degrees of success may be mentioned starch, resin, bitumen (oil pitch), and sulphite cellulose.

Coal briquettes can be used without change in the grates wherever coal is used, but it is usually recommended that the bars should be a ½ in. apart in hand-fired boilers. On railway locomotive boilers briquettes are successfully used all over the world, and in fact their successful use is purely a matter of skilful blending of coals for a particular purpose.

The requirements of a good briquette are as follows:—

1. It should be soundly made and free from cracks and thus be able to withstand handling by the ordinary methods used in its transport.
2. Its moisture content should not be above 2 per cent.
3. The ash content should not be above 8–8½ per cent.
4. The volatile matter should be 17–18 per cent.
5. The calorific power should be 7,850–7,900 calories (expressed on the dry fuel).
6. The cohesion should not be below 75 per cent.
7. The ash should not fuse below 1,350° C. (oxidising atmosphere).
8. The sulphur content should be below 1 per cent.
9. The pitch content should not be over 8½ per cent. and the briquette waterproof.

A briquette conforming to the above analyses will meet practically every industrial requirement and can be used in any climate, tropical or arctic, and it is of interest to note that in the Antarctic Expeditions of Mawson, Shackleton and Scott respectively, briquettes were used, not only as fuel for heating purposes, but also as walls to protect the animals from the icy blasts.

Briquettes vary considerably in shape and weight, but for industrial purposes the rectangular shaped block is favoured, varying in weight from 10 to 25 lbs. and of a size 10 × 5¾ × 3¾ ins. to 10 × 8 × 6 ins. Smaller rectangular blocks, weighing from 2 lbs. upwards, are also made, but these find an outlet chiefly for domestic purposes, as well as those of the ovoid or boulet type, which weighs from 1½ ozs. up to 7 ozs.

The British briquetting industry is chiefly centred on the seaboard ports of South Wales, where supplies of good steam-raising coals are available at hand, and shipment of the briquetted fuel is convenient, for the trade is almost entirely an export one.

The pitch used for briquetting coals is known as 'Medium Soft Coal Tar Pitch,' and is the residue left on the distillation of coal tar, during which process the naphtha, creosote and anthracene oils are distilled off. Pitches made from horizontal, inclined and coke oven tars are the most suitable for briquetting, while those made from vertical retort tar are not so good. Pitches made from blast furnace, producer or low temperature tars are not used except under economic stress.

The essentials of a good briquetting pitch are (*a*) covering power; (*b*) binding or agglomerating power; and (*c*) setting power; the pitches made from the three favoured types of tar mentioned above generally conform to these requirements.

High coking temperatures adversely affect the pitch in producing too much 'free carbon' with consequent decrease in binding power, while the same tar distilled from a pot still gives a better pitch than if distilled in a continuous still.

The evaluation of a pitch for briquetting purposes is carried out by tests laid down in the 'Handbook of the Standardisation of Tar Products Tests Committee (S.T.P.T.O.) 1939.' These tests, briefly outlined, are:—

Softening (Melting) Point.—A ½-in. square cube of the properly sampled pitch is moulded or cut and suspended on 18 I.W.G. copper wire in a bath of previously boiled and cooled water. The water is heated 2° C. per minute and the temperature at which the cube drops off the wire is taken as the softening point. This test is perhaps the most important in that it determines whether the pitch can be properly melted in the steam-heated briquetting appliances to be described later.

Volatile Matter.—One gramme of finely divided pitch is heated for 3 minutes in the standard specified platinum crucible over a Meker burner at a temperature of 960° C. The loss in weight is the volatile matter.

'Free Carbon.'—One gramme of the finely divided pitch extracted with pure toluole (or benzole). The undissolved residue is the 'free carbon.'

Ash.—One gramme of the finely divided pitch (or the coke left from the volatile matter determination) is ashed in a platinum or silica crucible.

A suitable pitch, tested by the methods laid down by the S.T.P.T.C., will give volatile matter 65–75 per cent.; softening (melting) point, 75–79° C.; free carbon, 14–21 per cent.; ash not above 0·4 per cent.

The following table gives typical analyses of pitches used for briquetting:—

—	Coke Oven.	Inclined Retort.	Horizontal Retort.	Vertical Retort.
Volatile matter . .	71·0 %	68·5 %	65·5 %	75·0 %
Softening (melting) point .	78° C.	75° C.	76° C.	80° C.
Free carbon . . .	16·0 %	20·0 %	29·0 %	15·0 %
Ash	0·3 %	0·4 %	0·5 %	0·3 %

In actual practice it is customary to mix the various types so that deficiences in one type are balanced by virtues of another type and so preserve a more or less uniform mixture of pitch for briquetting.

Coals.—Suitable coals must be moderately low in ash content, sulphur and moisture, and high as regards the fusibility of its ash.

The coals used vary in volatile matter from anthracites (for domestic purposes only) of 6 per cent. volatile matter to bituminous coals of 22 per cent. volatile matter. Higher volatile coals can be used when special requirements have to be fulfilled, but they are not generally favoured for industrial purposes. The coals used are known as 'small coals' and may be Breaker Duffs (large coal mechanically broken and screened), purchased Washed Duffs, Rubbly Smalls (these being washed at the works) and Unwashed Duffs.

All Washed Duffs are dried before use so as to reduce the moisture content to about 2–3 per cent.

It is usual to build up a briquette to the desired specification by using a variety of coals of different types for, as a general rule, it may be said that no single coal fulfils all requirements for a particular purpose and, at the same time, have chemical and physical characteristics sufficiently flexible to meet other specific requirements. These deficiencies can, however, be overcome by skilful mixing of coals, and only a good knowledge of their chemical and physical properties, as well as of the briquetting process, will enable a judicious blend to be made.

These coals may be classified according to their volatile matter as below:—

	Per Cent.		Per Cent.
Anthracites . .	6–8	Semi-bituminous . .	16–18
Dry steams . .	10½–12	Bituminous . . .	19–22
Steams . . .	12–14½		

A suitable briquette for industrial and railway locomotive purposes would be composed of the following mixture:—

Bituminous and semi-bituminous coals . .	25 per cent.
Steam coals	45 ,,
Dry steam coals	22 ,,
Pitch	8 ,,

This will give an analysis conforming to figures given previously in this article.

It is, of course, necessary that a good proportion of these coals be washed coals which have been dried prior to briquetting and also that care must be exercised in their choice as regards the fusibility of the ash. This latter factor is an important one for with modern day conditions of high grate temperatures the ash should not melt below 1,350° C., but better still not below 1,400° C., a condition easily attained in the South Wales industry.

Anthracite up to about 5–7 per cent. is included in the coal blend in briquettes intended for land boilers where the steaming load is not heavy, the anthracite in this case replacing an equivalent quantity of the dry steam class.

Mention has been made of the cohesion test, which is an empirical test designed to examine the soundness of the briquette to withstand transport conditions, such as the rolling of a boat at sea. This test consists of rotating in a cylinder (3 ft. × 3 ft.) of iron, fitted with radial ribs, 100 lbs. of the briquettes cut to definite sized pieces, each weighing approximately 2 lbs. The iron drum is rotated 50 times in 2 minutes, the contents then being sieved through a 1½-in. mesh sieve, supported at an angle of 40° to the horizontal and the quantity retained on the sieve is expressed as a percentage cohesion of the original quantity used.

A low cohesion (below 70 per cent.) may be due to unsuitable pitch, excess moisture or poor pressing.

Domestic fuel is made in many instances on a small scale at individual collieries, where the coal is generally highly bituminous. This type consists of one class of coal only and may be in the form of rectangular blocks weighing 2–6 lbs. The ovoid or boulet type is also made from this class of coal.

The ovoid type finds an outlet in domestic and central heating plants and the composition of the coal blend is varied according to requirements: from purely anthracite with or without admixture with an equal part of the steam coal, to an ovoid approaching the composition of the briquettes intended for industrial purposes previously described. One of the difficulties encountered in the use of anthracite solely in the manufacture of ovoids is the fact that on combustion on the grate the ovoids crumble to powder due to the lack of caking power of the anthracite. This, however, need not be the case if care is taken in the choice of a suitable pitch. The idea of using anthracite implies that a certain degree of smokelessness is aimed at and while the volatile matter of an anthracite ovoid is normally about 11 per cent. (which is the same as for a smokeless steam coal) yet the pitch used produces a certain amount of smoke. With the object of eliminating this smoke, efforts have been made to incorporate starch (cassava) with about 1 per cent. of pitch as the binder, the idea of this small amount of pitch being to render an otherwise moisture susceptible ovoid impervious to moisture. There are, however, difficulties in the actual manufacture and the other method of rendering the ovoids smokeless (*i.e.* the pitch-bound ovoids) is to subject them to a carbonisation process in a static or continuous oven, but here again the operation of carbonising suffers from mechanical and other defects, not the least being the sticking of the ovoids in a solid mass in a static oven. The ovoids so treated can be used for burning in enclosed stoves and are smokeless.

An interesting development is the production of a modified ovoid of a cobble size weighing about 2 lbs., intended for locomotive use and general industrial purposes. The composition of its coal blend is that given previously for industrial rectangular blocks and, due to its very convenient shape, presents a very large surface for combustion and on trials in locomotive boilers has given excellent results. Another advantage being that it does not require man handling but can be picked up with grabs.

Process and Plant.

The following is an outline of the general practice of briquetting as applied to rectangular blocks:—

1. Preparation of a suitable blend of coals and grinding to a predetermined size.
2. Breaking and grinding the pitch and automatically measuring the requisite amount to be added to the coal blend.
3. Mixing and heating the coal-pitch mixture in iron cylinders (' pugs ') where jets of superheated steam cause the pitch to melt and cover the coal particles with a film of pitch.
4. Removal of the occluded steam with simultaneous reduction of moisture and cooling of the plastic coal-pitch mass to the temperature required for briquetting in the presses.

1. Grinding of the coals is carried out in mills of the cage type (Carr mills) or the swing-hammer type, the coal, unless previously washed, being passed over a magnetic seperator to remove pieces of iron which are frequently found in coal wagons.

The fineness of grinding has a bearing on the pitch consumption, and while very fine grinding improves the appearance of the briquettes yet it means increased pitch consumption. Another factor in the pitch consumption is the rank of the coal, *i.e.* the nearer the approach to the anthracite the higher the consumption as a general rule.

2. The pitch is broken down roughly by means of pitch crackers, which have rolls with claws on their periphery, and then ground finely, the finer the better, in small Carr mills.

In summer time trouble is experienced with pitches which are too soft and in some instances the fine grinding is not carried out but the coarsely ground pitch is fed along with the coal to be ground in the coal grinder only, thus enabling a soft pitch to be used.

3. The heating of the coal-pitch mixture is carried out in iron cylinders (pugs), about 8 ft. high and 4 ft. 6 ins. diameter, for presses with an output of about 25 tons per hour. Through the middle of the pug is a stout central iron rotating axis, to which are attached radial arms or knives, the object being to slowly agitate the coal-pitch mass which, on its downward travel, is subjected to the action of superheated steam emitted from jets which are fixed through the perimeter of the pug and stepped up spirally. Efficient heating or ' pugging ' is one of the essentials

of the process. An outlet at the bottom of the pugs enables the flow of the hot coal-pitch mixture to be regulated to the pressing machines.

4. The steaming hot mixture must be rid of its occluded steam and reduced to a temperature suitable for pressing for, in actual practice, the briquette must withstand five to six times its own weight of similar shaped briquettes, superimposed on each other on trolleys in a time period of about 6 to 8 minutes from the time it comes off the press. It will thus be seen that the selection of the pitch to produce this setting and the regulation of temperatures play a very important part.

The type of press favoured in this country is the steam press, while on the Continent the mechanical press is preferred. Both give good results under proper operating conditions, though the stream press is the more flexible in that it produces a sound block even from an improperly filled mould, while the mechanical press with its fixed motion will not do so.

The excess of steam is removed prior to briquetting and use is made in some cases of a series of trays where the mixture is mechanically dropped through slots while a current of warm air passes upwards through the falling mass. A very successful method is to subject the hot steaming mass to the action of a direct flame, the hot mass being tossed about. This causes rapid evaporation of the excess steam and at the same time cools the mass to the required temperature for pressing.

The presses are single or double acting, and pressures of 2 tons per sq. in. which are said to be obtainable are rather optimistic for, as a rule, the figure is rarely over 1 ton pressure and higher pressures are not productive of good blocks.

Ovoid or Boulet Type.

This form is made between two rolls on which cups or indents are made corresponding to half the shape of the ovoids, so that when each roll is properly sychronised with the opposing roll, the two halves meet and form the recess in which the ovoid is pressed. Care in aligning is necessary, otherwise badly formed ovoids with halves overlapping are produced, while unsuitable pitch and excessive moisture may cause the ends to gape or open out ('Duck Bills.'). Small amounts of tar are sometimes sprayed in hot by means of a steam jet to the coal-pitch mixture in the pugs. The ovoids should be well polished and resist considerable impact almost immediately on coming out of the press.

Another aspect of briquetting is that of 'pure coal' briquetting, *i.e.* no binder used, high pressures being utilised to bind the fine coal particles together. While good sound blocks of very small sizes up to ½ lb. have been made in experimental presses, no commercial process for making briquettes for fuel purposes seems to have withstood the test of time. Grinding to 200 mesh is necessary for producing a good briquette from bituminous coals, but considerably finer grinding is necessary if attempts are made on steam coals. Pressures of around 10 tons per sq. in. are found necessary for the former and around 20 tons for the latter.

The chief difficulty appears to be the problem of obtaining metals to withstand such high pressures without distortion and the constant strain of 10 tons per sq. in. imposed on certain parts of the machines is no doubt the cause of the failure of large-scale industrial plant not remaining in operation for any length of time. In Germany, however, the use of presses of the Ring-Roll type operating at pressures of about 15 tons per sq. in. has been successfully applied for briquetting brown coal without a binder.

Firebars.

The following proportions are for the usual cast-iron bars : –

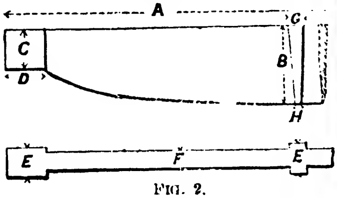


FIG. 2.

A.		B.	C.	D.	E.	F.	G.	H.
Ft.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
2	0	3¼	1½	1¾	1¼	⅝	⅜	⅜
2	6	3½	1⅝	1¾	1¼	⅝	⅜	⅜
3	0	3¾	1¾	2	1¼	¾	⅜	⅜
3	6	3⅞	2	2	1⅜	⅞	⅜	½
4	0	4¼	2	2¼	1⅜	⅞	⅜	½
4	3	4½	2	2¼	1⅜	⅞	⅜	½
4	6	5	2¼	2¼	1½	1	⅜	½
4	9	5¼	2¼	2⅜	1⅝	1	7/16	⅝
5	0	5¼	2¼	2½	1¾	1⅛	7/16	⅝

Firebars require to be suited to the fuel. Clean riddled coal-nuts or cobbles may be burned on a ½-in. bar with ¼ in. to ⅜ in. spaces, but slack coal requires wider bars and narrow spaces. Bars are often made with projecting fingers or corrugations in plan, thus increasing the length of air space considerably, though reducing it in respect of the increased width of the bar body. Such bars are made to rock for the purpose of shaking the ashes out of the fire and of breaking up clinker. Air, to supply a secondary demand behind the bridge for the combustion of the gases, is often admitted through hollow bars which heat the air and keep themselves cooler thereby, and under suitable conditions good results may be obtained.

Some firegrates are built up of short bars or links pinned together to form a side travelling band or chain; such grates are self-cleaning and are much used in mechanical stokers, and are good, if provided with suitable air control damper plates below. Other firegrates have a somewhat steeply inclined surface, and may be built up either of longitudinal bars or of transverse bars, in which latter case they form the so-called 'step' grate. Such grates may have a creeping or a shaking movement of the bars in order to assist gravity to lower the burning fuel down hill to make up at the foot of the slope the vacant space left as the fuel burns away. The fuel is mechanically fed in, above, or at the head of the sloping grate surface, and gradually descends under the combined influence of gravity and the moving bars. A dumping device is often added at the foot of the slope by means of which the ash and refuse may be cleared from the furnace by movement of a lever.

Grates for anthracite coal, which develops an intense heat in the fire—not sending off much gas by the heat-absorbing process of distillation—are often made in the form of flat plates with a number of perforated nipples or little mounds raised half an inch above the general surface. The space between the air mounds fills level with dust and preserves the iron surface from heat and checks the adherence of clinker.

Firegrate bearers must never be rested upon the actual plates of furnace tubes of internally fired boilers, but must rest on angle bracket pieces riveted to the plates by not less than two rivets each. For convenience in handling it is usual to employ two or three bars in the length of a 6-ft. grate.

Hollow bars are necessarily (or more conveniently) made in one length; this, however, makes them cumbersome, and is one cause of their being sometimes unpopular, for they are heavy to remove and replace when cleaning; but where conditions are favourable, hollow bars with outlets to the surface and a steam-impelled supply of air through the bars will beget economy, by enabling a cheaper grade of fuel to be employed.

Coal Combustion and Smoke.

Bituminous coal cannot be burned smokelessly except under certain proper conditions. These are that the whole of the products of combustion must be allowed to travel over the full length of the fire towards the bridge; that additional air must be admitted in fine streams above the fire after each fresh charge of fuel. The fresh air thus admitted mingles with the gases from the fuel, and the mixture must be hot enough to burn at the bridge, beyond which point there must be a clear space in which combustion may complete itself. The combustion chamber may with advantage be wholly or partially brick-lined. The draught necessary depends on the output of the boiler. To compel air to enter over the fire, dampers are sometimes fitted to the ash pits.

The most satisfactory results in coal combustion are usually obtained with thin fires, but the thickness of a fire depends on the draught and on the size of the fuel. The largest fuel will bear the thickest fire.

The combustion space should be at a high temperature so as to ensure burning all the combustible into carbon dioxide and water, which would give smokeless operation. Although about 14 lbs. of air is required to burn one pound of soft coal completely, only half this amount should be admitted through the grate; the remainder should be supplied over the fuel bed. If this can be well pre-heated, improved combustion will be obtained.

Leaving the door open a short interval after feeding fresh coal supplies an extra quantity of this secondary air when most needed. Poor draught always causes smoke and lowered efficiency. Ample combustion space is an advantage with bituminous coal and pulverised coal and with oil.

There are two general ways of firing bituminous coal. One is to place a thick bed of coal on the front grates. Here it will coke, and the gases given off will pass over incandescent fuel and be consumed, the coked fuel then being pushed over the remaining grate area. This is well adapted to return tubular boilers.

The second method is that of spreading the new coal evenly over the grate in small quantities over one-half the fuel bed. One-half of the fuel bed is kept incandescent. A uniform size of coal, neither coarse nor fine, gives best results. High-volatile coals require a bed of four to six inches, and those with less volatile somewhat thicker beds.

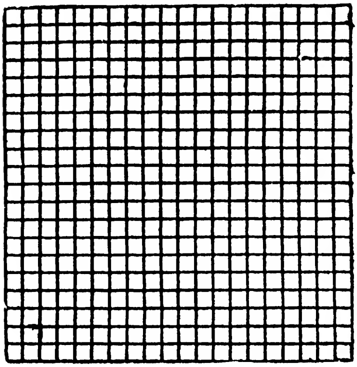
Where gases rise directly upwards from a grate amongst water tubes without the admixture by travel pointed out above, smoke is liable to be made.

Fine dust fuel can now be burned at the rate of 40 lbs. per sq. ft. of furnace area per hour. Given furnace room, the cheapest fuel may be the most economical. In practice the best economy is that due to the use of the fuel which can be employed to raise the steam necessary at the cheapest rate.

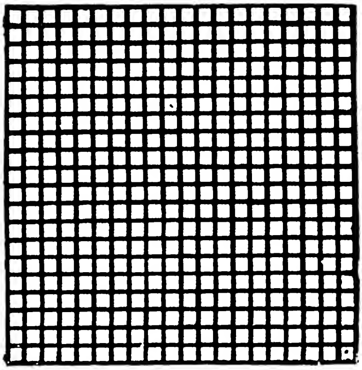
The foregoing of course, is qualified by the amount of ash in the coal where the expense of removal is great.

With externally fired boilers which permit of a liberal design of furnace with suitably disposed firebrick, smoke can readily be prevented. It is less easy to secure perfect combustion with internally fired boilers, because of the difficulty of applying durable firebrick linings. But the question of smoke prevention is wholly one of air mixture and temperature. The burning gases must not too soon be brought into contact with cold plates or tubes.

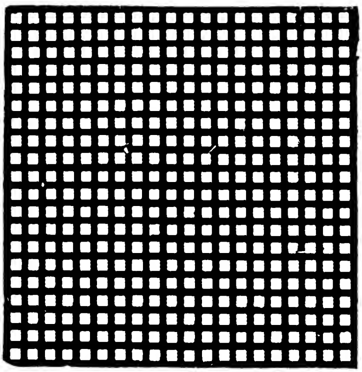
In burning bituminous coal only a part of the total heat is generated at the grate. Part of the heat of the fuel is absorbed in driving off the volatile parts of the coal in gaseous form. This heat becomes latent and very materially reduces the furnace temperature. The gases pass off, containing a third of the total heat capacity of the coal, as well as the heat of gasification. Unless mixed with air and burned later, these gases may be totally lost. To burn bituminous coal not only demands thorough air mixture, but also a sufficient temperature, and the heat of the burning gases must not be abstracted by too early bringing them into contact with the cool surfaces of the boiler. Sufficient length and clear space must be afforded for combustion to complete itself,



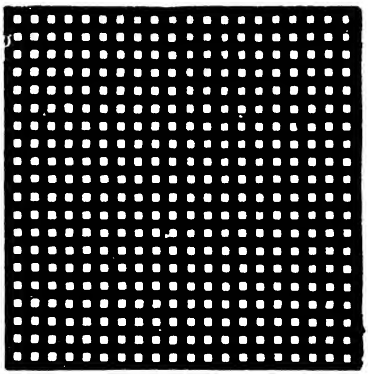
1. Light grey smoke.



2. Darker grey smoke.



3. Very dark grey smoke.



4. Black smoke.

FIG. 3.

or black smoke will be formed. The furnace arrangements of the Lancashire boiler are correct in form, but the water-cooled plates are a hindrance to complete combustion in the case of hand-firing; when high temperature mechanical-stoker conditions are obtained, however, it is easy to prevent smoke with bituminous coal in a Lancashire boiler.

With the ordinary arrangement of bridge, however, the flames pass over the bridge in more or less parallel streams, and the secondary air which passes over the fire has thus only access to the upper side of the flames. When fresh coal is added there is a rush of volatile hydrocarbons, and the underside of the flames produced have insufficient air to complete combustion and smoke is produced. There are on the market several smoke consuming devices in which a regulatable amount of air is admitted through the bridge to the underside of the flame, and by judicious use

of these it is possible to supply the requisite air to avoid the production of smoke. Another method is to cause a violent turbulence in the gases before they have been cooled sufficiently to prevent completion of combustion. This causes complete mixing of the flame gases, promotes and accelerates complete combustion, and assists heat transmission. Such a device can of course only be used successfully where the draught is adequate, but gives freedom from smoke and improved working results when used under suitable conditions.

Water-tube boilers require a special setting to enable them to burn coal smokelessly. No furnace can be smokeless where the gases from the fire rise vertically and pass amongst cold-water tubes. Bituminous fuel can rarely be burned satisfactorily unless there is some extent of firebrick lining to protect the gases from too early chilling. Cross water-pipes in the inner flues of the Lancashire or Cornish boiler are bad, and will seriously hinder perfect combustion and produce black smoke.

The use of preheated air—the air for combustion being heated by the waste gases from the boilers—is becoming increasingly common in the best practice. In addition to the added heat in the air, the use of the hot air promotes rapid combustion, giving hotter and more intense flames with less tendency to smoke production, and thus increases the capacity of both furnace and boiler. It also enables inferior coals to be burned with greater ease. Care must however be taken as the increased furnace temperature will cause more troublesome clinker and greater wear and tear on the grates if unsuitable coals are used.

Colour of Smoke.

The colour of smoke is indicated by Ringelmann's smoke charts, which are numbered from No. 0, all white, to No. 5, all black. In fig. 3 intermediate sizes 1, 2, 3, 4, are shown, the ratio of line to space breadth being respectively, 10 : 90 ; 23 : 77 ; 37 : 63 ; and 55 : 45. Large charts drawn to these proportions and hung about 50 feet from the observer where the chimney top can be seen simultaneously, if possible, are very great aids in correctly estimating the relative blackness of smoke.

The four scales of the figure are called light grey, darker grey, very dark grey, and black the extreme all white or all black being respectively no smoke or dense black.

MECHANICAL STOKERS.

There are several types of mechanical stokers ; for internally fired boilers (such as the Lancashire), the underfeed coking and sprinkler, and, for water-tube boilers, the travelling grate, the single horizontal retort, and the multiple inclined retort.

With regard to the internally fired boilers, by reason of their construction a stoker of the self-cleaning type is not easily arranged, and for this reason the efficiency of such boilers is rarely as good as that of the water-tube boilers. In the underfeed stoker the coal is forced onwards and upwards in a central trough by a helix conveyor, the hydro-carbons being distilled while the fuel is rising from the trough, and burnt while passing through the incandescent fuel bed, thus removing the possibility of smoke ; and the coke is burnt while passing over the side grates, so that the ash and clinker collect at the extreme sides of the furnace, from which position they can be conveniently removed. In the coking stokers, the fuel is usually fed on to the front of a moving grate, the volatile portion being expelled and burnt near the front, and the coke burning while travelling towards the back of the fire under the influence of the moving grate surface. In the sprinkler stokers the fuel is thrown on to the grate surface by means of a throwing shovel or similar device. All three types of stoker are capable of burning the fuel efficiently, but the underfeed type has the advantage as regards smokeless combustion.

The single retort machine is quite satisfactory for small boilers, and has the advantage that it is driven by a self-contained steam motor. The multiple retort type of stoker gives satisfactory results with many kinds of coal.

The travelling grate stoker is highly efficient and is capable of dealing with a wide variety of fuels ; it has the advantage that each portion of the grate is in the furnace only one-third of each complete revolution, so that it may be adequately cooled and repaired, if necessary, without shutting down the boiler ; the ash is discharged automatically over the back end of the grate.

In its original form it had the disadvantage that the fuel bed tapered from full thickness at the front to nothing at the back and thus offered a diminishing resistance to the air supply, the majority of which passed through the thinner fuel layer. In plants subject to varying loads this led to excessive amounts of excess air to secure smokeless combustion and limiting

the power of the stoker. It has now been improved by dividing the wind box into a number of compartments provided with suitable regulating dampers so as to secure regular combustion under all load conditions, and this has resulted in greater outputs and considerably increased thermal efficiency.

It is necessary to study carefully the position, size and arrangement of the 'ignition arch' used with such stokers to suit the particular class of fuel to be used if ready ignition and even and smokeless combustion is to be attained.

MECHANICAL DRAUGHT.

Various mechanical methods have been introduced for increasing the draught for boilers; this may be accomplished by an exhausting fan drawing the products of combustion through the flues, or by blowing air into a closed stokehold or an enclosed ashpit, and forcing the air through the fire with the necessary pressure, which varies according to rate of combustion desired. There is also a combination of these mechanical means, termed 'balanced draught,' in which air is forced under pressure into the ashpit by one fan and the gases are drawn off by an exhausting fan so arranged that there is neither pressure nor vacuum above the fire. Another method is the 'ejector draught' system in which the flue gases are drawn up the chimney by a high-speed jet of air.

The advantage of mechanical draught produced by a fan is that the first cost is less than half that of a chimney; also the draught is independent of atmospheric conditions, can be varied instantaneously to suit the demand for steam, permits of economisers and allows the reduction of the temperature of the waste gases to be carried to the lowest practicable point. It increases the capacity of the boiler, and permits of a more perfect combustion with less excess of air. It allows the use of lower grade and less expensive fuel, and so produces steam more cheaply.

Any desired rate of combustion from the normal of 20 lbs., to 30 or 40 lbs. of coal per square foot of grate area, may be maintained with a mechanical draught plant. The air volume required depends upon the style of boiler, condition of boiler setting, skilful stoking, the kind of coal used, and other factors; in round numbers 200 to 250 cu. ft. of air will be required per lb. of coal; these figures represent about 12 per cent. and 10 per cent. CO_2 respectively.

To drive a fan requires about ½ to 1 per cent. of the total power generated. The full benefits of mechanical draught are only obtained where means are employed to utilise to the fullest extent the heat of the flue gases (by means of superheaters or economisers) before the latter are discharged into the chimney. The cost of installing a fan and engine is very much less than the cost of a chimney, so that mechanical draught is economical in first cost.

Fan draught enables a heated air supply to be employed, and it is said that by heating the air supply to 180° the fire temperature will be raised an additional 360°, owing to better combustion, etc.

In order to ensure success it is necessary to have each portion of the plant properly proportioned so as to give a proper 'balance' to the whole. In many plants much power is wasted and results obtained are unsatisfactory owing to inadequacy of flues, etc., or the incorrect proportioning of some other portion of the plant throwing the rest out of 'tune.'

PULVERISED COAL.

General.

Coal in pulverised form was first successfully applied to cement-burning kilns. This method of heating cement kilns has now become universal practice, the steady long flame with its resulting continuous and unvarying heat development inside the rotary kiln being ideal for this work. Moreover, the ash which has always been a source of trouble in other applications forms part of the finished product in this industry.

For cement burning the coal is reduced to a fineness of about 80 per cent. through a 100-mesh screen (10,000 holes to the square inch). It is essential to pulverise to a greater degree of fineness for general applications. A coarser product can be permitted when burning high volatile fuel (30 per cent. V.M. and upwards), or in the case of very large steam-raising units, as now installed in the latest super-power stations in America and elsewhere, but the American standard of fineness, *i.e.* 85 per cent. through a 200-mesh screen (40,000 holes to the square inch) should be accepted for general boiler firing and for all metallurgical work.

This fine degree of coal-dust, coupled with a low pressure on the air supply, thereby decreasing the velocity of the gases passing through the furnace or boiler, and a wider adaptation of the principle of turbulent mixing of the fuel and air, resulted in the removal of the former troubles due to the formation of ash slag in chequer-work, on metal to be heated, or on boiler tubes.

It is now a recognised and successful practice to apply pulverised coal to all classes of metallurgical furnaces, and in a very marked degree to power-house boilers. (Open-hearth steel furnaces require special consideration.)

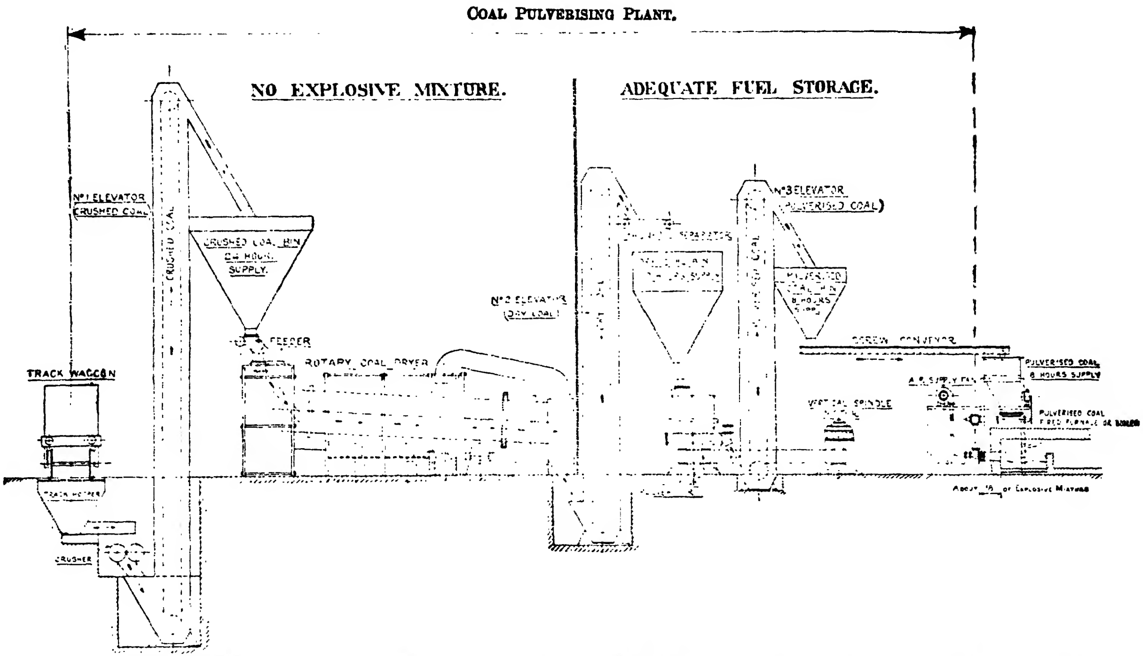
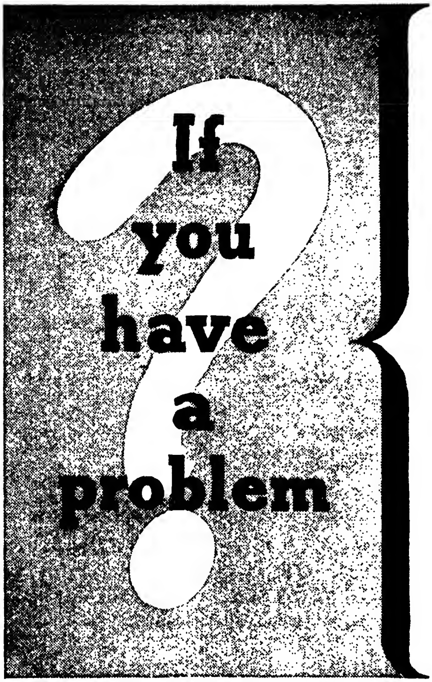


Fig. 4.—General View of 'Central' Pulverised Coal Plant, showing flow of coal from the Track Wagon to the furnaces, and indicating the positions of the Coal Crusher, Elevators, Coal Dryer, Pulveriser Mill, Magnetic Separator, Screw Conveyor, Furnace or Boiler Bin, and Screw Feeder. In place of the Screw Conveyor either the pump or the air-pressure system is now generally adopted for long-distance transmission.



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The use of pulverised fuel as a means of firing boilers, furnaces, cement kilns, locomotives, public supply, and incidental purposes throughout the world now accounts for a vast quantity of industrial fuel, perhaps 100 million tons per annum.

Although chimneys of good height are advisable for large installations, apparatus has been devised to trap much of the dust carried in the flue gases. Tests carried out at Milwaukee show that as little as 15 per cent., and as much as 85 per cent., of the ash in coal passes out at the chimney top under certain conditions. These conditions are controllable, and whatever may be the amount permitted to escape into the atmosphere, this dust is unlikely to become a nuisance, for it is in such fine particles that the dust rises into space, and its deposition is very widely spread.

Pulverised coal-fired boilers are now guaranteed to give a continuous evaporation of 13 lbs. (from and at 212° F.) of water per sq. ft. of heating surface, at a boiler and furnace efficiency of 81 to 83 per cent. and a peak load evaporation approaching 20 lbs. per sq. ft., with water-tube furnace linings in series with the boiler elements.

Mill House Plant for Central System.

Fig. 4 shows an outline arrangement of a typical coal-pulverising plant with screen separator mill and rotary coal-fired dryer.

The coal delivered by rail wagon is crushed to $\frac{3}{4}$-in. cube and passed up the No. 1 elevator to the crushed coal storage bin.

From this bin the crushed coal is fed by gravity into the rotary dryer, wherein the moisture content of the coal is reduced to 1 per cent. This is a necessity for bulk pulverisation, storage, and for conveying the fuel through pipes.

Rotary coal dryers are fitted with furnaces, either hand fired, stoker fired, or arranged for burning powdered coal. The hot gases from the furnace should not pass through the dryer in direct contact with the coal until these gases have been considerably cooled. Some dryers are so designed that the hot gases do not pass over the coal before they have travelled the full length of the dryer through a central tube, when they are diverted into the outer portion of the dryer and pass back to the furnace end in direct contact with the coal. For high volatile coal this type is essential.

The average coal consumption in the dryer is about 20 to 30 lbs. of coal per ton for a reduction of 10 per cent. in the moisture content. When coal is very wet it then becomes necessary to pass the coal twice through the dryer. The final temperature of the coal leaving the dryer should not be above 250° F.

From the dryer the coal passes up No. 2 elevator to the dried coal bin, passing over a magnetic separator pulley in order to remove any pieces of iron from the coal, and is fed by gravity into the pulveriser.

With the static dryer, which has no moving parts, a portion of the waste gases from the boilers, generally about 10 per cent., is forced through the coal, with the result that the coal is thoroughly dried without any loss of the volatile constituents or fine dust.

The stationary type of dryer has been generally adopted for power-house installations. Initial troubles, regarding the flow of coal from the dryer and uneven or over-heating, have been eliminated.

Raw coal can be delivered to the bunker above the dryer, from whence it falls by gravity through the dryer to the pulveriser mill. Thus, the rotary dryer motor, the No. 2 elevator, and, if necessary, the dry coal bunker, shown in fig. 4, are dispensed with, and only in special cases are rotary coal-fired dryers necessary.

The application of this dryer does away with a considerable portion of conveying machinery, for handling both the raw coal and the dry coal at the same time, eliminating expensive plant and fuel used to dry coal. As much as 2 to 3 per cent. of the coal purchased is sometimes employed in drying fuel in revolving dryers, with the static dryer this coal is replaced by waste gases from the boilers. On the Continent the revolving disc, grid, or shelf type of dryer heated by superheated steam is favoured.

Screen separator mills have been used very extensively in the cement industy for pulverising the coal and for grinding the cement clinker. In this type of mill the fine particles of coal are thrown tangentially through the mill screen; the fine product is then raised by means of No. 3 elevator to the powdered coal bin, or it is delivered direct into the screw conveyor or into a pump, for transport to the bins at the furnaces.

In the air-separator type of mill the fine coal-dust is exhausted through a fan and deposited by means of the upward air current into the cyclone collector, whence it is delivered into the powdered coal storage bin, or direct on to the screw conveyor or pump.

The power required for operating a screen separator mill with its pulverised fuel (No. 3) elevator is somewhat less than the power required for operating a screen separator mill and exhaust fan. The wear and tear on the latter may be excessive when abrasive hard coal is pulverised, but the vacuum extraction of the coal-dust from the mill tends towards a cleaner mill-house.

Against the screen separator mill may be placed the occasional damage to the screen by a stray piece of iron. Any perforation of the screen due to this cause permits coarse coal to escape with the pulverised product.

Both types are in general use, and at the present time the advocates of one are as numerous as the advocates of the other.

Hardinge ball mills in conjunction with Hummer screen separation are also in use for pulverised coal production.

Transportation of Pulverised Coal.

The original method of transporting coal-dust was by means of weathertight screw conveyors. This method, fig. 4, although used to some extent for small and compact installations, has given place to air pressure and pump transmission systems for larger plants or greater distance delivery.

For any one of these methods of conveying pulverised fuel the product must be dry, otherwise packing troubles will occur.

Feeders.

Whenever coal-dust is held in bulk at furnace bins, feeders are attached thereto fitted with screws or worms to introduce the dust into the air-blast as it enters the furnace, or compressed air is used as a means of syphoning out the fuel.

The screw or worm of the feeder is usually operated by means of a variable speed motor.

Burners.

The usual type of coal-dust burner met with in general practice has been of very simple construction, merely consisting of an air-carrying supply pipe into which coal dust is fed by the screw feeder. These simple types of burner have given fairly good results, but later practice has established the necessity for a more intimate mixing of the air and powdered coal.

Many special burners have therefore been devised whereby a pre-mixing of coal dust and air is effected before the final stream of air for combustion is introduced.

Exceptionally good results have been recently obtained with the turbulent mixing types of burners, such as the Peabody, the Buell, the Bailey-Tenney, Couch, Fraser & Chalmers, the International Co.'s Type 'R,' the Clarke Chapman, and the Fuel Research Stations Grid and Multijet burners, to effect a short diffused flame.

In each type a well-conceived combination of air and coal passages with adjustable or deflection plates or vanes renders it possible to control the direction, length, and intensity of flame.

By reason of the thorough mixture of air and coal, pulverised coal can be applied direct to the fire tubes of Lancashire and Cornish boilers without the use of external combustion chambers, unless high ash coal is used, when it is preferable to collect much of the ash prior to its passage through the boiler tubes. These burners also eliminate the extensive furnace alterations required for long flame burners.

No difficulty is experienced in burning low ash coal (10–15 per cent.) without external combustion chambers.

By reason of wide diffusion of the air and coal-dust mixture, the areas of combustion chambers usually provided for hand or stoker firing are in most cases sufficient.

An actual example of results obtained with the 'Buell' burners, as applied to the stoker settings of a B. & W. boiler of 3,240 sq. ft. of heating surface, without external combustion chambers, is as follows:—

	Stoker Firing.	'Buell' Burner.
Coal—B.Th.U. per lb.	12,562	10,370
,, lbs. per hour	1,792	1,175
Water evaporated from and at 212° F. per hour	13,365	10,552
,, Per lb. of coal	7·4	9·0
Efficiency of boiler and furnace	57 per cent.	83 per cent.

Another method of liberating a very greatly increased number of B.Th.U. in a small combustion area has been developed in the 'Well'-type Burner, which consists of a rectangular chamber with pulverised coal burners at each angle, these producing a high degree of turbulence within the water-cooled burner chamber.

Power of Motors.

MILL HOUSE PLANT (FOR 40 TONS PER 10 HOURS).

Preliminary crusher (not required with slack coal)	15 to 20 h.p.
Nos. 1, 2, and 3 elevators	7½ h.p. each.
Dryer	10 to 15 h.p.
Pulveriser mill, screen separation	60 „ 75 h.p.
„ „ air „	80 „ 100 h.p.

(Mill capacity, 4 to 5 tons per hour.)

FUEL TRANSPORTATION (say 5 tons per hour, 300 ft. and 40 ft. lift).

*By screw conveyor	15 h.p. (motor).
†By air pressure	75 „ (compressor).
*By air mixture	60 „ (fan mixer).
*By screw pump	20 „ (screw pump and compressor).

* Continuous operation. † Intermittent operation.

COAL-DUST FEEDERS.

Constant or variable speed motors . . . . . 1 to 3 h.p.
(½ h.p. motors are sometimes used, but are generally unsuitable under mill conditions.)

AIR FANS.

No accurate figure can be given for the power required for air fans, as this will depend upon the size of burner used. The total quantity of air per pound of coal allowed is usually 180 cu. ft., 60 to 100 cu. ft. being supplied at a pressure of 4 to 6 in. w.g.; the remainder is induced at the burner when furnaces operate under induced or chimney draught.

PRE-GASIFICATION OF COAL.

Much of the success obtained in the various applications of pulverised coal has been due to the fine and uniform pulverisation of the coal, and the provision of combustion chambers of considerable cubic area.

Until the advent of the turbulent mixing type of burner, the latter requirement precluded the development of pulverised coal-firing for such purposes as marine boilers, Lancashire boilers, locomotives, and the smaller range of trial furnaces.

A greater measure of success is still to be obtained by subjecting pulverised coal to temperatures in the neighbourhood of 1,000° C. at high velocity, and with an air supply sufficient only to maintain this refractory surface temperature, gas can be generated and delivered direct to furnaces or boiler firing chambers with little loss of the sensible heat.

Developments are now in train for the practical application of pulverised coal in this manner, and a measure of success has already been obtained. Due to the extraction of dust in the pre-gasification chamber, this system lends itself particularly to those applications where dustless conditions on the hearth or in the furnace are essential. Moreover, it will doubtless be possible to utilise such generators or pre-gasifiers with either liquid fuel or pulverised coal.

Self-contained Pulveriser Units.

Great attention has been given to the perfecting of these small unit mills, which can be placed alongside the boilers or furnaces to be heated. So much progress has been made in design, reduction of power consumption, maintenance costs, etc., that their installation at French and German iron and steel works and at American power houses has been generally accepted. Whereas power consumption was anything up to 30 kW. per ton of coal, these units now operate at 12 to 15 kW. for similar fuel as burned at the furnace, including air supply. These units are also pulverising fuel containing up to 15 per cent. of moisture, but this moist fuel must necessarily be supplied direct to the furnace burners, and not stored in bunkers.

The figures given in the table on p. 1348 represent the capital outlay required for complete mill-house plants of various capacities. The initial cost of same, with transportation system, furnace bins, motors, feeders, etc., necessitates an expenditure which, in many cases, will or can only be contemplated when actual results have proved that the collective advantages and economies thereby introduced render such an investment wholly attractive.

For smaller fuel requirements and for testing pulverised-coal firing upon individual boilers or furnaces, a self-contained pulveriser unit can be conveniently installed. There are now many efficient machines of this type available.

Coal containing comparatively high percentages of total moisture—20 per cent. or so—can be pulverised in several of the well-known types of units, and through the medium of hot air, up to 500° C.; the resultant fuel powder will be practically bone dry. Coal containing 5 per cent. to 10 per cent. of moisture can be dried and pulverised with air at atmospheric temperature, the resultant degree of fineness depending upon the humidity in the atmosphere at the time of grinding.

By the installation of self-contained units the value of pulverised-coal firing for any specific purpose can be ascertained, so that the question of capital outlay upon a complete mill-house plant and attendant equipment can be based upon the results ascertained in this manner.

Pulverised Coal at Sea.

In the following table certain available data are set out concerning four ships in which pulverised coal is or has been used.

Name of Ship:	s.s. *Hororata.*	s.s. *Metz.*	s.s. *Berwindlea.*	s.s. *Mercer.**
Shipping Line, or Owners:	New Zealand Shipping Co.	Blue Star Line, Ltd.	Berwindlea Steamship Co.	U.S. Shipping Board, leased to Consolidation Navigation Co. (Oriole Line)
Tonnage	11,240	—	7,500	6,219
Trade route	London – New Zealand *via* Panama	Towage on Rhine	U.S.A. and West Indies	New York–England
Name of system	Howden-Buell	'Resolutor' pulveriser	Clarke Chapman	Peabody
Equipment supplied by	R. & H. Green and Silley, Weir, Ltd., London	Fours & Apparells Stein	Clarke Chapman & Co., Ltd., Gateshead	Peabody Engineering Corporation, New York
First trip on pulverised coal	December 22, 1928	—	August 15, 1929	November 1927
Trips or miles to date †	Over 40,000	—	Over 6,000	Over 60,000
Number and type of boilers in ship	6 Scotch marine 4 furnace 17′ 0″ × 12′ 0″	2 S.M. single-ended	2 S.M. single-ended	3 S.M. 3 furnace
Boilers pulverised coal-fired	3 Scotch marine	2 S.M. single-ended	2 S.M. single-ended	3 S.M. for pulverised fuel and oil
Heating surface to combustion area. Ratio	1 cu. ft. comb. vol. to 5 sq. ft. H.S.	—	—	—
Maximum fuel burned per hr. per cu. ft. Comb. area	2·88 lbs.	—	4·2 lbs.	—
Maximum fuel burned per hr. per boiler	1,870 lbs.	—	1,344 lbs.	1,950 lbs.
Number, type, and rating of pulverisers	3 ball type, 2,000 lbs. per hr.	2 'Resolutor' each 2 tons	2 'Resolutor' (1 per boiler), each 1 ton per hr.	2 ball mills, each 3,600 lbs. per hr.
Method of driving pulverisers	Steam turbines	Steam turbines	D.C. electric motors	Non - condensing steam engines

* This ship has been treated largely as a pioneer experimental ship.
† As at September 1929.

Name of Ship :	s.s. *Hororata.*	s.s. *Metz.*	s.s. *Berwindlea.*	s.s. *Mercer.**
Shipping Line, or Owners :	New Zealand Shipping Co.	Blue Star Line, Ltd.	Berwindlea Steamship Co.	U.S. Shipping Board, leased to Consolidation Navigation Co. (Oriole Line).
Maximum moisture in raw coal	7% New River coal, Panama	12%	12%	—
Method of drying coal	Heated primary air through pulverisers	Heated air through pulverisers	Heated air through pulverisers	Heated primary air through pulverisers
Method of fuel distribution	Howden - Buell patent	Ring main	Ring main	1 Peabody patent mechanical 2 patent pneumatic 3 Peabody patent
Number, rating, and type of burners per boiler	Four 480 lbs. per hr. Howden - Buell patent	1 each furnace Woodeson-Stein	1 each furnace Woodeson patent, 560 lbs. per hr.	Combined coal and oil burners
Total power per ton of coal conveyed to burners	18 kW.	15 kW.	15 kW.	—
General results:				
Quantity, economy in fuel	10%	20%	—	—
Cost, economy in bunkers	10%	—	New ship, 1·277 lbs. coal per i.h.p. hr.	—
Labour economy in stokehold and trimming	50% (firemen)	—	Self - trimming bunkers. One man only required in stokehold for 6 furnaces; no trimmers	A saving of 6 firemen per 24 hrs. as compared with hand-firing
Special consideration applying to the system	Suitable for usual run-of-mine coal as bunkered for hand-firing at coaling ports		—	—

* This ship has been treated largely as a pioneer experimental ship.

Some valuable results of pulverised coal-firing in marine Scotch boilers, for the Harrison Line steamers of the Charente Steamship Company, Ltd., s.s. *Musician* and s.s. *Recorder*, were given in *The Engineer* of October 30, 1931. These results were amplified in a paper presented before the Institute of Marine Engineers on January 31, 1933, by Mr. W. G. Gibbons and Mr. M. Arthurson, which was abstracted in *The Engineer* of February 17, 1933, from which article the following particulars are taken.

The *Musician*, built in 1919, is a more or less standard Class 'A' ship, bought by the company in 1924. She is 385 ft. b.p. by 51 ft. 9 ins. beam moulded by 29 ft. 3 ins. depth moulded. She has a load line displacement of 10,680 tons, with a block coefficient of 0·777. Her engines are 27 ins. + 44 ins. + 73 ins. by 48 ins., with a working pressure of 180 lbs. She has three single-ended boilers, 15 ft. 7½ ins. mean diameter by 11 ft. 6 ins. long; three furnaces in each boiler, 3 ft. 10 ins. diameter; tubes, 2½ ins. outside diameter by 7 ft. 8 ins. b.p.; total heating surface, 7,668 sq. ft.; fitted with Howden forced draught; no superheat.

The *Recorder*, one of nine sisters, was built in 1930, and commenced work in November of that year. She is one of the latest types of ships built to suit the owners' special requirements, and is 420 ft. b.p. by 54 ft. 3½ ins. beam moulded by 32 ft. 7 ins. depth moulded; with a load line displacement of 13,000 tons and a block coefficient of 0·75. The engines have cylinders 27 ins. + 46 ins. + 77 ins. by 54 ins., with a working pressure of 210 lbs. per sq. in. She has three single-ended boilers, 16 ft. 3 ins. mean diameter by 12 ft. long; three furnaces in each boiler, 4 ft. internal diameter; tubes, 3½ ins. diameter by 7 ft. long b.p. The combustion chambers are large, being 3 ft. 7½ ins. long at top and 3 ft. 10 ins. at bottom inside. The total heating surface in the three boilers is 7,200 sq. ft., the capacity of the furnaces 755 cu. ft., and of the combustion chambers 1176·7 cu. ft., or a total combustion space of 1931·7 cu. ft.

There are three pulverisers of Clarke, Chapman and Co.'s make, one to each boiler, direct driven by Belliss and Morcom steam engines at 1,200 to 1,350 r.p.m., according to the type of coal. The fuel feed mains are so arranged that with the spare lengths of pipe supplied any pulveriser can feed other boilers in addition to its own boiler. Hot air for the pulverisers and secondary air for the furnaces is supplied by a fan through a tubular heater in the uptake. The main steam is superheated by a Sugden uptake superheater to about 520° F. at the engines.

The following are working results. The *Musician* is doing the same work on 29–30 tons of inferior coal as she did previously on 34–35 tons of better coal.

It is difficult to get exactly similar passages of the *Recorder* and her sister ships, owing to the different displacements, consumption, speeds, weather, slip and condition of bottom, but the three passages given below are very good comparisons.

	s.s. *Recorder*. Liverpool to Port Said, May 30, 1931.	ss. *Contractor*. Liverpool to Port Said, May 9, 1931.
Distance	3,179 nautical miles	3,186 nautical miles
Average speed . . .	11·4 knots	11·3 knots
Daily consumption . . .	37·7 tons	40 tons
Slip	5·3 per cent.	5·6 per cent.
Draught mean . . .	21 ft. 3 ins. leaving	21 ft. 1½ ins. leaving
Total coal burned on passage .	439 tons	468 tons
Description of coal . . .	Staffs. slack	Washed Yorkshire mixed

The *Recorder* shows a saving of 37 per cent. in cost of fuel for this passage.

	Liverpool to Corpus Christi, Oct. 19, 1931.	Liverpool to Galveston, Aug. 28, 1931.
Distance	5,081 nautical miles	4,958 nautical miles
Average speed . . .	10·8 knots	11 knots
Daily consumption . . .	37·7 tons	36·6 tons
Slip	9·2 per cent.	6·7 per cent.
Draught mean . . .	15 ft. 0½ ins.	17 ft. 4 ins.
Total coal burned on passage .	739 tons	686 tons
Description of coal . . .	Bolsover slack	Washed Yorkshire mixture

The *Recorder* shows a saving of 28 per cent. in cost of fuel for this passage.

The cost of replacing wearing parts in the pulverisers works out at about the same as the cost of hand-fired furnace fittings.

A number of analyses made of the coal in the *Recorder* and the samples of ash obtained from the combustion chambers, tubes, and uptakes, show that there is only 1·4 per cent. of the combustible matter in the coal left in the ash.

As regards engine-room crew, the *Musician* has been reduced from fifteen to thirteen firemen, trimmers, and greasers, and given one extra junior engineer. The *Recorder* has twelve, instead of fifteen to eighteen firemen, trimmers, and greasers, and one extra junior engineer.

Steam is easily controlled and all readings, such as pressure, temperature, CO_2, etc., are very steady, and hardly vary throughout a passage. No trouble is experienced when working in and out of port. The boilers can be fired much harder than with hand firing, without trouble or loss of efficiency. Speaking generally, very little trouble has been experienced with either ship.

Pulverised coal firing in a ship makes much greater demands on the engineers in charge, as there is much more to look after and keep adjusted, but it eases the work of the firemen. So far as the authors can see, it is much easier on the boilers; there is no 'firebar line' along the furnace sides, and the furnaces scale more or less all round. In the *Musician*, as is usual in forced draught jobs of her type, the owners were constantly renewing combustion chamber stay nuts opposite the furnaces, but since fitting pulverised coal firing to two of her boilers, they have not renewed a combustion chamber nut, and so far there are no signs of such repairs being required in these two boilers, though they are doing over 75 per cent. of the work of the ship.

Pulverised Coal for Locomotives.

The advantages for pulverised coal, lignite, etc., as a locomotive fuel are—

30 to 40 per cent. fuel economy as against hand-firing, running mileage and cylinder power of engines increased, cost of firing tools and repairs to fire grating eliminated, no cinders or sparks from the smoke stack, reduction of fire-box strains, firing conditions with pulverised-coal firing analogous to oil-firing with similar fuel-bunkering facilities. Any local fuel within reason can be used.

The following fuels have been and can be used:

	A	B	C	D
Chicago and North-Western Railway.				
	Per Cent.	Per Cent.	Per Cent.	
Illinois bituminous	3·18–15·36	10·00	34·47	10,720–12,400
Kentucky bituminous	1·9 – 2·8	8·00	36·54	13,964
North Dakota lignite	1·8	9·32	47·25	10,960
New York Central Railway.				
Pennsylvania bituminous	0·51–0·91	8·94–13·21	21·63–31·35	13,671–14,096
Brazilian bituminous	1·73–9·15	23·14–29·33	9·5 –28·04	8,820–10,177
The Delaware and Hudson Railway.				
Pennsylvania anthracite culm	0·5	12·22	8·30	12,000(average)
„ bituminous	0·5	9·0	33·0	13,750

A. Range of moisture. B. Ash. C. Volatile matter.
D. Calorific value, B.Th.U. per lb. (dry).

On the Lehigh Valley Railway a locomotive has been run on a mixture of 55 per cent. raw anthracite silt and 45 per cent. bituminous coal. Full steam pressure is maintained easily at all times, and the anthracite silt contains 38·42 per cent. of ash, and has a B.Th.U. value of 9,675, while the soft coal which is mixed with this has an ash content of 11·6 per cent. and a B.Th.U. value of 13,650. Analyses of these coals are as follows:—

Anthracite Silt.	Per Cent.	Soft Coal.	Per Cent.
Moisture	2·26	Moisture	1·90
Volatile	10·39	Volatile	36·59
F.C.	49·30	F.C.	49·89
Ash	38·42	Ash	11·60

The raw coal and lignite to be burned on locomotives in Italy has the following analysis, figures representing the limits in each case are given:—

	Per Cent.		Per Cent.
Moisture	12·52 to 26·80	Volatile	31·5 to 60·94
Ash	11·6 to 32·60	Carbon	35·26 to 68·50

B.Th.U. values from . . . 3,640 to 10,440

Refractory Linings for Combustion Chambers.

The construction and composition of the linings of combustion chambers for pulverised coal fired furnaces are questions of considerable importance.

Although the experience gained of recent years has rendered it possible so to design combustion chambers that the refractory linings are not subjected to the severe destructive action from the flame and ash slag as heretofore, it is still highly desirable under all conditions to have a lining that will not only withstand high temperatures, but will at the same time resist the corrosive chemical action of the ash under these conditions.

In the case of high temperature metallurgical furnaces the factor of corrosion is one that must be carefully considered in view of the presence of metallic oxides introduced through the furnace charge.

Most coal ash consists chiefly of alumina, silica, and iron oxide, together with very much smaller quantities of five or six metallic oxides.

Metallic oxides are of primary importance, in view of the fact that many of them have a marked fluxing action on almost any refractory material. Thus, a fuel ash that contains much iron oxide has in itself a destructive action on furnace linings, irrespective of any of its other constituents.

Apart from this, some fuel ash contains a predominating amount of basic material, whilst in other cases there may be an excess of acid material. When a furnace is to be fired generally or entirely with a fuel containing a basic ash it is highly desirable to provide a basic lining, for by so doing much of the trouble and expense for renewal and repair of furnace linings will be avoided.

Preliminary analysis and investigation in connection with the composition of the coal ash and furnace linings will often save expense and prevent lining troubles during the subsequent operation of the plant.

The more common and cheaper grades of refractory bricks usually contain a more or less equal amount of silica and alumina, and the slight excess of one or the other should be taken advantage of. Combustion chambers of suitable design and having the requisite combustion area will then run satisfactorily with linings of ordinary firebricks.

If, however, there are exceptional conditions to contend with, such as high temperatures, confined combustion space, chemically active ash, etc., it is necessary to make use of the best grades of refractory materials.

In many cases carborundum (silicon carbide) can be used as a facing material for linings with good results. Carborundum, however, although generally chemically inert, will not resist the action of iron oxide or ash containing much potassium or sodium compounds.

In some few cases pure silica bricks can be used when the fuel ash composition is suitable and high temperatures have to be withstood, but special provision must be made for expansion of the bricks.

Pure alumina, magnesite, or bauxite are all high-grade refractories that may be used to resist high temperatures when ash or slag is of a basic nature.

Advantages of Pulverised Coal.

The chief advantages of using coal in pulverised form are :—

Utilisation of low grade coals, peats (coals with 30 per cent. and 40 per cent. ash); lignites, anthracite slush and pit washings can be used.

Control of flame temperatures with uniform heat regulation and greatly reduced requirement of attendant labour—in fact, labour required is equivalent to oil-firing practice.

Increased efficiencies of furnaces due to direct firing of the fuel in minute particles, causing instantaneous and complete combustion, without smoke production.

Some approximate savings in fuel consumptions that have been realised in practice are as follows :—

Boilers—80 per cent. efficiency, and savings of 15 to 50 per cent. over hand firing.

Open-Hearth Steel Furnaces—a saving of 300 lbs. per short ton of steel as against producer gas for small plants.

Puddling Furnaces—a saving of 1,500 lbs. per short ton of iron over hand firing, and from 300 to 500 lbs. over mechanical stokers.

Reheating Furnaces—20 to 40 per cent. saving over hand firing.

Heavy Forge Furnaces—25 per cent. saving over hand firing.

Light Forge Furnaces—20 to 40 per cent. saving over hand firing.

Annealing Furnaces—30 per cent. saving over hand firing.

Tinning and Galvanising—20 to 30 per cent. saving over hand firing.

Sheet and Pair Furnaces—30 to 40 per cent. saving over hand firing.

Copper Smelting Furnaces—40 per cent. saving over hand firing.

Locomotives—30 to 50 per cent. saving over hand firing.

Cupolas—30 to 50 per cent. saving in coke and hotter iron.

Crucible (Brass Melting) Furnaces—The use of coal in place of metallurgical coke or oil.

The output from many of these furnaces has been greatly increased by reason of coal-dust firing, what are known as 'stickers' in the tin-plate trade having been reduced 60 per cent., whilst the quantity of tin and zinc used has been reduced considerably owing, in a great measure, to the improved surface of the metal, and also to the more even heating of the kettles.

Reduced oxidisation of steel and iron heated in powdered coal-fired furnaces and increased life of annealing boxes have accounted for an annual saving of many thousand pounds in some works.

Very important economies have been made in firing malleable iron melting and annealing furnaces with pulverised coal, so much so that The Grindle Fuel Equipment Co. of America have specialised exclusively on this work.

Mills and Millhouse Plant.

TABLE SHOWING NUMBER OF PULVERISER MILLS REQUIRED FOR INSTALLATIONS OF VARIOUS CAPACITIES, LABOUR HOURS, COAL DRYERS, ETC.

Daily Output in Tons.	Output per Hour in Tons.	Number and Size of Mills.		Number and Size of Dryers.		Number of men per Shift.	Labour. Hours per Day.	No. 8-Hour Shifts per working Day.	Maximum Output per Shift Tons.	Approx Output per Labour Hour Tons.
		Running Plant.	Standby Plant.	Running Plant.	Standby Plant.					
		Ins.	Ins	Tons.	Tons.					
5	½	1–24	none	1–4	none	1	16	2	4	0·32
10	2	1–33	”	1–4	”	2	16	1	16	0·62
15	2	1–33	”	1–4	”	2	32	2	16	0·47
20	2	1–33	1–33	1–4	”	2	32	2	16	0·625
30	2	1–33	1–33	1–4	”	2	32	2	16	0·94
40	2	1–33	1–33	1–4	”	2	32	2	16	1·25
50	4	1–42	1–42	1–4	”	2	32	2	32	1·55
60	4	1–42	1–42	1–4	”	2	32	2	32	1·87
70	4	1–42	1–42	1–4	”	2	32	2	32	2·19
80	4	1–42	1–42	1–4	”	2	48	3	32	1·63
90	4	1–42	1–42	1–4	”	2	48	3	32	1·87
100	6	3–33	1–33	1–8	”	2	48	3	48	2·1
150	8	2–42	1–42	1–8	1–8	2	48	3	64	3·1
200	12	3–42	1–42	1–10	1–10	3	48	2	96	4·2
250	12	3–42	1–42	1–14	1–14	3	72	3	96	3·5
300	16	4–42	1–42	1–14	1–14	3	72	3	128	4·2
350	16	4–42	1–42	1–14	1–14	3	72	3	128	4·9
400	20	5–42	1–42	2–10	1–10	4	96	3	160	4·2
450	20	5–42	1–42	2–10	1–10	4	96	3	160	4·7
500	24	6–42	1–42	2–14	1–14	5	120	3	192	4·2
550	24	6–42	1–42	2–14	1–14	5	120	3	193	4·6
600	28	7–42	2–42	2–14	1–14	6	144	3	224	4·2
650	28	7–42	2–42	2–14	1–14	6	144	3	224	4·5
700	32	8–42	2–42	2–14	1–14	6	144	3	256	4·9
750	36	9–42	2–42	4–10	1–10	7	168	3	288	4·5
800	40	10–42	3–42	4–10	1–10	8	192	3	320	4·2
850	40	10–42	3–42	6–10	1–10	8	192	3	320	4·4
900	40	5–57	1–57	4–14	1–14	6	144	3	320	6·25
950	48	6–57	1–57	4–14	1–14	6	144	3	384	6·6
1,000	48	6–57	1–57	4–14	1–14	6	144	3	384	6·9

In a properly designed plant, and with due attention to ordinary common-sense regulations, there is *no fear of explosion* from the coal-dust, and, as has been already stated, the old troubles with the ash slag have been overcome.

There are certainly many instances where the use of fuels in powdered form would show a handsome return for capital invested, and the following table will enable an estimate to be made as to economy that can be expected.

APPROXIMATE COST OF COMPLETE MILL HOUSE, PLANTS AND OPERATION EXPENSES.

If static dryers are installed the cost of plant can be reduced by 5 per cent. up to 350 tons, and by 7½ per cent. thereafter, and the cost for coal for rotary dryers is then omitted.

Daily Output in Tons.	Labour. Cost per Ton.	Power. Cost per Ton.*	Dryer Fuel, Cost per Ton. 2 per cent. of coal dried.	Repairs, Cost per Ton.	Without Stand-by Plant. Cost of Mill House Plant, including Building.	Without Stand-by Plant. Interest per Ton at 6 per Cent.	Without Stand-by Plant. Depreciation per Ton at 10 per Cent.	Without Stand-by Plant. Total Preparation. Cost per Ton.	With Stand-by Plant. Cost of Mill House Plant, including Building.	With Stand-by Plant. Interest per Ton at 6 per Cent.	With Stand-by Plant. Depreciation per Ton at 10 per Cent.	With Stand-by Plant. Total Preparation. Cost per Ton.
	s. d.	s. d.	d.	d.	£	s. d.	s. d.	s. d.	£	s. d.	s. d.	s. d.
5	5 10	3 0	10½	4	3,500	2 9½	4 8	17 6	—	—	—	—
10	2 11½	2 10	10½	4	6,700	2 8	4 5½	14 1½	—	—	—	—
15	3 11	2 10	10½	4	6,700	1 9½	3 0	12 8¾	—	—	—	—
20	2 11	2 10	10	4	9,300	1 9	2 10½	11 6½	10,300	1 11½	3 3	12 1½
30	1 11½	2 10	10	4	9,300	1 3	2 1	9 3¾	10,300	1 4½	2 3½	9 7¾
40	2 2	2 8	10	4	9,300	11	1 7	8 6	10,300	1 0½	1 9	8 9½
50	1 2	2 8	10	4	11,100	10½	1 6	7 4½	12,700	1 0	1 8½	7 8½
60	1 0	2 8	10	4	11,100	9	1 3	6 10	12,700	10	1 5	7 1½
70	10	2 8	10	4	11,100	7½	1 1	6 4½	12,700	8	1 2½	6 7¾
80	1 1	2 6	10	3¾	11,100	6½	11	6 2¼	12,700	7	1 1	6 5¼
90	1 0	2 6	10	3¾	11,100	6	10	5 11¾	12,700	6½	11	6 1¼
100	10½	2 6	9¼	3¾	14,300	7	11½	6 0¾	15,350	7¾	1 0½	6 2
150	7	2 6	9¾	3¾	15,000	4¾	8	5 3¼	18,100	5¾	9¼	5 6
200	5½	2 6	9¾	3¾	20,000	4¾	8	5 1½	25,500	6	1 0	5 6¾
250	6¼	2 6	9¾	3¾	20,000	4	6½	5 0¼	25,500	5	8¼	5 3
300	5½	2 6	9¾	3¾	26,100	4½	7	5 0	31,600	5	8½	5 2¼
350	4½	2 6	9½	3¾	28,300	4	6½	4 10½	32,000	4½	7¼	4 11¾
400	5¼	2 6	9½	3¾	28,300	3½	5¾	4 9¾	32,000	4	6½	4 11
450	4¾	2 6	9½	3¾	31,000	3¼	5½	4 8¾	35,500	3¾	6¼	4 10
500	5½	2 6	9½	3¾	31,000	3	5	4 8½	35,500	3½	5¾	4 9½
550	4¾	2 4	9½	3¾	32,000	2¾	4¾	4 5¼	37,000	3¼	5½	4 6½
600	5½	2 4	9½	3¾	32,000	2½	4¼	4 5	37,000	3	5	4 6¼
650	5	2 4	9½	3¾	32,000	2½	4	4 4¼	37,000	2¾	4¾	4 5¼
700	4½	2 4	9½	3½	35,000	2½	4	4 3¾	41,000	2¾	4¾	4 4½
750	5	2 4	9	3½	37,000	2½	4	4 4	42,500	2¾	4½	4 4¼
800	5½	2 2	9	3½	43,000	2½	4¼	4 2¼	50,500	3	5	4 4
850	5	2 2	9	3½	45,500	2½	4¼	4 2¼	53,500	3	5	4 1
900	3½	2 2	9	3½	48,000	2½	4¼	4 0¾	54,000	2¾	4¾	4 3½
950	3¼	2 2	9	3½	49,500	2½	4¼	4 0½	55,200	2¾	4½	4 1½
1000	3¼	2 2	9	3½	53,500	2½	4¼	4 0½	59,500	2¾	4¼	4 1¼

Cost of plant exclusive of foundations and erection.

* Can be reduced to ⅓rd for power-station plants.

Table is based on costs in 1939 and the following assumptions :—

Labour at £4 per man per 44-hour week.

Power at 1½*d.* per kilowatt-hour (a high cost even for local supply), and an average allowance of 20 kw.-hours per ton of coal pulverised and delivered to the exit of mill house, including power for all operations. Dryers to be fired with pulverised coal, which is taken as 40*s.* per ton at the dryer burners; an average allowance of 2 per cent. of the coal dried is made for dryer fuel, assuming the moisture content of coal to be reduced from 15 per cent. to 1 per cent.

Yearly output is based on 300 working days.

REFUSE DESTRUCTORS.

Calorific Value of Refuse.

The calorific value of refuse is about $\frac{1}{5}$ to $\frac{1}{7}$ that of an equal weight of good dry coal.

Average composition and calorific value of London ashbin refuse, containing the average percentage of water :—

Component parts of Refuse.	Amount in 100 lbs. of Refuse.	Calorific Value.
	Lbs.	B.Th.U.
Coal	·35	3,260
Coke	·15	1,200
Bones and offal	·37	1,970
Breeze and cinder	25·55	102,000
Rags	·40	1,330
Paper, straw, fibrous material, and vegetable refuse	13·15	33,100
Ash	47·00	Nil
Dust and dirt	9·75	"
Bottles	·48	"
Metals, including tins	·68	"
Crockery	1·72	"
Glass	·37	"
	99·97	142,860

The above is the average of 1,000 cart-loads.

About 40 per cent. of the refuse is combustible.

The character of the refuse varies a great deal. In factory districts the percentage of unburnt cinder is high, whereas in residential districts with gardens it is much lower.

When it is considered that ordinary green garden refuse when piled in a heap over a nucleus of fire will consume itself, it is obvious that refuse containing cinders will contain enough heat if properly applied, for its perfect destruction. It is essential that combustion should take place in brick cells, and that no heat should be abstracted until combustion is complete, after which the gases may be used for steam raising. If 40 per cent. only of the refuse is combustible, its calorific value per pound will be about 5,800 B.Th.U. if it were dry. Taking it only as 1,500, this is more than would raise the temperature of the whole gaseous and solid matter to a red heat. If half the weight were water, this would absorb about 600 B.Th.U. to evaporate and about 300 to become red hot, or 900 in all. The remainder of 600 B.Th.U. would raise the gases and solids to nearly 5,000° F. Thus destruction may be perfect with only the equivalent of one-tenth its weight of dry carbon. Inodorous combustion demands high temperature, and hence must occur in brick cells, with suitable movement of the fuel, so that the gases and steam cannot escape until completely decomposed.

London Refuse.

Towns' refuse varies in general composition, moisture and heat value, and the annexed table by the Ministry of Health gives a general analysis of London refuse for the twelve months of 1925–6, when 1,202,003 tons were collected and disposed of in various ways.

Seasonal Analysis of London Refuse for Twelve Months.

Content of Refuse.	Spring 1925.	Summer 1925.	Autumn 1925.	Winter 1926.	Total Tons.	% of Total.
	%	%	%	%		
Fine dust minus $\frac{5}{16}$ in.	28·05	16·93	28·05	35·84	336,233	28·14
Fuel cinder over $\frac{5}{16}$ in. but minus $\frac{3}{4}$ in.	15·18	8·26	15·18	17·76	174,102	14·57
Fuel cinder above $\frac{3}{4}$ in. but minus $1\frac{1}{2}$ ins.	12·94	7·05	12·94	14·07	144,729	12·11
Putrescible matter	12·57	24·07	12·57	8·82	164,183	13·73
Paper.	14·73	23·54	14·73	9·84	179,694	15·04
Metal : containers and other metals	3·54	4·71	3·54	2·98	43,112	3·61
Rags, including bagging, etc.	1·75	2·05	1·75	1·64	21,271	1·78
Glass : bottles and cullet	3·23	3·28	3·23	2·42	3 ,852	3·00
Bone	1·03	0·80	1·03	1·96	11,509	0·97
Combustible debris, unclassified	3·89	6·92	3·89	3·40	51,288	4·29
Incombustible debris, unclassified	3·09	2·39	3·09	2·27	40·030	2·76
					1,202,003	100·00

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From tests of towns' refuse collected in the winter time the following calorific values were obtained :—

Fulham	3,700	B.T.U. and	24	per cent.	moisture.
East Ham	3,350	„ „	37	„	„
Lichfield	2,300	„ „	16·4	„	„
Glasgow	5,290	„ „	14·0	„	„
Hove	3,950	„ „	33·4	„	„

Summer refuse has about 65 per cent. of the heat value of winter refuse. All of the above values are as collected.

The largest and most completely equipped refuse disposal plant in the world is that at Glasgow, built by Heenan & Froude, Ltd.

The nominal capacity of the plant is 670 tons of crude city refuse per day, but records show that 750 tons can be dealt with if required.

The refuse is mechanically handled throughout and, with the exception of the removal of metals, the whole is cremated. The steam generated is used for driving two 5,000 kw. 3-phase 50-cycle turbo-alternators running condensing.

Over 90 per cent. of the current is transformed from 6,500 V. to 20,000 V. and delivered to the Dalmarnock Power Station for distribution.

The remainder (usually about 7 per cent.) runs the station auxiliaries and charges the batteries of the fleet of refuse collection vehicles after being transformed and converted as required.

Forty cells are provided for the burning of the refuse arranged in 8 groups of 5 cells each, complete with water-tube boiler and superheater. The grates are withdrawn by hydraulic power, the clinker dropping on to a lower platform, where it remains for the time being and gives up its heat to the air for the combustion of the fresh refuse on the grate above. This ensures rapid ignition and an increased thermal efficiency of the plant. When next clinkered, the old clinker is scraped off the lower platform by knuckled scrapers on the underside of the grate and the new clinker drops into its place, in turn igniting the fresh refuse and heating the air for combustion. The old clinker is removed in wagons running on a light railway in the basement.

Refuse at Glasgow shows on analysis about 5,000 B.Th.U. per lb., and forms about 38 per cent. of clinker and dust, and the official acceptance test of 16 weeks' duration at the Govan Plant is summarised in the following : Refuse is burnt in 30 out of 40 cells, 25 per cent. being spare plant. Water-tube boiler equipment supplies superheated steam to two 5,000 kW. turbo-alternators arranged condensing :—

Dates of test—First period, April 15 to June 15, 1929 ; Second period, February 10 to April 12, 1930.

Total period under observation (actual) = 16 weeks.
Total refuse passed through plant = 45,604 tons 11 cwts.
Total material extracted from refuse = 2,293 tons 14 cwts.
Total material burnt = 43,310 tons 17 cwts.
Percentage of refuse burnt to total received = 95 per cent. approximately.
Electrical units generated = 13,520,890 kW. hrs.
Maximum load recorded during runs = 10,400 kW.
Electrical units per ton of refuse burnt, including starting up and shutting down periods = 313·35 kW. hrs.
Cell hrs. burning, including starting and stopping periods = 66,105 hrs.
Average burning rate per cell per hr. = 13·1 cwts.
Average burning rate per day of 24 hrs. on whole plant = 628 tons.
Refuse burnt, excluding stopping and starting periods, etc. = 26,056 tons 10 cwts.
Cell hrs. burning, excluding stopping and starting periods, etc. = 37,530·5 hrs.
Burning rate per cell per hr. = 13·89 cwts.
Burning rate on whole plant per 24 hrs. = 667 tons.
Electrical units produced = 9,746,609 kW. hrs.*
Electrical units produced per ton of refuse burnt = 374·06.
Average steam pressure per sq. in. = 152·54 lbs.
Average steam superheat = 249·84° F.
Clinker, flue dust, etc., produced = 16,450 tons 15 cwts.
Total residue to refuse burnt = 38 per cent.

During 1932 from refuse alone at these works, 40,200,700 units (kW. hrs.) were generated, of which 35,056,425 were delivered into the mains for distribution about the city.

One week's run by the Glasgow officials gave a total of 875,410 kW. hrs. generated with load factors of 93·5, 98·7, 99·1 and 98·0 for four consecutive days. The load was kept steady at about 6,800 kW. hrs. until the last two days before shutting the plant down for the week-end, when it was increased to 7,000 kW. hrs. Refuse only was the fuel used in all of the above results.

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* Includes allowance for turbine operation, etc.

LIQUID FUELS.

Whereas solid fuels are solely used for external combustion it is possible to employ almost all liquid fuels for either external or internal combustion. On comparing the most important liquid fuel, petroleum with the most important solid fuel, coal, many remarkable advantages of the former over the latter are found to exist. The heat value of petroleum is some 50 per cent. greater than that of average coal, the stowage space being slightly less. Liquid fuels are more convenient to store (places inaccessible to coal may be employed) and are more easily transported than solid fuels. Liquid fuel is almost entirely free from ash. When used for internal combustion the thermal efficiency of liquid fuels is double that of the solid fuels used for power production. Certain liquid fuels can be employed in high speed engines, whereby a higher power per unit weight of engine is obtainable than by any other type of prime mover.

The most important liquid fuels are petroleum and shale oils and coal tar and its products.

Petroleum.

Petroleum occurs in the earth, and is generally found in certain types of sedimentary rock or oil sands. It is usually accompanied by gaseous hydrocarbons. The crude oil is usually brown or black, and the products of different fields vary greatly both in physical and chemical properties.

Petroleum consists almost entirely of carbon and hydrogen. These are present in the form of compounds known as hydrocarbons. Hydrocarbons may be divided into three main groups, known as aliphatic, aromatic and hydroaromatic hydrocarbons. These groups may be subdivided into numerous series, the members of which possess certain definite relationships between the number of carbon and hydrogen atoms present in the molecules. The physical properties of the members of these series generally progress by fairly regular steps. The carbon atoms in the aliphatic hydrocarbons are connected together in the form of chains, and the more important series of this type of hydrocarbons are :—

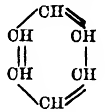
Paraffin Series.
General formula—C_nH_{2n+2}

Methane	CH_4
Ethane	C_2H_6
Propane	C_3H_8
Butane	C_4H_{10}
Pentane	C_5H_{12}

Olefine Series.
General formula—C_nH_{2n}

Ethylene	C_2H_4
Propylene	C_3H_6
Butylene	C_4H_8
Pentylene	C_5H_{10}

The carbon atoms of the aromatic compounds are connected together in the form of rings, a typical compound being :—

Benzene, formula C_6H_6 . . . 

The hydroaromatic compounds also contain carbon rings, but are richer in hydrogen than the aromatic hydrocarbons. A typical hydroaromatic compound is cyclohexane C_6H_{12}.

Petroleum oils contain members of all these series, but the proportion varies according to the locality from which the oil has been obtained. Pennsylvanian oils consist mainly of aliphatic hydrocarbons, Russian oil mainly of hydroaromatic compounds. The products of the destructive distillation of coal consist mainly of aromatic compounds. Crude petroleum from Mexico and South America generally contains quantities of a viscous sulphur-containing body known as asphaltum, which has great influence on their properties as fuels. Little petroleum is consumed in the crude state, it being usual to separate it by distillation into fractions boiling over various ranges. Other processes, as for example cracking, reforming, etc., may also be employed to change the yields and/or the qualities of the fuels obtainable from a given crude oil. Many petroleum fractions are chemically treated in order to remove certain undesirable constituents, such as sulphur, oxygen and nitrogen compounds.

The principal fuels obtained from petroleum are :—

		Specific Gravity.
Gasoline (or petrol)	boiling from 30° C.–150/200° C., 86–302/392° F.	0·700–0·760
Kerosine (or paraffin oil)	boiling from 150° C.–270/300° C. 302°–518/572° F.	0·780–0·830
Gas oils, etc.	boiling from 200° C., 392° F. upwards	about 0·870
Residual fuel oils	boiling range varies greatly.	0·880–over 1·000.

The world production of crude petroleum in 1938 was 272,043,000 metric tons, of which the U.S.A. production was 164,346,000 metric tons or 60·5 per cent. of the world total. The production in 1945 was 2,768,885,000 barrels (about 370,000,000 metric tons) of which the U.S.A. produced about 63 per cent.

GASOLINE (PETROL OR MOTOR SPIRIT).

Gasoline, petrol or motor spirit is a light petroleum fraction used as a fuel in spark ignition engines. Petrol was formerly obtained from crude petroleum solely by fractional distillation and in the early days of motoring the principal requirement was that the fuel should be sufficiently volatile for use in the carburetting devices then employed, and little attention was then given to anti-knocking properties. With the development of more efficient engines employing higher compression ratios the importance of anti-knocking properties became evident and in recent years much research has been performed in order to increase the anti-knock value of motor fuels. Further reference is made to anti-knock values in the section on detonation in petrol engines.

In addition to the production of gasoline from crude petroleum by fractional distillation, other processes have since been developed both with the object of improving the total yield of gasoline obtainable from a given crude oil and also with a view to improving the quality of the product. The principal processes used at the present time are described below.

(1) What is termed 'straight run gasoline' is obtained by fractional distillation, products boiling in the gasoline range being separated from the crude petroleum. The characteristics of gasoline obtained in this manner are largely dependent upon the quality of the crude oil from which they are produced.

(2) Gasoline may be obtained by cracking petroleum products boiling outside the gasoline range. Briefly described, the cracking process consists in heating the raw material to a high temperature, frequently under pressure, whereby the oil charged is decomposed with the formation of both lighter and heavier products. Gasoline can be obtained by cracking, for example, kerosine, gas oil or heavy petroleum residues. Cracking processes may be operated in the liquid or vapour phase, and operations may be conducted in the presence or absence of a catalyst. The characteristics of cracked gasolines are determined more by the type and conditions of the cracking process than by the nature of the raw material. Catalytic cracking processes now play an important part in modern petroleum refining.

(3) Straight run gasolines of low anti-knock value are sometimes subjected to what is known as a 'reforming' process. In effect this is a cracking operation, the object of which is to change the chemical nature of the hydrocarbons composing the straight run gasoline, in order to produce a 'reformed gasoline' of higher anti-knock value. Reforming processes may be applied to all or to only a portion of the products from a given crude oil which boil in the gasoline range. For example, straight run gasoline of low end point may be removed by fractional distillation, heavier distillates being subjected to either reforming or cracking operations depending on the exact procedure which is followed by the refiner.

(4) Extremely volatile spirit may be obtained from the natural gases occuring with crude petroleum by compression and absorption, or by either of these processes. Spirit obtained in this manner is known as 'casinghead gasoline' and usually contains dissolved hydrocarbon gases. The storage and handling of casinghead gasoline frequently involves high evaporation losses, and various stabilising processes have been developed. These stabilising processes usually consist in the fractional distillation of the casinghead gasoline under pressure, in order to remove the major portion of the dissolved gases, and so reduce the vapour pressure of the stabilised product. Casinghead gasoline which has been stabilised in this manner is consequently termed 'stabilised casinghead gasoline.' It may be noted that stabilising processes are also frequently applied to cracked gasoline in order to obtain a finished product of suitable vapour pressure.

(5) Processes are also operated by means of which products boiling in the motor spirit range are obtained by the polymerisation of cracked gases. These products, known as 'polymer gasolines,' are usually characterised by high octane numbers and high octane number blending values, differentiation being made owing to the fact that certain polymerised products have higher values as blending agents for increasing octane number than is indicated by the octane number of the product alone. Polymer gasoline may also be produced from certain gaseous components of natural gas, the processes usually involving pyrolysis or cracking of the natural gas followed by polymerisation of the unsaturated compounds so produced.

Later developments include the production of aviation gasoline blending stocks by processes of selective polymerisation and subsequent hydrogenation, and the 'alkylation' processes for the production of blending stocks by direct combination of selected saturated and unsaturated hydrocarbons.

Modern petroleum gasolines usually consist of mixtures of products obtained by means of the various processes mentioned above.

Petrol has a much higher rate of expansion with increase in temperature than water, the coefficient of expansion varying for different spirits but approximating 0·00063 per ° F. (= 0·001134 per ° C.). The coefficient for correction of the specific gravity is not the same figure as the coefficient of expansion, although related. The official specific gravity/temperature corrections for petroleum products have been standardised by the Institute of Petroleum, and include the following :—

Sp. Gr. 60F./60F.	Factor per °F.	Sp. Gr. 60F./60F.	Factor per °F.
0·6420–0·6530	0·00052	0·7538–0·7649	0·00043
0·6531–0·6649	0·00051	0·7650–0·7760	0·00042
0·6650–0·6775	0·00050	0·7761–0·7869	0·00041
0·6776–0·6899	0·00049	0·7870–0·7988	0·00040
0·6900–0·7025	0·00048	0·7989–0·8124	0·00039
0·7026–0·7166	0·00047	0·8125–0·8283	0·00038
0·7167–0·7300	0·00046	0·8284–0·8599	0·00037
0·7301–0·7424	0·00045	0·8600–0·9250	0·00036
0·7425–0·7537	0·00044	0·9251–1·0249	0·00035

(*Note.*—The above factors apply only when the specific gravity has been determined with glass apparatus referred to water at 60° F.)

The elementary composition of motor spirit is approximately 15 per cent. hydrogen and 85 per cent. carbon. The net calorific value is approximately 18,600 B.Th.U. per lb. of fuel. Theoretically, approximately 15 lbs. of air are required to burn 1 lb. of petrol, and under conditions in which this quantity or a larger quantity of air is present there should be complete combustion with no carbon monoxide in the exhaust. With quantities of air in excess of 15 lbs. free oxygen is present in the exhaust. In actual practice, however, it is found that under practically all conditions there are very considerable quantities of carbon monoxide delivered from the exhaust of petrol engines, in spite of the mixture containing excess air, so that both free oxygen and carbon monoxide are present in the exhaust at the same time.

Maximum thermal efficiency is obtained on a petrol engine with a slight excess of air ; maximum power is obtained very close to the theoretical mixture. One lb. of petrol vapour, occupying 4 cu. ft., requires for its complete combustion approximately 15 lbs. of air occupying 196 cu. ft. at 60° F. ; 16 lbs. of mixture thus occupies 200 cu. ft. at 60° F. and contains 2 per cent. by volume of petrol vapour, that is on the assumption that the petrol is gasified and not merely atomised.

Owing to the high coefficient of expansion of petrol, tanks and closed vessels should never be quite filled with the liquid, so as to avoid risk of rupture following a rise in temperature.

Theoretical Efficiency of Petrol Engines.

The theoretical efficiency, E, of an engine operating on the constant volume cycle is—

$$E = 1 - \left(\frac{1}{R}\right)^{n-1}$$

where,

R = expansion ratio.

If no heat is lost, then $n = 1·45$, but in practice heat is lost during compression, with a result that the exponential n is reduced to the value of 1·35. Ricardo states that when heat leakage, frictional losses, and losses due to friction of gases have been allowed for, then $n = 1·25$, so that the maximum efficiency obtainable in practice may be considered to be

$$E = 1 - \left(\frac{1}{R}\right)^{0·25}$$

Detonation in Petrol Engines.

As will be observed from the formula for thermal efficiency of a petrol engine, it is desirable to use the highest possible compression ratio in order to keep the fuel consumption to a minimum and the power output of the engine to a maximum. The compression ratio, which can be used in a petrol engine is, however, limited by the nature of the fuel employed, and if too high a compression ratio is used the engine 'pinks,' or knocks, when running at heavy loads and slow speeds. The limiting compression ratio for various motor spirits differs considerably. Highly aromatic spirits and cracked spirits will withstand high compressions without signs of detonation or 'pinking.' Straight-run paraffin spirits are very prone to detonation. Certain bodies, particularly lead tetraethyl, possess the property of reducing the tendency to detonate when they are added in very small quantities. Benzole and alcohol possess high anti-knock values and before the war various motor fuels were marketed which contained either or both of these products.

The method of testing detonation at the present time is by matching the spirit in question against a blend of iso-octane and heptane, and the resulting figure is referred to as an 'octane number.' The octane number is equal numerically to the percentage by volume of iso-octane in the heptane-iso-octane mixture which equals the fuel under investigation in tendency to knock, when tested in an engine under standardised conditions. Iso-octane is of high anti-knock value, whilst heptane knocks very freely, consequently a high octane number indicates strongly anti-knock spirit.

The comparison with iso-octane/heptane mixtures is performed on special variable compression engines, the construction of which have been standardised by the Co-operative Fuel Research Committee (C.F.R.). Various testing proceedings have been standardised by official bodies in this country and elsewhere, the procedure to be used being influenced by the type of fuel under test and the information required. The Institute of Petroleum specifies the C.F.R. Motor Method for rating fuels possessing octane numbers below 100, and the 25° Motor Method for obtaining relative ratings on fuels of above 100 octane number. Other methods include the Aviation Fuels Division of the C.F.R. Committee methods 1–C, specified by the Institute of Petroleum for rating aviation fuels under weak mixture or 'cruising' conditions and the A.F.D. 3–C. Method, which may be used for rating such fuels under rich mixture or 'full boost' conditions.

The knock-ratings of a given fuel may vary according to the testing procedure employed, the numerical difference between the results depending on the chemical nature of the fuel tested. It should be noted that relation between the performance of a fuel under the fixed conditions of the various test procedures and the performance of the same fuel under service conditions in engines of other types is not necessarily identical, and considerable experience is therefore required in order to interpret the practical significance of laboratory results.

General.

Various tests are used for the laboratory examination of motor fuels—the majority of those usually employed in Great Britain are described in 'Standard Methods for Testing Petroleum and its Products,' published by the Institute of Petroleum.

Kerosine.

Illuminating oil or kerosine, generally known in this country as paraffin oil, possesses a specific gravity in the neighbourhood of 0·780 to 0·830. It is slightly lower in calorific value than petrol, but on account of its higher specific gravity it yields more heat per unit volume of fuel. The ultimate composition of kerosine is approximately 87 per cent. carbon and 13 per cent. hydrogen. The distillation range is generally between 150° C. and 300° C. For illuminating purposes paraffin-base kerosines are preferable as they burn cleanly without smoking. Kerosine, however, is also used as a fuel in spark-ignition engines fitted with vaporisers. For use in such engines a highly aromatic kerosine is preferable on account of its superior anti-knock qualities, a somewhat lower end point is also desirable, the upper limit of distillation range being for preference in the region of 280° C.

Kerosine will not withstand as high a compression ratio as petrol obtained from the same crude, and this applies throughout the range of fractions of petroleum, namely, the lighter fractions withstand the higher compression ratio or possess the better anti-knock properties.

Gas Oil.

In addition to motor spirit and kerosine a heavier petroleum distillate known as gas oil is also prepared for use as a fuel. Gas oil is a pale yellow or brownish red clear liquid of specific gravity about 0·850 and has a nett calorific value of approximately 18,000 B.Th.U per pound. The oil is used for the preparation of oil gas and considerable quantities are used in gas works for the enrichment of coal-gas. It is also used as a fuel for semi-Diesel and Diesel engines. When gas oil is intended for the manufacture of oil gas in gas works, specifications frequently include a minimum aniline point requirement. This test gives an indication of the chemical constitution of the oil, which in turn affects its suitability for oil gas preparation. Similarly the suitability of a given grade of gas oil for use in high-speed Diesel engines is also affected by the chemical constitution of the oil, and various tests have been devised in recent years for the measurement of what is termed 'ignition quality.' Reference to these tests is made in the Diesel Oil Section.

Residual Fuel Oils.

After the removal of petrol, kerosine, gas oil and possibly lubricating oil distillates from crude petroleum, the residue may consist of bitumen, heavy lubricating oil or residual fuel oil, the nature of the residual product depending on the grade of crude processed and also on the extent of the distilling operation. Other residual fuel oils may be derived from cracking operations, from the solvent extraction of certain residual feed stocks and the like. The characteristics of residual fuel oils vary considerably. Heavy residual fuels are frequently used as boiler oils, while it is possible to prepare a range of fuel and Diesel oils by blending suitable residual fuel oils with gas oil or other light distillate.

Diesel Fuel Oils.

The characteristics of petroleum fuels employed for Diesel engines are varied. For example, the small high-speed Diesel engines used in road vehicles, motor boats, etc., usually employ light gas oil as fuel. Large land and marine Diesel engines, however, may employ somewhat heavier grades of gas oil, residual oil or residual oil blended with distillates. Specifications for Diesel fuel oils frequently include the following tests: closed flash point, hard asphalt, coke, ash, viscosity, water and cold test.

Closed flash point is included as a precaution against fire hazards in transportation and storage. The hard asphalt and coke tests are considered to give an indication of the tendency of the fuel to form deposits in the engine. The maximum permissible percentage of ash is limited in order to prevent excessive wear in engine cylinders and fuel sprayers. Clauses specifying a maximum viscosity and sulphur content and the minimum calorific value are also frequently included in Diesel fuel oil specifications.

When fuel is injected into the combustion chamber of a Diesel engine there is a slight period of delay between the moment of injection and the moment when the fuel commences to burn. This small interval of time is known as the delay period, and varies for different fuels. When used in the same engine the delay period is longer for fuels of low ignition quality than for fuels of high ignition quality.

Largely as a result of the introduction of small high-speed Diesel engines considerable attention has been paid in recent years to the ignition quality of Diesel fuels, and a test has been devised whereby ignition quality is expressed in terms of cetane numbers. The cetane number is the volumetric percentage of cetane in a blend of cetane and alpha methyl naphthalene, which possesses the same ignition characteristics of the fuel under examination. Comparison of the fuel under test with the cetane/alpha methyl naphthalene blends may be performed in a special C.F.R. variable compression Diesel engine or, alternatively, in normal type Diesel engines fitted with special measuring devices. (A.S.T.M. D613—43 (T); I.P.—41/48 (Tentative).)

In addition to the direct determination of ignition quality by matching the performance of the fuel in specially equipped engines against that of blends of standard bodies of known ignition quality, various tests have been proposed in order to estimate ignition quality from a consideration of physical or chemical characteristics. Among these may be mentioned spontaneous ignition temperature, aniline point and Diesel index. Reference is made to the determination of spontaneous ignition temperatures on page 1363 and on the following page a table is given showing the spontaneous ignition temperatures of various bodies. It is considered that the aniline point of petroleum fuels gives an indication of ignition quality in Diesel engines, and at the present time minimum aniline points are included in some Diesel oil specifications. In the specification of the British Standards Institution, for example, it is stated that fuels for use in high-speed Diesel engines should possess aniline points not lower than 60° C. Methods for determining aniline points are specified by the Institute of Petroleum for various types of Diesel fuel, and consist essentially in the determination of the minimum temperature of miscibility of equal volumes of the fuel and freshly redistilled aniline. (I.P. —2/47).

The Diesel Index is an arbitrary figure calculated from the A.P.I., gravity and the aniline point in degrees Fahrenheit (I.P.—21/42). The formula is given below:—

$$\text{Diesel Index} = \frac{\text{Aniline Point °F.} \times \text{A.P.I. Gravity}}{100}$$

It is claimed that Diesel indices calculated in this manner are a convenient guide to the ignition quality of petroleum Diesel fuels. Aniline point and Diesel index, however, are not reliable when applied to other than petroleum fuels or to fuels containing 'dopes' intended to improve the ignition quality.

Various other formulæ have been proposed for the estimation of ignition quality. These include formulæ based on specific gravity and viscosity, specific gravity and surface tension, and A.P.I. gravity and boiling range.

Diesel fuel ignition quality is of particular significance when considering fuels for use in small high-speed Diesel engines such as are employed in road vehicles and motor boats, and the engine tests which have been developed for the measurement of cetane values are similar in general principle to those employed for measuring the octane number of motor spirit. It should be noted, however, that while the maximum compression ratio which can be employed in a petrol engine is limited by the design of the engine and the octane number of the fuel used, the maximum compression ratio of Diesel engines is not similarly limited by the cetane number of the Diesel fuel. Fuel ignition quality thus does not restrict the efficiency and maximum power output of compression ignition engines in the same manner that anti-knock values restrict the performance of spark ignition engines.

Shale Oil.

Shale exists in many parts of the British Empire, the shale oil industry being most fully developed in Scotland. The shale is mined in a similar manner to coal, and is then subjected to destructive distillation, the principal products of value being ammonia and crude shale oil. The crude oil is subjected to distillation, whereby a series of products similar to the petroleum distillates is obtained. The shale oils are richer in unsaturated bodies than petroleum. In fuel properties they very closely resemble the petroleum products.

Coal Tars and Coal Tar Distillates.

When bituminous coal is subjected to destructive distillation, gas, coke, tar and ammonia are obtained. The physical and chemical properties of the tars are dependent upon the type of coal from which they originate, and the carbonising conditions employed. The appliances in use for the carbonisation of coal are mainly gas manufacturing plants and coke manufacturing plants. Certain special plants exist which do not belong to either of these categories.

Gas manufacturing plants may be divided into three classes—horizontal retort, inclined retort, and vertical retort plants. Retorts of the first type yield tars of high specific gravity, 1·15 to 1·25, rich in free carbon, and viscous. Vertical retorts yield thin brown tars of low gravity (about 1·05), which are comparatively low in free carbon content and relatively rich in aliphatic hydrocarbons. The tars from inclined retorts are intermediate between the products of horizontal and vertical plants in their chemical and physical properties.

Coke-oven tars are very variable, but usually approximate to horizontal retort tars in composition.

Relatively small amounts of tar are also produced in low temperature carbonisation processes, in blast furnaces, and in producer and water gas plants.

The majority of coal tars may be employed for boiler firing purposes. Tars from vertical retorts have been found suitable for use on both two and four cycle Diesel engines, providing that certain special adjustments or fittings are employed, large quantities of such tars being used in Diesel engines on the Continent. Some coke-oven tars are suitable for use in Diesel engines. The compositions of several British coal tars are given in the accompanying table :—

Properties of Tars. (*Moore.*)

Description.	Sp.Gr. at 15° C.	Water Per cent.	Chemical Composition. Per cent. C.	H_2.	O_2+N_2	S.	Ash. Per cent.	Coke. Per cent.	Calorific Value (dry tar) cal./g. Gross.	Net.	Free Carbon. Per cent.
Horizontal retort tar	1·180	1·75	91·5	5·2	2·6	0·5	0·2	24·0	9093	8645	18·2
Inclined retort tar	1·157	1·11	89·9	6·0	3·6	0·5	0·02	18·5	9096	8671	14·0
Vertical retort tar	1·089	2·25	88·0	6·8	3·8	0·6	0·03	6·1	9246	8664	1·7
Otto Hilgenstock coke oven tar	1·208	6·00	90·0	5·4	3·8	0·8	0·02	26·8	8921	8624	23·9
Simon-Carvés coke oven tar	1·090	0·50	88·1	5·6	6·1	0·2	0·07	6·0	9695	9261	traces
Chamber oven tar	1·082	1·29	88·2	6·9	4·6	0·3	traces	7·3	9229	8737	3·0
Low temperature carbonisation tar	1·058	3·00	85·8	8·1	5·5	0·9	0·11	8·2	9196	8776	2·2
Water gas tar	1·054	0·59	92·2	6·8	0·4	0·6	traces	18·7	8951	8647	6·8
Blast furnace tar	1·172	3·00	89·5	5·7	0·5	0·84	0·36	23·4	8563	8288	9·5

The lower-boiling hydrocarbons ('benzole') obtained in the carbonisation of coal, (mainly benzene, toluene, and xylenes) are mainly used as fuel for automobile engines. During the war, coal tar fuels, which consisted of mixtures of creosote and medium soft pitch, were extensively used. In the fuels covered by B.S.S. 1469—1948 (C.T.F. 100, etc.), the figures refer to the temperature in Fahrenheit degrees at which the particular fuel has a viscosity at which it is suitable for atomisation. The properties of some miscellaneous liquid fuels are shown in the table on p. 1357.

PROPERTIES OF MISCELLANEOUS LIQUID FUELS.

(*Moore.*)

Description	Sp. Gr. 60° F.	Water Per cent.	Chemical Composition. Per cent. C.	H_2.	O_2+N_2.	S.	Ash Per cent.	Coke Per cent.	Flash Point. ° C.	Viscosity. Redwood Units. 70° F.	100° F.	Soft Asph. Per cent.	Calorific Value. Cal./g. Gross.	Net.
Coal-Tar Creosote	1·053	Trace	89·7	7·4	2·23	0·66	Traces	1·39	93	10·3	8·5	—	9,342	8,964
Blast-Furnace Oil	·907	Nil	82·6	9·9	7·22	0·29	Nil	1·26	70	5·7	—	—	9,992	9,603
Vertical Retort Tar Oil	1·019	0·7	85·8	7·9	5·1	0·5	Traces	1·65	88	9·7	7·0	—	9,189	8,762
Scottish Shale Oil	·945	1·6	87·0	9·5	1·04	2·35	0·11	0·18	56	23·6	12·5	—	9,917	9,245
Premier Tarless Process Oil	·995	Trace	82·6	8·6	8·8	0·50	0·10	1·52	76	17·1	9·4	—	9,472	9,008
Mexican Fuel Oil (Petroleum)	·952	1·00	83·6	10·9	1·5	2·85	0·17	12·2	93	—	267	36·8	10,350	9,734
Mexican Fuel Oil (Petroleum)	·960	0·5	83·0	10·7	2·54	2·8	0·02	—	—	4,800	822	24·6	10,130	9,548
Californian Fuel Oil (Petroleum)	·952	1·9	83·4	11·5	0·49	2·3	0·4	—	46	387	108	11·5	10,110	9,476
Californian Fuel Oil (Petroleum)	·903	9·5	35·1	12·2	1·0	1·4	0·3	—	138	88·5	34·2	—	10,480	9,832
South American Oil	·887	—	86·9	10·9	1·7	0·6	None	—	27	14	9	—	10,740	10,076
Borneo Oil	·939	0·25	88·3	11·3	0·07	0·1	Trace	—	—	63·2	23·1	1·7	10,360	9,712
Persian Oil	·894	—	86·7	11·7	0·13	1·5	0·01	—	93	46·5	15·7	7·6	10,550	9,902
Mid-Continental Oil	·904	None	83·9	12·1	3·63	0·4	Trace	—	110	43·9	—	—	10,815	10,150
" " "	·889	"	86·7	12·5	0·11	0·6	Trace	—	113	—	—	1·5	10,780	10,108
" " Gas Oil	·850	"	85·9	12·8	1·09	0·2	Nil	—	84	—	—	0·4	10,890	10,198
Texas	·936	0·5	86·1	11·0	2·0	0·3	0·1	—	121	—	68	—	10,605	9,989
Assam Oil	·890	Trace	86·4	11·7	1·7	0·2	0·03	—	65	14·0	9·1	—	10,312	9,664
Benzole	·875	—	91·1	8 0	0·0	0·9	None	0·27	—	4·5	3·6	—	9,877	9,461
Mineralised Alcohol (Commercial)	·822	—	48·6	12·7	40·56	0·13	—	—	Below 20	5·2	4·7	—	6,115	5,421

Lignite Tars and Bituminous Coal Tars.

Lignite yields valuable liquid products by destructive distillation. The tar is of a butter-like consistency and possesses a specific gravity of ·860 to ·920. Its sole use in the raw state is as fuel for Diesel engines, for which purpose its richness in aliphatic hydrocarbons makes it behave similarly to petroleum. It therefore does not require either raised cylinder compressions or the use of an ignition oil.

Fuels for petrol engines and heavy oil engines can be obtained from lignite tar by distillation. Their properties are very similar to those of the corresponding fractions obtained from petroleum.

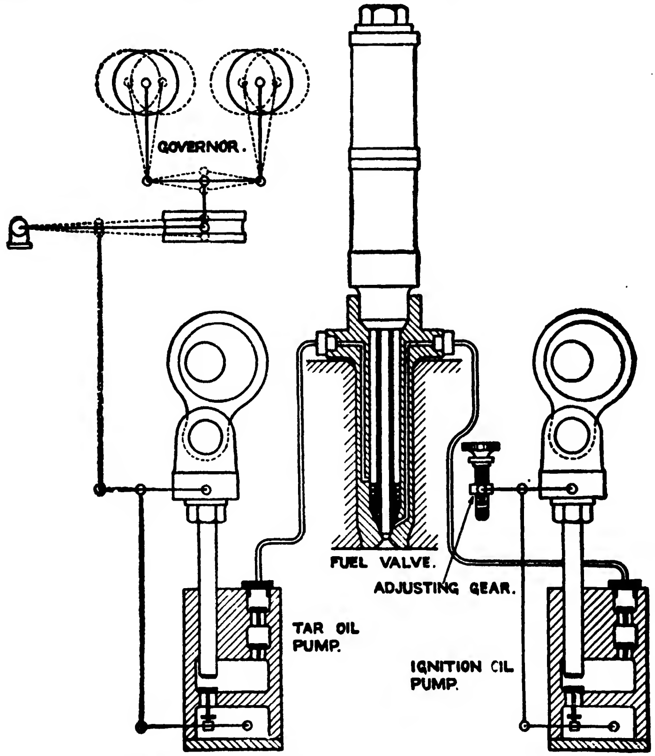


FIG. 5.—Diagrammatic Arrangement of Fuel Pumps for Tar Oil Engine.

Tar oils were used in considerable quantities in Diesel engines in this country during the 1914–18 war on account of the scarcity of petroleum. The difficulties encountered in burning tar oils are mainly due to their comparatively high spontaneous ignition temperature. The ignition of the fuel charge in a Diesel engine is dependent upon the temperature of compression, no special ignition device being provided. With the fuel oils derived from petroleum or shale it has been found that a compression pressure of 450 to 500 lbs. is sufficient to ensure regular ignition under starting conditions and at all loads. Whereas the spontaneous ignition temperature (in atmosphere of oxygen) of petroleum and shale oils is in the neighbourhood of 270° C., this value for tar oils is generally over 400° C.* As a result of the high ignition point engines will not start, or run at

* Moore, *Jour. Soc. Chem. Ind.*, February 15, 1917, p. 109.

low loads, on tar oil unless special means are taken for raising the temperature of the air in the combustion space at the time of fuel injection. The more important methods adopted for this purpose are—(1) to heat the cycle air, blast air and fuel, (2) to increase the compression ratio, (3) to employ an ignition oil, and (4) to inject a small charge of tar oil prior to the main tar oil charge.

Heating of cycle air, blast air and fuel is difficult to effect under starting conditions and has a deleterious action on the fuel valve when running. The increase of compression necessitates a specially heavy design of engine to withstand the higher stresses on liners, cylinder cover, and bearings. To ensure certain starting on tar oils the compression would require to exceed 600 lbs. The starting difficulty may be overcome by running the engine on a petroleum oil before closing down and when starting. If the engines can be always maintained at half load or over, a compression pressure of about 550 lbs. per square inch will give good running.

The introduction of a small charge of a petroleum oil into the working cylinder prior to the main charge of tar oil has been found the most satisfactory method of burning tar oils. The ignition oil charge is pumped to the atomiser nearer the cylinder than the place where the main tar oil charge enters. Thus when the fuel valve opens the ignition oil is first injected into the cylinder and is immediately followed by the tar oil charge. The combustion of the ignition oil heats the air in the cylinder so that the temperature exceeds the spontaneous ignition temperature of the tar oil. The proportion of ignition oil is 6 to 8 per cent. of the total fuel at full load; at other loads the quantity of ignition oil remains practically constant, the proportion increasing to about 50 per cent. at no load. This method has been found to yield satisfactory results at all loads, and under starting conditions, with cylinder compression pressures of 480 to 500 lbs. per square inch. An arrangement of ignition oil gear as given in a paper by Mr. C. Day,* is shown in fig. 5. Devices have been designed to give a small charge of tar oil prior to the main charge of tar oil. The device is so arranged that the initial tar oil charge is not accompanied by any appreciable blast air, the expansion of which would reduce the temperature of the combustion space. Such devices are said to have given good results on the Continent, but are little used in this country.

In addition to the ignition question tar oils are particularly liable to form deposits in the atomisers of Diesel engines. A résumé of some experiences of burning tar oils on Diesel engines was given by Mr. G. Porter in a paper read before the Diesel Engine Users' Association, May 24, 1917. In this paper several specifications for tar oils for use in Diesel engines are given. An abstract of these specifications is appended below :—

Specifications of Tar Oil Fuel for Diesel Engines.

(1) Specification of the M.A.N. Company :—

The tar oil must be a distillate of coal tar.

The oil must flow freely at 61° F., and on being cooled down to 46·5° F. and resting in a place undisturbed by vibration no separation shall take place at this temperature within the space of half an hour.

The following constituents shall not be present in quantities greater than the stated percentages—

Ash	·05 per cent.
Sulphur	1·0 „

Proportions of water and coke residue not stated.

Per cent. insoluble in xylol 0·2.

(2) Continental specification for tar oil quoted by Mr. C. Day in his paper on 'Tars and Tar Oils as Fuel for Diesel Engines' (January 19, 1916):—

Tar oils must not contain more than a trace of constituents insoluble in xylol.

The water content should not exceed 1 per cent.

The coke residue should not exceed 3 per cent.

At least 60 per cent. of the oil should be distilled on heating up to 300° C. (572° F.).

The net calorific value should not be less than 15,840 B.Th.U. per pound.

The open flash point must not be below 65° C. (143·6° F.).

The oil must be quite fluid at 61° F.

* Paper read before the Diesel Engine Users' Association, January 19, 1916.

(3) Specification of tar oil proposed by Mr. C. Day (January 19, 1916) :—

The tar oil must be a product of the distillation of coal tar. No product that has not undergone distillation must be present.

Per cent. insoluble in xylol	Not more than		2·0
,, ash	,,	,,	0·08
,, water	,,	,,	2·5
,, coking residue	,,	,,	3·0

The oil must be liquid at 60° F. when maintained at that temperature for half an hour.

(4) Specification of tar oil by Mr. Batho (the Diesel Engine Users' Association, February 23, 1916) :—

Specific gravity—Between 1·0 and 1·1.
Viscosity—Generally 2° Engler at 5·0°C.
Flash point—100° F. to 130° F.
Colour—One drop on white paper should show no black residue, as is the case with tar.
Lower calorific value—Between 15,800 and 16,500 B.Th.U. per 1 lb.
Ash—Should not exceed 1 per cent. (unburnt residue of tar oil is mostly harmless).
Water—Should not exceed 1 per cent.
Sulphur—0·5 to 1 per cent.

The burning of raw tars on Diesel engines has been accompanied by the same difficulties as the burning of tar oils, but an additional trouble is encountered on account of the free carbon which is always present in raw coal tars.

Furnace Firing with Liquid Fuel.

Liquid fuel possesses many advantages over coal or other solid fuels for furnace firing. The advantages when using heavy petroleum oils may be briefly enumerated as follows :—

(1) Petroleum fuel oil possesses a gross heat value of approximately 19,200 B.Th.U., whilst the gross heat value of coal averages about 14,000 B.Th.U. This shows two pounds of oil are equivalent in heating to nearly three pounds of coal.

(2) In spite of its lower specific gravity, oil can be stored in less space than a corresponding weight of coal.

One ton of coal occupies about 43 cubic feet.
,, oil ,, 40 ,,

For marine purposes oil is conveniently stored in ballast tanks and inaccessible places, which could not be utilised for coal bunkerage.

(3) Oil rarely contains more than 0·1 per cent. of ash, a negligible quantity, while coal may contain as much as 15 per cent. ash. This ash necessitates the removal of clinker from the fires, and besides involving expense it causes a serious loss of heat.

(4) Oil can be conveyed by pipe line, a cheaper, cleaner and more rapid method than any method of coal conveying.

This is of great importance in naval work.

(5) Oil firing does not involve so much manual labour and attendance charges which accompany furnace firing with solid fuel.

(6) A boiler fired with oil will generally raise more steam than the same boiler fired with lump coal. Increases in output up to 30 per cent. have been obtained.

(7) The furnace doors can be kept closed when oil firing, thereby increasing the efficiency of the boiler and lessening the wear and tear on the boiler plates caused by frequently being chilled with cold air, as is the case when burning solid fuel.

(8) Boilers can raise steam much more quickly when oil fired, thus lessening stand-by charges.

(9) Boilers fitted for burning both solid and liquid fuel may be run on coal under normal conditions, while oil can be burned for peak loads, thus minimising the number of boilers in use.

1 lb. coal under favourable conditions raises 9 lbs. steam.
1 lb. oil under similar conditions raises 14½ lbs. steam.

It is essential that firing devices for liquid fuel should break the oil into an exceedingly fine state of division, so that the air can reach the oil readily and ensure speedy combustion.

The methods of breaking up the oil are :—

(1) By means of steam.
(2) By means of compressed air.
(3) By a high pressure on the oil, which is pulverised on passing through a fine jet.
(4) By means of mechanical breaking up in a pulverising chamber.

Steam Jets.

A number of oil-burning installations used on land are operated by steam. Steam is always available at the face of the boiler, and at the same time steam burners, though not giving quite so high an efficiency as air burners, give a soft flame, which is less liable to injure the plates or tubes with which it comes into contact. For marine purposes there is a distinct disadvantage in the use of steam on the oil burners, as it causes the loss of a considerable quantity of water, which has to be carried in storage tanks or else has to be specially prepared by a distillation plant whilst at sea. This drawback does not apply to land installations, where plenty of water is usually available. Steam burners operate on a variety of principles, the majority of constructions being somewhat in the form of an injector.

The Kermode steam jet burners, which have met with much success on land service, are designed to impart a rotary motion to the gases by passing the oil through a helical groove. The steam is passed through an annular space surrounding the oil duct, and in addition to causing atomisation, it is used to induce an air current which assists in the combustion of the fuel.

With the Holden steam jet burner, fuel steam and air are passed through annular spaces, the centre of the burner being left open to facilitate cleaning.

Another principle used for pulverising oil by a steam jet is to allow the oil to trickle from a weir over a thin slit through which the steam is passing. This system is used in the common type of burner known as the 'Mexican trough.' A modification of this principle is also used in the Ajax burner, made by Meldrums Ltd.

Compressed Air Burners.

The use of compressed air for atomising purposes allows a higher efficiency on boiler-firing installations. At the same time the saving is not so great as might be anticipated, as the steam or power used for driving the compressors is almost as great as the amount lost through steam burners. The flame has a higher temperature than that of the steam jet, and greater care must be taken to prevent it impinging on the plates. Some of the most recent types of compressed air burners will operate on air at low pressure (about 10 lbs. per square inch), and therefore do not necessitate costly air compressors, as they can be operated with complete satisfaction by low-pressure air from a rotary blower. Such burners are the 'White' burner, the 'J. Samuel White,' and one of the burners by Alldays & Onions, Ltd.

Low Pressure Air Burners.

Low pressure air at 6 in. to 30 w.g. is supplied by a centrifugal fan to this class of burner, which varies considerably in design. Most of them are of the stationary type, but in some the oil is distributed over the air blast from the lip of a rapidly rotating cup. Typical burners of the rotary class are the 'Rotamiser,' Combustion, Ltd., and the 'Ray,' W. S. Ray Manufacturing Co., while the stationary design is exemplified in the Wallsend, the Laidlaw Drew and the Rotovac burners.

The Oil Pressure System.

This is mainly used for marine purposes, and complete oil-burning installations on this principle are comparatively expensive, on account of the necessity for well-constructed and effective preheaters, filters, and the specially constructed fuel pumps.

In this type of burner the oil is forced through fine jets under a comparatively high pressure. Many burners are so constructed as to impart a rotary motion to the pulverised fuel (*e.g.* Korting burner). The pressure system, although expensive, is probably the most efficient of oil-burning appliances. The power consumed in driving the fuel pumps is almost negligible, and the combustion is excellent. The pressure system has met with much success in marine work and large land installations and invariably is used in gas turbines.

Air Required for Oil Fuel Combustion.

The amount of air required in practice at normal atmospheric temperature for the combustion of 1 lb. of oil fuel, is from 250 to 450 cub. ft.; this is 1·6 to 2·5 times the theoretical amount.

British Admiralty Specification for Oil Fuel for the Navy.

(Abstract.)

'The oil fuel supplied shall consist of liquid hydrocarbons, and may be either (*a*) shale oil or (*b*) petroleum, as may be required, or (*c*) a distillate or a residual product of petroleum, and shall comply with the Admiralty requirements as regards flash-point, fluidity at low temperatures, percentage of sulphur, presence of water, acidity and freedom from impurities.

'The flash-point shall not be lower than 175° Fahr., close test (Abel or Pensky-Martens). The proportion of sulphur contained in the oil shall not exceed 3·00 per cent. The oil fuel supplied shall be as free as possible from acid, and in any case the quantity of acid must not exceed 0·05 per cent., calculated as oleic acid when tested by shaking up the oil with distilled water, and determining by titration with decinormal alkali the amount of acid extracted by the water. methyl orange being used as indicator. The quantity of water delivered with the oil shall not exceed 0·5 per cent.

'The viscosity of the oil supplied shall not exceed 2,000 secs. for an outflow of 50 cubic centimetres at a temperature of 32° Fahr., as determined by Sir Boverton Redwood's standard viscometer (Admiralty type for testing oil fuel). The oil supplied shall be free from earthy, carbonaceous, or fibrous matter, or other impurities which are likely to choke the burners.'

SPECIFICATIONS FOR BRITISH STANDARD FUEL OILS FOR HEAVY-OIL ENGINES.*

No. 209, 1937.

	Marine and Industrial Diesel Fuel.	Heavy Diesel Fuel.
—	For engines where the speed does not exceed about 800 r.p.m., *e.g.* industrial units and main and auxiliary machinery for marine purposes.	For large engines where the speed does not exceed 250 r.p.m., and where means can, if necessary, be provided for heating and cleaning the fuel, *e.g.* large engines for stationary and marine installations where oil engines for auxiliary purposes are not required to use the same fuel as in the main engines.
Flash point (closed)	Minimum 150° F.	Minimum 150° F.
Hard asphalt	Maximum 2·0 per cent.	Maximum 4·0 per cent.
Ash content	„ 0·03 per cent.	„ 0·10 per cent.
Viscosity (Redwood No. 1) at 100° F.	„ 60 secs.†	„ 750 secs.
Water content	„ 0·5 per cent.	„ 1·0 per cent.
Pour point	„ 30° F.‡	(It is considered undesirable to specify limits to the pour point.)
Conradson carbon	Maximum 3·0 per cent.	Maximum 8·0 per cent.
Sulphur content	„ 2·0 per cent.	—
Aniline point§	Minimum 45° C.	—
Gross calorific value B.Th.U./lb.	„ 18,750	Minimum 18,250

British Admiralty Specification for Oil Fuel for use in Diesel Engines.

'Quality:—The oil fuel supplied shall consist of liquid hydrocarbons, and may be either (*a*) shale oil, or (*b*) petroleum, as may be required, or (*c*) a distillate, or a residual product of petroleum, and shall comply with the Admiralty requirements as regards flash-point, fluidity at low temperatures, percentage of sulphur, presence of water, acidity and freedom from impurities.

'The flash-point shall not be lower than 175° Fahr., close test (Abel or Pensky-Martens). (This compares with a flash-point of 200° Fahr. in 1910.)

'The proportion of sulphur contained in the oil shall not exceed 3·00 per cent. (as against 0·75 in 1910).

* By permission of the British Standards Institution.
† The maximum viscosity permitted under this Specification shall be increased to 100 secs. Redwood No. 1 at 100° F. whenever the Conradson carbon does not exceed 2 per cent. and the hard asphalt does not exceed 1·5 per cent.
‡ This limit is intended for temperate climates only.
§ Tentative test for ignition quality (for fuel of petroleum origin only), pending the development of an engine test.

'The oil fuel supplied shall be as free as possible from acid, and in any case the quantity of acid must not exceed 0·05 per cent., calculated as oleic acid when tested by shaking up the oil with distilled water, and determining by titration with decinormal alkali the amount of acid extracted by the water, methyl orange being used as indicator. (In 1910 it was required that the oil should be free from acidity.)

'The quantity of water delivered with the oil shall not exceed 0·5 per cent.

'The viscosity of the oil supplied shall not exceed 2,000 secs. for an outflow of 50 cubic centimetres at a temperature of 32° Fahr., as determined by Sir Boverton Redwood's standard viscometer (Admiralty type for testing oil fuel).

'The oil supplied shall be free from earthy, carbonaceous, or fibrous matter, or other impurities which are likely to choke the burners.

'The oil shall, if required by the inspecting officer, be strained by being pumped on disch from the tanks, or tank steamer, through filters of wire gauze having 16 meshes to the ir

'The quality and kind of oil supplied shall be fully described. The original sourc which the oil has been obtained shall be stated in detail, as well as the treatment to whic been subjected and the place at which it has been treated. The ratio which the oil bears to the original crude oil should also be stated as a percentage.'

Special Oil Mixtures used for External Combustion

Petroleum-Tar Mixtures.

The addition of tar or tar products to heavy petroleum oils causes a thick of pitch to separate, and this has till recently prevented the use of such mixture has been overcome by the use of emulsifying agents which are capable c constituents in a state of emulsion for prolonged periods.

Coal Dust/Petroleum Mixtures.

The U.S. Navy authorities have met with success in preparing fuel coal dust mixed with petroleum fuel oil. The coal is prevented from emulsifying agents. Such oils are claimed to have given satisfactic U.S. warships.

More recently similar trials with powdered coal/petroleum fuel formed by one of the large British steamship companies and it is un being continued.

Spontaneous Ignition Tempe

When a fuel is mixed with air or oxygen there is a certa sufficient to produce combustion. This temperature is refer temperature.

Experimental determination of spontaneous ignition te conditions under which the experiments are conducted. T fuel are those determined by Moore in 1917, using an igr with a crucible-shaped cavity into which a stream of p introduced in the form of drops. The conditions are su at which ignition can take place. A certain definite do does actually take place. The results obtained by Ha

Other experimenters have made investigations in is subjected to heat is considerably reduced so as to the figures obtained are not a definite constant but shorter exposure giving higher temperatures. Mc which allow any length of exposure which will liquid fuels are as follows:—

TEMPERATURES OF SPONTANEOUS IGNITION.*

Description.	Specific Gravity.	Spontaneous Ignition Temperature in Oxygen.	Spontaneous Ignition Temperature in Air.
Petroleum distillates.		°C.	°C.
Pratts Perfection Spirit No. 1	0·710	272	383
Petrol (Mex)	0·718	279	361
Pratts Spirit No. 2	0·724	270	371
Taxibus Spirit (A.A.O. Co., Ltd.)	0·729	272	390
Paraffin oil from A.A.O. Co., Ltd.	0·807	251	—
Petrolite kerosene	0·814	251·5	432
Empire paraffin	0·782	253	395
Petrol from Anglo-American	0·735	—	392
amp oil from Anglo-American	0·787	—	367
s oil (A.A.O. Co., Ltd.)	—	254	358
leum (crude and residue).			
de petroleum (Egypt)	0·851	260	—
oi oil (Assam)	0·890	261	384
-Persian Oil Co.'s oil	0·894	254	408
petroleum (Texas)	0·895	256	387
American fuel oil	0·900	269	430
lexican oil	0·908	259·5	417
troleum (Texas)	0·936	268·5	416
" (Borneo)	0·939	269	380
uel oil	0·948	259·5	424
oleum (Mexico)	0·949	258	425
" (Trinidad)	0·950	274	424
(California)	0·952	264	—
etroleum	0·955	275	429
m (California)	0·961	262	420
Broxburn Oil Co.)	0·768	253	333
Broxburn Oil Co.)	0·803	251	322
	0·860	484	—
	0·863	516	—
	0·875	566	—
il	0·992	349	—
n & Holden)	1·010	415	—
(Stainsby & Lyons)	1·036	473	—
on Carves)	1·046	478	—
perature carbonis-	0·987	307	508
s Works)	1·074	464	—
	1·077	415	—
Heywood Gas	1·114	445	—
ockport Gas	1·123	454	—
	1·132	494	—
	1·140	488	—
vens)	1·145	495	—
Works)	1·172	498	—
Co.)	—	410	—
	0·817	395	518
	0·842	275	275
ting			
	0·875	265·5	405
el	0·894	265·5	401·0
	0·921	273	470
	0·730	190	347
	—	245	—
	—	402	—
	—	348	—
	—	260	—

quid Fuels for Internal Combustion Engines.'

CALORIFIC VALUES, ETC., OF SOME SOLID, LIQUID, AND GASEOUS FUELS.

Fuel. 1 lb. of :—	Sp. Gr. (H = 1).	Burnt to :—	Combustion per Lb. of Fuel. Lb. of Oxygen needed.	Lb. of Air needed.	Weight of Products in Lb. Burnt in Oxygen.	Burnt in Air.	Lbs. of Nitrogen.	Heat of Combustion in B.Th.U. per Lb. Higher Value.	Lower Value.
Column :—	1.	2.	3.	4.	5.	6.	7.	8.	9.
Hydrogen . . .	1	H_2O	8·0	34·80	9·0	35·80	26·80	61,500	52,800
Marsh gas . . .	8	H_2O & CO_2	4·0	17·40	2·75 CO_2 2·25 H_2O 5·00 Total	18·40	13·40	24,000	21,820
Carbon	—	CO	1·334	5·80	2·334	6·80	4·466	4,400	4,400
Carbon	—	CO_2	2·667	11·60	3·667	12·60	8·933	14,600	14,600
tylene . . .	13	H_2O & CO_2	3·077	13·385	3·3846 CO_2 0·6924 H_2O 4·077 Total	14·385	10·308	21,850	21,180
.	14	CO_2	0·571	2·484	1·571	3·484	1·913	4,370	4,370
		.O & CO_2	3·4285	14·914	3·143 CO_2 1·285 H_2O 4·428 Total	15·914	11·486	21,900	20,660
					1·913 CO_2 1·174 H_2O 3·087 Total	10·078	6·991	12,420	11,285
						14·385	10·308	18,000	17,330

GASEOUS FUELS.

The more important gaseous fuels used commercially are generally mixtures. The combustible ·es found in such mixtures are hydrogen, carbon monoxide together with the three hydro- ns—methane, ethylene, and acetylene—also higher hydrocarbons of the series of which these Petromples, together with benzene vapour, and occasionally other hydrocarbons.

Prat
Petr
Prat

Calorific Values, etc., of Gaseous Fuels.

Tar' table, p. 1365, the calorific values per lb. of some fuels are given, together with the weight Fa roducts of combustion, both when burnt in pure oxygen and also when burnt in dry air. Pet ;her value ' is that obtained when the water produced in the combustion is considered as Bm been condensed to the liquid condition, thus yielding up heat at the rate of about 1,055 Pe J. per lb. produced.

ially in calculations relating to internal combustion engines it is found more convenient with the volume rather than the mass of the gaseous mixture; accordingly in the g table the calorific value and weight per cubic foot at N.T.P. and also at 60° F. and rio pressure are given for the usual fuels :—

GASEOUS FUELS—DENSITY AND CALORIFIC VALUE.

ic Foot of Gas.	Weight in Lb. per Cub. Ft.		Calorific Value, B.Th.U. per Cub. Ft. (Satd.).			
	At N.T.P. (Dry).	At 60° F. 1 Atm. (Satd.).	At N.T.P. Higher.	At N.T.P. Lower.	At 60° F. & 1 Atm. Higher.	At 60° F. & 1 Atm. Lower.
un :—	1.	2.	3.	4.	5.	6
3. . . .	0·00563	0·00607	338	285	320	270
3 . . .	0·01177	9·01217	1,052	946	995	895
. . .	0·07243	0·06823	1,557	1,505	1,173	1,423
. . .	0·07820	0·07351	336	336	318	318
. . .	0·07829	0·07363	1,650	1,543	1,560	1,460

O_2 at N.T.P. weigls 0·12298 lb.; in its formation from free carbon and evolved.

nd 1 atm. one cubic foot of CO_2 weighs 0 11634 lb., and in its formation

the volumes of oxygen and of air just necessary for the complete com- of gas are given, together with the change of volume of the products at evolved per cubic foot of the mixture of gas and air before

N AND OF AIR NECESSARY PER CUBIC FOOT OF GAS, AND HEAT PER CUBIC FOOT.

Vol. of Mixture before Combustion in Cub. Ft. With Oxygen.	Vol. of Mixture before Combustion in Cub. Ft. With Air.	Cubic Feet of Nitrogen in Mixture.	Ratio of Vol. of Nitrogen to whole Volume.	Vol. of Mixture after Combustion in Cub. Ft. With Oxygen.	Vol. of Mixture after Combustion in Cub. Ft. With Air.	Ratio of Final to Initial Volume.	B.Th.U. evolved per Cub. Ft. of Initial Volume with Air at 60° F. & 1 Atm.
	4.	5.	6.	7.	8.	9.†	10.*
	3·4	1·9	·56	1·0	2·9	·853	79·5
	10·6	7·6	·716	3·0	10·6	1·000	84·3
	13·0	9·5	·731	3·0	12·5	·962	109·5
	3·4	1·9	·56	1·0	2·9	·853	93·6
	5·4	11·4	·740	4·0	10·6	1·00	94·8

* Lower values.

Hydrogen.

From the above table it will be noted that a hydrogen-air mixture contracts after combustion to only 0·853 of its initial volume, the pressure of the explosion being correspondingly reduced. Further, notwithstanding the very high calorific value of hydrogen *per pound*, its excessive bulkiness results in a heat evolution of only 79·5 B.Th.U. *per cubic foot* of its admixture with air at 60° F. and atmospheric pressure. Hydrogen-air mixtures are also very sharply explosive and trouble is experienced in engines from pre-ignition at the compression pressures usual in practice when hydrogen is present in large proportion in a gaseous fuel. Thus it is clear that hydrogen alone would be a very troublesome fuel to employ in the conventional gas engine, though its presence is often beneficial in conferring sufficiently prompt inflammability upon dilute mixtures of other gases. Special engines, however, such as the Erren engine, have been designed to operate on hydrogen. For heating purposes a large hydrogen content is advantageous on account of the high temperature obtained in its combustion. Hydrogen constitutes about 45 per cent. by volume of modern coal-gas, on the average.

Methane.

Methane is an excellent fuel for power purposes; it has a greater heat value per cubic foot of air mixture than hydrogen, and the volume of the products of combustion is equal to that of the mixture before combustion.

The 'natural gas' so largely used in the United States consists principally of methane, that of Olean (N.Y.), for example, containing no less than 96½ per cent. of this constituent. The gas usually issues from the wells at high pressure, occasionally approaching 1,000 lbs. per sq. in. The Snow engine of the N.Y. & Penn. Traction Co. at Olean is supplied by natural gas issuing at a pressure of 200 lbs. per sq. in., which is reduced down to 1·8 lb. per sq. in. before being admitted, mixed with air, to the engine. With natural gas compression pressures of 150 lbs. per sq. in. are common, and mean effective pressures of 80 lbs. per sq. in. are often attained.

Methane obtained from sewage disposal plants is compressed and transferred to high pressure cylinders for use as a fuel for transport vehicles by a number of local authorities in Great Britain. Research and development work on liquid methane as a fuel has also been carried out.*

Acetylene.

***Acetylene* has so far been but little used as a fuel for internal combustion engines. It is very violently explosive when mixed with air, and owing to its endothermic nature the pressure developed is unusually high.**

The range of explosive mixtures with air is also considerable; Grover found that at atmospheric pressure explosive mixtures were produced with one volume of acetylene to from 4 to 18 volumes of air; at a compression of three atmospheres even a 30 : 1 air-acetylene mixture was found to be explosive.

Acetylene is soluble in water to the extent of 1·1 volume; in alcohol to about 6 volumes; while acetone at 60° F. and atmospheric pressure absorbs about twenty-five times its volume of the gas, and at a pressure of about 180 lbs. per sq. in., about 300 volumes. A solution of acetylene in acetone contained in a porous matrix under a pressure of 150 lbs. per sq. in., is sold commercially as dissolved acetylene ('D. A.'); the 'D. A.' cylinders contain 100 times their own volume of acetylene at 60°, F. and ten atmospheres. Commercial calcium carbide on addition of water usually yields about 4½ cubic feet of acetylene per lb.

If bubbled through petrol, acetylene gas becomes heavily 'carburetted'; 100 cubic feet of acetylene thus treated will produce about 150 cubic feet of 'carburetted' acetylene. According to Dr. Caro, as a heating agent this exceeds pure acetylene about in the ratio of 3 : 2.

Used as a fuel for internal combustion engines some remarkable results have been obtained with acetylene. Some early tests on a 2 h.p. engine showed that only 8½ cub. ft. of the gas were required per b.h.p. hour, as against 35 cub. ft. of coal gas. In another case 6·35 cub. ft. of acetylene yielded 1 b.h.p. hour. And with a 6 h.p. engine using a 20 : 1 mixture, and a compression pressure of 115 lbs. per sq. in., Ouinat found that only 6·26 cub. ft. of acetylene were consumed per b.h.p. hour, the consumption by volume of coal-gas being about three times as great. Specially designed acetylene engines of from 1 to 10 b.h.p. are built; these are mainly employed in pumping water for the generators of large village installations. The average consumption is about 7 cub. ft. of acetylene per b.h.p. hour.

Carbon Monoxide.

***Carbon Monoxide* yields the large return of 93·6 B.Th.U. per cubic foot of mixture with air, notwithstanding its low calorific value per lb.; this results from its weight being fourteen times that of hydrogen. The products of combustion occupy, however, only 0·835 of the volume of the original mixture; it is the principal combustible constituent of producer and blast furnace gas. Carbon monoxide is extremely poisonous, forming with the red corpuscles of the blood a stable chemical compound of a bright red colour termed carboxyhæmoglobin; care must always be taken to prevent any leakage of the gas into the atmosphere, and all engine and producer houses should be freely ventilated.**

* 'The Significance of Liquid Methane as a Fuel,' A. G. Egerton, F.R.S. and M. Pearce, B.Sc., A.C.G.I., *Journal, Institute of Fuel*, August 1945.

Ethylene.

Ethylene is usually present to a small extent (up to about 1 per cent. by volume) in coal gas, and is a valuable producer of heat.

GASEOUS FUELS EMPLOYED IN GAS ENGINES.

1. Natural Gas.
2. Coal-Gas.
3. Coke-oven Gas.
4. Acetylene Gas.
5. Producer Gas.
6. Blast Furnace Gas.

1. NATURAL GAS.

Natural Gas.—Large quantities of natural gas are evolved in most petroleum oilfields. The composition of the gas varies in different localities, but it always contains a high percentage of methane. The net calorific power is usually 750 to 950 B.Th.U. per cubic foot. It is used in gas engines and for heating and illuminating purposes.

2. COAL-GAS.

Coal-Gas is still almost exclusively used in the smaller sized stationary gas engines, especially in towns; it is an admirable fuel for internal combustion engines of any size, however large, but its use for considerable powers is usually precluded by its relatively high cost.

Owing to the restriction of petrol supplies during the 1914–18 war many trials were made on the use of coal-gas as a fuel for the engines of motor vehicles. Large gas-bags were attached to cars, vans and coaches and good operation was obtained. The distance travelled on the one filling of the gas-bag was about 15 miles and materially limited the radius of action. Later, however, development work was carried out on the use of coal-gas carried in special high-pressure cylinders. With this arrangement the distance travelled per charge was increased to about 60 miles. Extensive use of this method has been made in Germany although gases of higher calorific value than coal-gas are generally used.

Calorific Value of Coal-Gas.

Since the introduction of the Gas Regulation Act in 1920 the gas undertakings in Great Britain ceased to operate on official standards of candle power and are now subject to penalties in default of maintaining their respective standards of calorific power. The choice of calorific value of the gas which is distributed is left in the hands of individual undertakings, thus a wide range of quality is found throughout the country. When utilised in gas engines the factor of calorific value plays an important part, so that the operating data and adjustment of the engine must depend largely on the quality of the gas supply in any particular district. In general, it may be said that the calorific value of the gas supplied by undertakings in the country varies between 470 and 500 B.Th.U. per cubic foot, but there are a few outstanding exceptions, as, for example, the South Metropolitan Gas Company of London which distributes a gas of 560 B.Th.U. per cubic foot, and the Nuneaton Gas Company whose standard is 380 B.Th.U.

CALORIFIC STANDARDS.

	Calorific value of gas.
London (Gaslight & Coke Company)	500 B.Th.U. per cu. ft.
,, (South Metropolitan Gas Company)	560 ,, ,, ,,
,, (Commercial Gas Company)	500 ,, ,, ,,
,, (Wandsworth Gas Company)	470 ,, ,, ,,
,, (South Suburban Gas Company)	500 ,, ,, ,,
,, (Tottenham Gas Company)	500 ,, ,, ,,
Manchester	450 ,, ,, ,,
Liverpool	475 ,, ,, ,,
Birmingham	500 ,, ,, ,,
Bristol	480 ,, ,, ,,

So far as the composition of towns' gas is concerned, it is not possible to lay down any arbitrary figures for the reason that the ultimate analysis varies not only in different localities, but it varies to some degree even in the same locality in accordance with the proportion of water-gas which is added

of the mixture of coal-gas and water-gas distributed. As a general guide, however, the analysis given below represents the composition of a normal straight coal-gas having a calorific value of 520 B.Th.U. per cubic foot.

Composition of 520 B.Th.U. Coal-Gas.

	Per cent. by Volume
Carbon dioxide	3·5
Unsaturated hydrocarbons	3·5
Oxygen	0·3
Carbon monoxide	7·7
Hydrogen	47·0
Methane	27·5
Nitrogen	10·5
	100·0

The gas supplied to-day by nearly all the principal undertakings consists, however, of a mixture of from 75 to 80 per cent. of straight coal-gas (of above analysis) and 20 to 25 per cent. of either blue or carburetted water-gas. In this case the typical composition will be represented by the following analysis:—

Composition of Mixed Gas consisting of 80 per cent. Straight Coal-Gas and 20 per cent. Blue Water-Gas. (480 B.Th.U.)

	Per cent. by Volume.
Carbon dioxide	3·7
Unsaturated hydrocarbons	2·8
Oxygen	0·2
Carbon monoxide	13·8
Hydrogen	48·0
Methane	22·2
Nitrogen	9·3
	100·0

It is important for the gas-engine user to bear in mind that in recent years it has been established that one B.Th.U. will always give the same service irrespective of the quality or concentration of the mixture. In other words, from the point of view of efficiency a poorer mixture (say, 400 B.Th.U.) can be utilised as efficiently as a rich one (say, 550 B.Th.U.) so long, of course, that combustion is always complete. Proper adjustment depends, therefore, on the variation of the air supply in accordance with the quality of the gas in any particular locality. The volume of air theoretically just necessary for complete combustion may be estimated by aid of the second table on page 1366.

Increased economy in consumption is obtained by using mixtures of gas with such excess of air, or other gases, that only about 50 B.Th.U. are introduced per cub. ft. of piston displacement for cylinders up to about 20 ins. diameter, while for larger cylinders (30 ins. dia. and above) the figure must be reduced to 40 or even 35 B.Th.U. in order to avoid overheating troubles.

When excess of oxygen is present in the mixture, practically no carbon monoxide is found in the exhaust; with deficiency of oxygen, or irregularity in the mixture, the contrary is to be expected.

Usually in practice the volume ratio of mixture employed is from 7 to 10 of air to 1 of coal-gas; the richer mixture gives a greater power output, the more dilute a greater economy in consumption of gas per horse-power hour.

3. COKE-OVEN GAS.

Coke-Oven Gas is produced in large quantities in the process of the manufacture of metallurgical coke; its composition varies somewhat, but in general, when debenzolised, it has similar characteristics to coal-gas made at the gas works.

The practice of stripping coke-oven gas of benzol by means of scrubbers is common and provides a small but none the less welcome supply of home produced light spirit.

About half the gas evolvd is used for heating the ovens and the remainder is used for industrial heating or sold as towns gas.

A typical analysis of debenzolised coke-oven gas is as follows :—

	Per cent. by Volume.
Oxygen	0·4
Carbon dioxide	2·0
Unsaturated hydrocarbons	2·6
Carbon monoxide	7·4
Hydrogen	54·0
Methane	28·0
Nitrogen	5·6
	100·0

The higher calorific value of this gas is 525 B.Th.U. per cu. ft.

4. ACETYLENE GAS.

Experiments carried out in Germany with acetylene gas for internal combustion engines showed that the proportion of air to gas should be in the ratio of 12·02 to 1 to avoid the formation of soot. The maximum compression which may be used for a mixture of acetylene and air is about three atmospheres, compared with 4–5 atmospheres for petrol and 10 atmospheres for coal-gas. One ton of carbide is sufficient to produce 10,000 cubic feet of acetylene.

5. PRODUCER GAS.

In normal combustion, when an excess of air is present in the combustion space, the carbon of the fuel is burnt to form carbon dioxide according to the reaction

$$C + O_2 \rightarrow CO_2.$$

Nitrogen from the air passes through the combustion space into the resulting gas without entering into the reaction. By this reaction one pound of carbon generates 14,647 B.Th.U.

By restricting the air supply and arranging to maintain a comparatively great depth of fuel upon the grate the carbon may be burnt into carbon monoxide according to the reaction

$$2C + O_2 \rightarrow 2CO,$$

nitrogen again passing through the grate without being affected.

By this latter reaction, 4,400 B.Th.U. are generated for each pound of carbon gasified, the residual heat value obtainable from the carbon by complete combustion, *i.e.* 14,647 − 4,400 = 10,247 B.Th.U., remaining in the resultant gas.

The greatest quantity of heat obtainable in the form of cool carbon monoxide from the air-carbon reaction is therefore only $\frac{10,247}{14,647} = 70$ per cent. of the heat originally available in the carbon.

Of the 30 per cent. of heat given out during the reaction, a large portion leaves the producer in the form of sensible heat.

The heat generated in the producer is sufficient to cause a temperature too high for satisfactory working.

In order to cool the producer and to utilise this excess of heat, steam is introduced into the combustion space, when, if conditions are favourable, the following reaction takes place:

$$C + H_2O \rightarrow CO + H_2$$

For each pound of fuel entering into the reaction 4,320 B.Th.U. *are absorbed* from the heated fuel.

By the use of air and steam in certain proportions, it is possible to absorb the excess heat generated by the air reaction and maintain a practically constant temperature in the producer.

By alternately blowing the producer with an excess of air and raising its temperature by the combustion of carbon, either to the monoxide or to the dioxide, and then cutting off the air supply and passing steam into the hot fuel until the producer temperature has again fallen, it is possible to generate water-gas (the result of the steam-carbon reaction) which may be collected separately.

Water-gas in conjunction with oil-gas (generated by the cracking of heavy petroleum distillates) is largely used for mixing with coal-gas in normal times.

As the fuel used is not pure carbon, other considerations than those mentioned in the above theoretical outline affect producer design and working. As coals undergo a destructive distillation in the producers it is necessary to make use of special devices for the removal of tar from the producer gas. The recovery of ammonia generated from the nitrogen of the coal may be

COMPOSITION OF PRODUCER GASES WITH HEAT VALUES, ETC.

Description of Producer Gas.	Kind of Fuel used.	Per Cent. Composition by Volume.						B.Th.U. per Cu. Ft. (lower) at 60°F. (15·5°C.) and 1 Atm.	Cu. Ft. of Gas made per Ton of Fuel.	Cu. Ft. of Air required per Cu. Ft. of Gas.	B.Th.U. evolved per Cu. Ft. of Mixture.	Per Cent. Thermal Efficiency of Producer on lower value of Gas.
		H_2.	CH_4.	CO.	N_2.	CO_2.	O_2.					
Column :—	1.	2.	3.	4.	5.	6.	7.	8.	9.	10.	11.	12.
Dowson Pressure	Anthracite	19·8	1·3	23·8	48·8	6·3	0·0	150·7	171,500	1·41	60·0	82
Dowson Suction	Anthracite	15·7	1·8	22·0	54·5	6·0	0·0	137·5	180,000	1·02	61·8	80
Dowson Suction	Gas Coke	13·0	0·55	25·4	55·65	5·4	0·0	130·0	168,000	0·95	63·3	75
Mond 1. Pressure	Non-Caking Bituminous Coal	22·65	3·5	16·05	44·55	13·25	0·0	147·3	147,500	1·26	65·2	60·4*
Mond 2. Pressure	Ditto	16·60	3·35	27·3	47·5	5·25	0·0	165·4	138,250	1·38	69·6	71·5*
Mond 3. Pressure	Ditto	17·36	1·20	25·55	49·82	5·77	0·30	142	190,000	1·13	66·7	—

* Including steam for blowing engine and washers.

1. With ammonia recovery and steam saturation temp. 175°F. (79·4°C.). 2. Ditto, and 140°F. (60·0°C.). 3. Without ammonia recovery.

of some importance in the economics of producer gas plants. These particulars are treated in greater detail in the following descriptions of various types of plant.

(a) Pressure producer gas from anthracite (Dowson).
(b) „ „ „ coke („)
(c) Suction and suction-pressure producer gas from anthracite (Dowson).
(d) „ „ „ „ wood, sawdust, chips (Dowson).
(e) „ „ „ „ charcoal (Dowson).
(f) „ „ „ „ coke and anthracite (Dowson).
(g) „ „ „ „ coke (Dowson).
(h) „ „ „ „ bituminous coal (Dowson).
(i) Pressure producer gas from bituminous coal (Mond)
(j) Suction and suction-pressure producer gas from anthracite (Mond).
(k) „ „ „ „ coke („)
(l) „ „ „ „ charcoal („)
(m) Suction-pressure producer gas from wood, semi-bituminous („)
(n) Suction producer gas from bituminous coal (Farnham).
(o) Producer gas from other substances, as colliery refuse, sawdust, wood chips, cocoanut shells, belt pickings, lignite, and peat.

In the table on page 1371 some results of analyses are given of producer gas from anthracite, coke, and bituminous coal. The number of cu. ft. of air necessary for just complete combustion is obtained by aid of table on page 1366 from the formula :—

$$\text{Cu. ft. of air} = \frac{4\cdot 8}{100}\left\{\tfrac{1}{2}(H_2 + CO) + 2\,CH_4 - O_2\right\}$$

WASTE HEAT RECOVERY.

The directions in which profitable recovery of waste heat must be looked for are

(i) Waste heat from various metallurgical and other furnaces;

(ii) Waste gases from internal combustion engines;

(iii) Exhaust steam from non-condensing engines, steam hammers, steam-hydraulic intensifiers, etc.

Waste heat from high temperature metallurgical operations, such as open-hearth melting furnaces, consists of the sensible heat of the waste gases and the heat carried away by radiation and convection from the furnace walls. The latter is permitted deliberately in order to prevent unnecessarily rapid burning of the walls. The heat of the waste gases is generally utilised for the purpose of preheating the incoming air and gas by means of regenerators and reversing valves. By jacketing the furnace walls and utilising the heat formerly wasted in radiation for air preheating, thus releasing approximately 50 per cent. of the heat of the waste gases for steam-raising in a waste heat boiler, a very substantial saving of fuel will result. The loss of heat by radiation and convection from unjacketed furnaces is certainly not less than 25 per cent. of the total heat of the fuel used. Even in a reheating furnace working at 1000° C., jacketing of the furnaces instead of regeneration in chequer chambers will often give an air temperature of over 300° C.

Small reheating furnaces for light forgings and similar work air-jacketed at the sides only will often give a temperature of over 150° C. in the air supplied for combustion. In all cases the result will be that the final temperature of the gases will be sufficient to make a waste heat boiler a profitable investment.

In metallurgical furnaces wherein the process requires a densely smoky atmosphere and the air supply is deliberately restricted, the use of a waste heat boiler for the products of combustion together with a supplementary coal fire, through which an excess of air is passed, will not only lead to large recovery of heat in the form of steam, but will also furnish a solution to the smoke-abatement problem. Without the supplementary fire-grate a supply of highly preheated air brought through ducts on the furnace settings will usually ensure reignition of the combustible matter in the smoky gases. Cold air will not do this owing to the low temperature of the gases to begin with, thus keeping the products below ignition point.

For ordinary waste heat boiler work without supplementary firing a temperature of at least 800° C. is needed to make the proposition profitable. With tubular boilers of the Cochran type the usual evaporation per pound of coal consumed in the furnace is :—

Temperature of Waste Gases.	1500° F.	1800° F.	2000° F.
Lb. water per lb. of coal from water at 62° F.	3·08	4·25	5·00

Gases from internal combustion engines usually allowed to expand in a silencer, with corresponding fall of temperature, are maintained at smaller volume and part with their heat to the water in the boiler with a final temperature of about 75° C.

This system is of considerable value also in dealing with the waste heat from gas retort furnaces.

Where low-temperature furnaces are in use only the waste gases may be used for heating of water for shop and office heating by means of economisers, and where both high- and low-temperature furnaces are worked, the hot water from the latter should be used for feed to the high-temperature waste heat boilers. An interchange of heat and power from waste sources between adjoining establishments is often a practicable proposition where the means of power and heat generation and utilisation are collectively equal.

The heat wasted in exhaust steam can be applied to the drive of low or mixed pressure turbines, which latter will have the additional economical effect of levelling out the load on the boiler plant, in that a falling-off of the supply of exhaust steam will cause the turbine to draw on the boilers directly for a high pressure supply. Other important advantages are the returning of soft water to the boilers for feeding purposes, either with or without preliminary electrolytic treatment for oil elimination.

Other applications of the heat of exhaust steam are for heating processes in manufacture, heating of buildings, etc., all of which permit of the return of the condensate for boiler feed purposes.

All proposals for the utilisation of waste heat of any kind require to be dealt with individually on their merits, the conditions being so variable that no hard and fast rules can be laid down which will be universally applicable.

See also Descriptive Section XXV.

SECTION XXVI

ELECTRICAL ENGINEERING

ELECTRICAL UNITS AND NOMENCLATURE — RESISTANCE — STANDARD VOLTAGES — STANDARD COPPER AND ALUMINIUM CONDUCTORS — FUSES — PORCELAIN INSULATORS — WIRING FITTINGS — REGULATIONS OF ELECTRICITY COMMISSIONERS — ELECTRICAL EQUIPMENT OF BUILDINGS, SHIPS AND AIRCRAFT — ELECTRICITY REGULATIONS FOR FACTORIES — ACCUMULATORS — ELECTRICAL MACHINERY AND MERCURY-ARC RECTIFIERS — ELECTRIC MOTORS — CARBON BRUSHES — SWITCHGEAR — STARTERS AND CONTROLLERS — TRANSFORMERS — INDUCTION REGULATORS — POWER FACTOR IMPROVEMENT—INSTRUMENTS — METERS — ELECTRICITY SUPPLY (METERS) ACT, 1936 — LIGHTNING CONDUCTORS — HEATING AND COOKING — AGRICULTURAL APPLICATIONS — INSULATING MATERIALS — PHOTO-ELECTRIC CELL USES—ELECTRIC PROPULSION—ELECTRICAL EQUIPMENT OF AUTOMOBILES — SYNCHRONOUS CLOCKS.

(pp. 1377-1529)

(Revised by J. M. Burnett, B.A., A.M.I.E.E.)

SECTION XXVI

ELECTRICAL ENGINEERING

ELECTRICAL UNITS AND NOMENCLATURE — RESISTANCE — STANDARD VOLTAGES — STANDARD COPPER AND ALUMINIUM CONDUCTORS — FUSES — PORCELAIN INSULATORS — WIRING FITTINGS — REGULATIONS OF ELECTRICITY COMMISSIONERS — ELECTRICAL EQUIPMENT OF BUILDINGS, SHIPS AND AIRCRAFT — ELECTRICITY REGULATIONS FOR FACTORIES — ACCUMULATORS — ELECTRICAL MACHINERY AND MERCURY-ARC RECTIFIERS — ELECTRIC MOTORS — CARBON BRUSHES — SWITCHGEAR — STARTERS AND CONTROLLERS — TRANSFORMERS — INDUCTION REGULATORS — POWER FACTOR IMPROVEMENT — INSTRUMENTS — METERS — ELECTRICITY SUPPLY (METERS) ACT, 1936—LIGHTNING CONDUCTORS—HEATING AND COOKING—AGRICULTURAL APPLICATIONS — INSULATING MATERIALS — PHOTO-ELECTRIC CELL USES — ELECTRIC PROPULSION — ELECTRIC EQUIPMENT OF AUTOMOBILES — SYNCHRONOUS CLOCKS.

(Revised by J. M. Burnett, B.A., A.M.I.E.E.)

Electrical Units and Nomenclature.

B.S. No. 560—1934. *Engineering Symbols and Abbreviations Rules:* (a) Particular values of varying quantities to be represented by small letters. Subscript letters may be used. (b) Effective (R.M.S.) or constant values of electrical quantities to be represented by capital letters. (c) Maximum values of varying quantities to be indicated by subscript 'max' or by small letter with circumflex accent. (d) Angles to be represented by small Greek letters. (e) Numerical constants to be indicated by small script or Roman letters; specific quantities or their ratios by small Greek letters.

Name of Quantity.	Symbol.	Name of Quantity.	Symbol.
Length, Mass, Time	l, m, t	Difference of potential	V
Acceleration of gravity	g	Electromotive force	E
Work	A	Current	I
Energy	W	Magnetic field-strength	H
Power	P	Magnetic flux	Φ
Efficiency	η	Magnetic flux-density	B
Rotational speed	n	Magnetomotive force	F
Temperature	θ or τ	Reluctance	S
Frequency	f	Permeability	μ
Wave-length, Period	λ, T	Resistance, Resistivity	R, ρ
Angular velocity	ω	Capacitance, Permittivity	C, κ
Phase displacement	ϕ	Self-, Mutual-inductance	L, M
Quantity of electricity	Q	Reactance, Impedance	X, Z

Symbols for names of Electrical Units to be employed only after numerical values:—

Name of Unit.	Symbol.	Name of Unit.	Symbol.
Ampere	A	Watt-hour	Wh
Volt	V	Volt-ampere	VA
Ohm	Ω	Ampere-hour	Ah
Coulomb	C	Milliampere	mA
Joule	J	Kilowatt	kW
Watt	W	Kilovolt-ampere	kVA
Farad	F	Kilowatt-hour	kWh
Henry	H	Microfarad	μF
Volt-coulomb	VC	Megohm	MΩ

B.S. No. 447—1932 refers to Graphical Symbols for use in connection with Interior Electrical Installations and No. 108—1933 for General Electrical Purposes.

B.S. No. 205—1943 is a Glossary of Terms used in Electrical Engineering.

One Ampere (international) is the unvarying electric current which, when passed through a solution of nitrate of silver in water, deposits silver at the rate of 0·001118 gramme per second.

One Coulomb is the quantity of electricity represented by 1 ampere passing for 1 second.

One Ohm (international) is the resistance to an unvarying electric current of a column of mercury of uniform cross-section and length equal to 106·3 cm. whose mass is 14·4521 grammes at the temperature of melting ice.

One Volt (international) is the electrical pressure which, when steadily applied to a conductor whose resistance is one international ohm, will produce a current of one international ampere.

One Farad is that capacity which is charged to a potential of 1 volt by a current of 1 ampere flowing for 1 second.

One Henry is the inductance of a circuit in which an E.M.F. of 1 volt is induced by a current variation of 1 ampere per second.

One Watt is the power, or the rate at which work is being done, when a current of 1 ampere passes through a resistance of 1 ohm.

One Joule is the amount of work done, or heat generated, by 1 ampere flowing for 1 second through a resistance of 1 ohm.

The following multiples and sub-multiples of these units are in common use :—

One millivolt	= one-thousandth of a volt.	One microhm	= one-millionth of an ohm.
One milliampere	= one-thousandth of an ampere.	One microfarad	= one-millionth of a farad.
		One millihenry	= one-thousandth of a henry.
One megohm	= one million ohms.	One kilowatt	= one thousand watts.

The following relations will be found useful :—

One horse-power is equal to 746 watts.

One foot-pound is equal to 1·356 joules = 0·323 calorie.

One joule is equal to 0·238 calorie.

One (gm.) calorie is the work done in raising 1 gramme of water 1° C., = 4·2 joules.

One (kg.) calorie = 1000 (gm.) calorie.

One joule is equal to 0·7373 foot-pound.

One dyne is the force which, acting on a mass of 1 gramme for 1 second, will give it a velocity of 1 cm. per second.

One erg is the work done by 1 dyne acting through 1 centimetre.

One watt = 0·00134 horse-power = 0·7373 ft.-lb. per second = 44·24 ft. lb. per minute = 0·0573 British thermal unit per minute = 1 joule per second.

One kilowatt-hour = 1,000 watt-hours = one Board of Trade Unit = 1·34 horse-power-hours = 2,656,400 ft.-lb. = 3,413 British thermal units = 860,000 calories.

The practical standard of electromotive force is the Weston Normal cell, which has an E.M.F. of 1·0185 volts at 15° C., decreasing by 0·00004 volt per degree C. rise of temperature.

RESISTANCE.

Effect of Temperature on Resistance.

The resistance of metals increases with the temperature in accordance with the approximate formula

$$R_t = R_o (1 + kt)$$

where k is the coefficient given in col. 5 or 6 below (according to the temperature scale employed) divided by 100.

If the resistance R_1 at temperature t_1 in ° C. be known, then for copper the resistance R_2 at temperature t_2 in ° C. will be :—

$$R_2 = R_1 \left(\frac{234 \cdot 5 + t_2}{234 \cdot 5 + t_1}\right)$$

Metallic Resistance Materials.

Metallic resistance materials are classified by the British Standards Institution for maximum temperatures of 60, 200, 300, 700, and 1,000° C. respectively. According to Specification No. 115—1938, the resistivity at 20° C. is not to differ more than 5 per cent. from that declared by the supplier, and the material must be uniform in resistance within 5 per cent. in the case of sheets, strips and tapes, and within from 5 to 2 per cent. in the case of wires ranging in diameter from below 0·001 to above 0·012 inch ; the temperature coefficients in the two low-temperature classes must not exceed 0·00002 and 0·00004 respectively for ten hours without the resistance changing more than 0·01 per cent. Lists of standard sizes of wires, sheets, and tapes are given in the Report.

Resistivity of Pure Annealed Metals and Alloys, at 0° C. (32° F.).
(*Fleming & Dewar.*)

Metal.	Microhms per cm. cube.	Microhms per inch cube.	Ratio to Copper.	Temp. Coefft. F. per cent.	Temp. Coefft. C. per cent.
Aluminium	2·665	1·048	1·68	0·242	0·435
Antimony	36·0	14·17	22·7	0·217	0·39
Arsenic	33·3	13·1	21	—	—
Bismuth	111·0	43·7	69·6	0·195	0·35
Copper	**1·588**	**0·6252**	**1**	**0·237**	**0·426**
Gold	2·20	0·866	1·38	0·209	0·377
Iron	9·07	3·57	5·71	0·347	0·625
Lead	20·4	8·03	12·8	0.228	0·411
Mercury	94·07	37·04	59·2	0·040	0·072
Nickel	12·3	4·84	7·75	0·345	0·62
Platinum	10·92	4·299	6·88	0·204	0·367
Silver	1·47	0·579	0·926	0·222	0·40
Tin	13·05	5·138	8·22	0·244	0·44
Zinc	5·75	2·26	3·62	0·226	0·406
Constantan	51	20	32	±0·0005	±0·001
German Silver	30	11·8	19	0·0152	0·0273
Manganin	46·7	8·4	29·4	0·0008	0·0015
Nichrome	95	37·5	60	0·0239	0·043
Platinoid	38	15	24	0·0122	0·022

STANDARD VOLTAGES.

B.S. No. 77—1947. Voltages for Transmission and Distribution A.C. Systems.

The term 'system' denotes all the conductors and apparatus electrically connected to a common source of voltage ; 'System Voltage,' the voltage between lines for which the system is designed and installed ; 'Declared Voltage,' the voltage at the consumer's terminals declared by the electricity undertaking and subject to maintenance within the limits laid down in the Electricity Commissioners' Supply Regulations.

The preferred standard system voltages in Britain are :—

Voltage to neutral	240 volts
,, between lines	415 ,,
,, ,, ,,	11 kV.
,, ,, ,,	33 ,,
,, ,, ,,	66 ,,
,, ,, ,,	132 ,,
,, ,, ,,	264 ,,

Other standard systems which are in use are :-

Voltage to neutral	. . . .	230 volts*
,, between lines	. . . .	400 ,, *
,, ,, ,,	. . . .	3·3 kV. †
,, ,, ,,	. . . .	6·6 ,, †
,, ,, ,,	. . . .	22 ,, †
,, ,, ,,	. . . .	88 ,,
,, ,, ,,	. . . .	110 ,,
,, ,, ,,	. . . .	165 ,,
,, ,, ,,	. . . .	220 ,,

STANDARD CONDUCTORS.

The following specifications have been issued for copper and aluminium conductors :—

No. 7—1946. Insulated rubber copper conductors.
,, 125—1947. Copper conductors for power lines.
, 446—1932. Braided copper cables for power lines.
,, 215—1934. Aluminium conductors for power lines.
,, 156—1943. Enamelled copper wire.
,, 115—1938. Metallic resistances.
,, 128—1929. Bare copper wire for machines.
,, 444—1932. Bare soft copper bars for machines.
,, 445—1932. Commutator bars.
,, 23—1933. Trolley and contact wire.
,, 159—1932. Brushes and busbar connections.
,, 480—1942. Metal-sheathed, paper-insulated copper conductors.
,, 518—1933. Medium-hard copper strip bars and rods.
,, 608—1943. Varnished cambric-insulated annealed copper conductors for electricity supply.
,, 672—1936. Cadmium-copper conductors for power lines.

COPPER CONDUCTORS.

B.S. No. 7—1939. Relates to rubber-insulated copper cables.
,, ,, 480—1942. Relates to lead-covered paper-insulated copper cables.
,, ,, 608—1935. Relates to varnished cambric-insulated copper cables.
,, ,, 760—1938. Relates to metal-sheathed paper-insulated copper cables for use in mines.

The International Standards of Resistance for Copper are as follows:—At 20° C. (68° F.), the resistance of a wire of standard annealed copper 1 metre long and of 1 sq. millimetre cross-section is $\frac{1}{58}$ = (0·017241) ohm. The density of such copper at 20° C. is 8·89 grams per cub. centimetre, and the temperature coefficient of resistance is 0·00393 per degree C. At 60° F. (15·56° C.) the resistance of a solid conductor of standard annealed copper 1,000 yards long and of 1 sq. in. cross-section is 0·0240079 ohm; the density is 8·892015, and the temperature coefficient of resistance is 0·0022221 per degree F.

At 15·5° C. (60° F.), the resistance of a copper conductor is accurately given by the expression :

Resistance in microhms = 8 × length in feet/cross section in square inches.

For the purposes of the specifications an increase of 2 per cent. in the length of each wire in the stranded conductor has been assumed, and the area of the stranded conductor has been taken to be that of the solid wire which has the same resistance. In the case of multicore cables, an increase of 2 per cent. has been taken, to allow for the laying-up of the cores.

Tolerances allowed on solid conductors are, in weight, 3 per cent.; in resistance, 3 per cent. plain, 4 per cent. tinned (0·036 in. diameter upwards), 5 per cent. (below 0·036 in.). On stranded wires, in weight 2 per cent.; in resistance, 2 per cent. plain, 3 per cent. tinned (0·036 in. diameter upwards), 4 per cent. (below 0·036 in.).

The resistance of hard-drawn copper is about 3 per cent. higher than that of annealed copper conductors.

The specifications also give tables of sizes, resistances and weights of standard solid and stranded circular copper conductors, flexible cords and cables. Details of standard thickness of dielectric for paper, rubber and vulcanised bitumen insulated cables the sizes and thickness of lead and tough rubber coverings, galvanised steel wire and tape for armouring are tabulated for cables up to 20,000 volts. Pressure tests are specified together with the method of procedure. An appendix contains diagrams showing sections of single and multicore cables.

* The standard low-voltage system before 1947 was 400/230 volts.
† Subsidiary non-preferred standards in Britain.

TABLE I.—STANDARDS FOR SOLID AND STRANDED CIRCULAR COPPER CONDUCTORS.* (I.E.E. Wiring Regulations—11th ed.)

Standard Sizes.		Number and Diameter of Wires.	Overall Diameter of Conductor.	Weight per 1,000 yards.	Resistance per 1,000 yards at 60° F.		
Nominal Area.	Calculated Area.				Standard.	Maximum for Plain Wires.	Maximum for Tinned Wires.
Sq. in.	Sq. in.	Inch.	Inch.	Lbs.	Ohms.	Ohms.	Ohms.
0·0015	0·001521	1/·044	0·044	17·58	15·79	16·26	16·42
0·002	0·001943	3/·029	0·062	23·37	12·36	12·61	12·85
0·003	0·002994	3/·036	0·078	36·02	8·019	8·180	8·260
0·0045	0·004546	7/·029	0·087	54·39	5·281	5·387	5·493
0·007	0·007005	7/·036	0·108	83·81	3·427	3·496	3·530
0·01	0·01046	7/·044	0·132	125·2	2·294	2·340	2·363
0·0145	0·01462	7/·052	0·156	174·9	1·643	1·675	1·692
0·0225	0·02214	7/·064	0·192	264·9	1·084	1·106	1·117
0·03	0·02840	19/·044	0·220	340·4	0·8468	0·8637	0·8721
0·04	0·03960	19/·052	0·260	475·5	0·6063	0·6184	0·6244
0·06	0·05999	19/·064	0·320	720·3	0·4002	0·4082	0·4122
0·1	0·1009	19/·083	0·415	1211	0·2380	0·2427	0·2451
0·15	0·1478	37/·072	0·504	1776	0·1625	0·1657	0·1673
0·2	0·1964	37/·083	0·581	2360	0·1223	0·1247	0·1259
0·3	0·3024	37/·103	0·721	3635	0·07939	0·08098	0·08177
0·4	0·4064	61/·093	0·837	4886	0·05908	0·06026	0·06085
0·5	0·4985	61/·103	0·927	5994	0·04816	0·04913	0·04961
0·75	0·7435	91/·103	1·133	8942	0·03229	0·03294	0·03326
1·0	1·0376	127/·103	1·339	12481	0·02314	0·02360	0·02383

* For comparative Tables of Old and New Standard Sizes, see p. 1382.

Table II.—Comparative Table of Old and New Standard Sizes.

New Standard.		Old Standard.		New Standard.		Old Standard.	
New Nominal Area in Sq. Inches.	Number and Diameter in inches of Wires.	Number and Gauge of Diameter in inches of Wires.	Old Nominal Area in Sq. Inches.	New Nominal Area in Sq. Inches.	Number and Diameter in inches of Wires.	Number and Gauge or Diameter in inches of Wires.	Old Nominal Area in Sq. Inches.
0·001	1/·036*	1/20 S.W.G.	0·0010	—	—	7/14 S.W.G.	0·0346
0·0015	1/·044	—	—	0·04	19/·052	—	—
—	—	1/18 ,,	0·0018	—	—	19/17 ,,	0·0459
—	—	3/22 ,,	0·0018	0·06	19/·064	19/16 ,,	0·0600
0·002	3/·029	—	—	0·075	19/·072*	19/15 ,,	0·0750
—	—	7/25 ,,	0·0022	—	—	19/14 ,,	0·0937
0·003	3/·036	3/20 ,,	0·0030	0·1	19/·083	—	—
—	—	7/23 ,,	0·0031	0·12	37/·064*	37/16 ,,	0·1168
0·003	1/·064*	1/16 ,,	0·0032	—	—	19/13 ,,	0·1250
—	—	7/22 ,,	0·0042	0·15	37/·072	37/15 ,,	0·1500
0·0045	7/·029	—	—	—	—	37/14 ,,	0·1824
—	—	7/21½ ,,	0·0049	0·2	37/·083	37/·083"	0·2000
0·007	7/·036	7/20 ,,	0·0070	0·25	37/·093*	37/·092"	0·2500
—	—	7/19 ,,	0·0086	0·3	37/·103	37/·104"	0·3000
0·01	7/·044	—	—	0·4	61/·093	61/·092"	0·4000
—	—	7/18 ,,	0·0125	0·5	61/·103	61/·104"	0·5000
0·0145	7/·052	—	—	0·6	91/·093*	61/·112"	0·6000
—	—	7/17 ,,	0·0170	0·75	91/·103	91/·101"	0·7500
0·0225	7/·064	7/16 ,,	0·0220	0·85	127/·093*	—	—
0·03	19/·044	—	—	1·0	127/·103	127/·101"	1·0000
—	—	19/18 ,,	0·0338				

* These sizes are now obsolete.

Flexible Cables.

A flexible cable is one in which the conductor (or conductors) exceeds 0·007 sq. in. in cross-section, and comprises a number of wires, the diameter of the wires and the material of the dielectric being such as to ensure flexibility.

Table III.—Dimensions, Current Rating and Resistance of Flexible Cables.*

Nominal Area in sq. in.	Number and Diameter of Wires comprising Conductor. Diameter.				Current Rating for V.I.R. Cables.		Standard Resistance in Ohms per 1,000 yards at 60° F. (15·6° C.).
	0·010 in.	0·012 in.	0·018 in.	0·029 in.	Two-Conductor.	Three-Conductor.	
					amps.	amps.	
0·01	140/·010	97/·012†	—	—	30	27	2·29
0·0145	195/·010	—	60/·018†	—	36	32	1·64
0·0225	296/·010	—	91/·018†	—	45	39	1·08
0·03	—	266/·012	117/·018†	—	52	46	0·847
0·04	—	368/·012	163/·018†	—	62	55	0·606
0·06	—	557/·012	248/·018†	—	82	71	0·400
0·1	—	—	416/·018	160/·029†	118	103	0·238
0·15	—	—	610/·018	235/·029†	151	132	0·163
0·2	—	—	810/·018	312/·029†	183	160	0·122
0·3	—	—	1248/·018	481/·029	238	—	0·0794
0·4	—	—	1677/·018	646/·029	286	—	0·0591
0·5	—	—	2057/·018	792/·029	330	—	0·0482

* I.E.E. Wiring Regulations—11th ed. † For trailing cables and similar purposes.

Flexible Cords.

A flexible cord is a flexible cable of cross-section not exceeding 0·007 sq. in.

Table IV.—Dimensions and Resistance of Flexible Cords.*

Nominal Cross-sectional Area.	Number of 0·0076-in. Diameter Wires comprising Conductor.	Current Rating in Amps. (subject to Voltage Drop) for Twin Cords.	Resistance in Ohms per 1,000† Yards at 60° F. (15·6° C.) Standard.	Maximum for Tinned Wires.	Maximum Permissible Weight supported by Twin Cord.
Sq. In.	Inch.		Ohms.	Ohms.	Lb.
0·0006	14/·0076	2	39·7	41·3	3
0·001	23/·0076	3	24·2	25·1	5
0·0017	40/·0076	5	13·9	14·4	10
0·003	70/·0076	10	7·94	8·26	10
0·0048	110/·0076	15	5·05	5·25	10
0·007	162/·0076	20	3·43	3·57	10

A flexible earth wire is not regarded as a conductor for the purpose of the above table.

* **I.E.E. Wiring Regulations—Eleventh Ed.**
† An allowance must be made for the extra length due to laying up.

TABLE V.—V.I.R. CABLES AND PAPER CABLES. 1/·044 TO 7/·029 INS.

Current rating (subject to voltage drop) for vulcanised-rubber-insulated or impregnated-paper-insulated cables * run :—

(i) Bunched, and enclosed in one conduit, troughing, or casing (col. 3 or col. 5 according to the type and number so run);

(ii) Bunched, and open (col. 3 or col. 5 according to the type and number so run);

(iii) Separated, and open (col. 3 only).

Conductor.		Not more than :—Four Single-core Cables, or Two Twin (or Concentric) Cables, or One Three-core Cable.		Not more than :—Eight Single-core Cables, or Four Twin (or Concentric) Cables, or Two Three-core Cables.	
Nominal Cross-sectional Area.	Number and Diameter (in.) of Wires.	Current Rating (subject to Voltage Drop) for D.C., or Single-phase or 3-phase A.C.	Approximate Length in Circuit for 1-volt Drop with Current Rating in Col. 3 :—*Lead plus Return*, for D.C., or Single-phase A.C.; *Lead only*, for balanced 3-phase A.C.	Current Rating (subject to Voltage Drop), for D.C., or Single-phase or 3-phase A.C.	Approximate Length in Circuit for 1-volt Drop with Current Rating in Col. 5 :—*Lead plus Return*, for D.C., or Single-phase A.C.; *Lead only*, for balanced 3-phase A.C.
1.	2.	3.	4.	5.	6.
Sq. In.		Amps.	Ft.	Amps.	Ft.
0·0015	1/·044	5	36	5	36
0·002	3/·029	5	47	5	47
0·003	3/·036	10	35	8†	42
0·0045	7/·029	15	34	12†	42

NOTE.—Table V applies to cables employed in the wiring of buildings, but does not apply to every condition under which cables may be used. (Braided vulcanised-rubber-insulated cables run open are required under Regulation 403 to be spaced on insulators.)

In conditions of abnormally high ambient air temperature, the Notes to Tables VI and X should be consulted for vulcanised-rubber-insulated and impregnated-paper-insulated cables respectively.

The lower limit set to the size of conductor by the permissible voltage drop is dealt with in Regulation 304, p. 1433.

* Including tough-rubber-protected cables and lead-covered cables, but excluding (for use with alternating current) single-core cables armoured with wire or tape of magnetic material and such ferrous-sheathed cables as are prohibited under Regulation 308.

† These figures (8 and 12) may be increased to 9 and 13·5 amps. respectively, where a diversity factor can properly be applied to the circuit which feeds the cables forming the group of final sub-circuits.

I.E.E. Wiring Regulations—Eleventh Ed.

Table VI.—V.I.R. Cables. 7/·036 to 127/·103 ins. (See also Table VII.)

Current rating (subject to voltage drop) for vulcanised-rubber-insulated Cables* run :—

(i) Bunched, and enclosed in one conduit, troughing, or casing (cols. 3 and 4 or col. 7, according to the type and number so run);

(ii) Bunched, and open (cols. 3 and 4 or col. 7, according to the type and number so run).

Conductor.		Not more than :—Two Single-core Cables.				Nor more than :—Four Single-core Cables, One or Two Twin Cables, or One Concentric Cable.		
		Current Rating (subject to Voltage Drop).		Approximate Length in Circuit (*Lead plus Return*) for 1-volt Drop with Rating in Col. 3 or Col. 4.		Current Rating (subject to Voltage Drop).	Approximate Length in Circuit for 1-volt Drop with Current Rating in Col. 7.	
Nominal Cross-sectional Area.	Number and Diameter (in.) of Wires.	D.C.	Single-phase A.C.	D.C.	Single-phase A.C.	D.C., or Single-phase or 3-phase A.C.	*Lead plus Return*, for D.C.	*Lead plus Return*, for Single-phase; *Lead only*, for balanced 3-phase.
1.	2.	3.	4.	5.	6.	7.	8.	9.
Sq. In.		Amps.	Amps.	Ft.	Ft.	Amps.	Ft.	Ft.
0·007	7/·036	29	29	27	27	23	34	34
0·01	7/·044	38	38	32	32	30	41	41
0·0145	7/·052	46	46	37	37	36	46	46
0·0225	7/·064	56	56	45	45	45	57	57
0·03	19/·044	65	65	50	49	52	61	61
0·04	19/·052	78	78	58	57	62	73	73
0·06	19/·064	102	102	67	65	82	84	84
0·1	19/·083	147	147	79	74	118	98	96
0·15	37/·072	189	189	90	78	151	113	103
0·2	37/·083	229	229	99	79	183	124	103
0·3	37/·103	298	298	117	78	238	146	103
0·4	61/·093	358	358	130	72	286	162	96
0·5	61/·103	413	413	138	66	330	173	86
0·75	91/·103	575	530	149	55	—	—	—
1·0	127/·103	740	648	161	47	—	—	—

* Including tough-rubber-protected cables and lead-covered cables, but excluding (for use with A.C.) such of the following as are prohibited under Regulation 308:—(*a*) Single-core armoured or ferrous-sheathed cables; (*b*) Single-core cables above 0·2 sq. in. encased in brass, copper, etc.

Note.—Table VI applies to cables employed in the wiring of buildings, but does not apply to every condition under which cables may be used. (Braided vulcanised-rubber-insulated cables run open are required under Regulation 403 to be spaced on insulators.)

Table VI refers to situations where the ambient air temperature does not exceed 90° F. (32·2° C.). Where the ambient air temperature is abnormally high, see note to Table IX.
The permissible voltage drop is dealt with in Regulation 304, p. 1433.

I.E.E. Wiring Regulations—Eleventh Ed.

TABLE VII.—V.I.R. CABLES. 7/·036 TO 37/·083 INS. (See also Table VI.)

Current rating (subject to voltage drop) for vulcanised-rubber-insulated cables * run :—

(i) Bunched, and enclosed in one conduit, troughing, or casing (col. 3 or col. 6 according to the type and number so run).

(ii) Bunched, and open (col. 3 or col. 6 according to the type and number so run).

Conductor.		Not more than Six Single-core Cables, or Three Twin Cables, or One Three-core or Four-core Cable, or Two Concentric Cables.			Not more than Ten Single-core Cables, or Five Twin Cables, or Two Three-core or Four-core Cables, or Three Concentric Cables.		
		Current Rating (subject to Voltage Drop).	Approximate Length in Circuit for 1-volt Drop with Current Rating in Col. 3.		Current Rating (subject to Voltage Drop).	Approximate Length in Circuit for 1-volt Drop with Current Rating in Col. 6.	
Nominal Cross-sectional Area.	Number and Diameter (in.) of Wires.	D.C., or Single-phase or 3-phase A.C.	*Lead plus Return*, for D.C.	*Lead plus Return*, for Single-phase *Lead only*, for balanced 3-phase	D.C., or Single-phase or 3-phase A.C.	*Lead plus Return*, for D.C.	*Lead plus Return*, *for* Single-phase *Lead only*, for balanced 3-phase
1.	2.	3.	4.	5.	6.	7.	8.
Sq. In.		Amps.	Ft.	Ft.	Amps.	Ft.	Ft.
0·007	7/·036	20	40	40	17	45	45
0·01	7/·044	27	45	45	23	54	54
0·0145	7/·052	32	52	52	27	61	61
0·0225	7/·064	39	65	65	34	76	76
0·03	19/·044	46	70	70	39	84	84
0·04	19/·052	55	82	82	47	95	95
0·06	19/·064	71	97	97	61	113	113
0·1	19/·083	103	113	110	88	131	129
0·15	37/·072	132	128	118	113	150	137
0·2	37/·083	160	142	118	—	—	—

* Including tough-rubber-protected cables and lead-covered cables, but excluding (for use with A.C.) such of the following as are prohibited under Regulation 308 :—(*a*) Single-core armoured or ferrous-sheathed cables; (*b*) Single-core cables above 0·2 sq. in. encased in brass, copper, etc.

NOTE.—Table VII applies to cables employed in the wiring of buildings, but does not apply to every condition under which cables may be used. (Braided vulcanised-rubber-insulated cables run open are required under Regulation 403 to be spaced on insulators.)

Table VII refers to situations where the ambient air temperature does not exceed 90° F. (32·2° C.). Where the ambient air temperature is abnormally high, see note to Table IX.
The permissible voltage drop is dealt with in Regulation 304, p. 1433.

I.E.E. Wiring Regulations—Eleventh Ed.

TABLE VIII.—V.I.R. BRAIDED CABLES ON CLEATS. 19/·083 TO 127/·103 INS.

Current rating (subject to voltage drop) for single-core, unarmoured, vulcanised-rubber-insulated, braided and compounded cables * (with or without tape) run open on cleats as defined on p. 1391.

Conductor.		Current Rating (subject to Voltage Drop) for Cables run under the conditions defined on p. 1391.			Approximate Length in Circuit for 1-volt Drop.		
Nominal Cross-sectional Area.	Number and Diameter (in.) of Wires.	D.C.	Single-phase A.C.	Three-phase A.C.	*Lead plus Return, for D.C.*, with Current Rating in Col. 3.	*Lead plus Return*, for Single-phase A.C., with Current Rating in Col. 4.	*Lead only*, for balanced 3-phase A.C., with Current Rating in Col. 5.
1.	2.	3.	4.	5.	6.	7.	8.
Sq. In.		Amps.	Amps.	Amps.	Ft.	Ft.	Ft.
0·1	19/·083	172	172	170	67	61	61
0·15	37/·072	219	219	216	77	64	64
0·2	37/·083	262	262	259	85	65	65
0·3	37/·103	342	342	338	101	59	60
0·4	61/·093	425	423	410	110	47	48
0·5	61/·103	490	485	470	116	43	45
0·75	91/·103	647	610	585	132	39	41
1·0	127/·103	785	697	669	152	39	40

NOTE.—Table VIII applies to cables employed in the wiring of buildings, but does not apply to every condition under which cables may be used.

Table VIII applies to two or three cables run spaced as shown on p. 1392. Where four or more cables are so spaced the current ratings are reduced to 90 per cent. of those set out in col. 3 or col. 4 above for direct-current or alternating-current (either single-phase or three-phase) loading respectively.

For two or three smaller cables (7/·036 to 19/·064 ins. inclusive) so spaced the current ratings are those given in col. 3 or col. 4 of Table VI, and for four or more such smaller cables the current ratings are 90 per cent. of those given in col. 3 or col. 4 of Table VI, for direct-current or alternating-current (either single-phase or three-phase) loading respectively.

Table VIII refers to situations where the ambient air temperature does not exceed 90° F. (32·2° C.). Where the ambient air temperature is abnormally high, see note to Table IX.

The permissible voltage drop is dealt with in Regulation 304, p. 1433.

* Including single-core, unarmoured, tough-rubber-protected cables; but excluding (for use with alternating current) such cables as are prohibited under Regulation 308.

TABLE IX.—V.I.R. LEAD-COVERED CABLES ON CLEATS. 19/·083 TO 127/·103 INS.

Current rating (subject to voltage drop) for single-core, unarmoured, vulcanised-rubber-insulated, lead-covered cables, run open on cleats as defined on p. 1391.

Conductor.		Current Rating (subject to Voltage Drop) for Cables run under the conditions defined on p. 1391.			Approximate Length in Circuit for 1-volt Drop.		
Nominal Cross-sectional Area.	Number and Diameter (in.) of Wires.	D.C.	Single-phase A.C.	Three-phase A.C.	*Lead plus Return*, for D.C., with Current Rating in Col. 3.	*Lead plus Return*, for Single-phase A.C., with Current Rating in Col. 4.	*Lead only*, for balanced 3-phase A.C., with Current Rating in Col. 5.
1.	2.		4.	5.	6.	7.	8.
Sq. In.		Amps.	Amps.	Amps.	Ft.	Ft.	Ft.
0·1	19/·083	160	160	157	72	64	65
0·15	37/·072	207	206	201	81	66	68
0·2	37/·083	251	249	242	89	64	68
0·3	37/·103	320	314	304	108	63	65
0·4	61/·093	402	377	355	116	51	54
0·5	61/·103	458	421	390	124	49	52
0·75	91/·103	580	510	455	147	45	52
1·0	127/·103	701	583	512	170	44	51

NOTE.—Table IX applies to cables employed in the wiring of buildings, but does not apply to every condition under which cables may be used.

Table IX applies to two or three cables run spaced as shown on p. 1392. Where four or more cables are so spaced the current ratings are reduced to 90 per cent. of those set out in col. 3 or col. 4 above for direct-current or alternating current (either single-phase or three-phase) loading respectively.

For two or three smaller cables (7/·036 to 19/·064 ins. inclusive) so spaced the current ratings are those given in col. 3 or col. 4 of Table VI, and for four or more such smaller cables the current ratings are 90 per cent. of those given in col. 3 or col. 4 of Table VI, for direct-current or alternating-current (either single-phase or three-phase) loading respectively.

Table IX refers to situations where the ambient air temperature does not exceed 90° F. (32·2° C.). Where the ambient air temperature is abnormally high the current ratings given in Table IX shall be multiplied, and the lengths for 1-volt drop divided, by the appropriate factor as follows :—

Ambient air temperature . .	95°F.	100°F.	105° F.	110° F.	115° F.
Factor	0·90	0·80	0·69	0·55	0·38

The lower limit set to the size of conductor by the permissible voltage drop is dealt with in Regulation 304, p. 1433.

Table X.—Paper Lead-covered Cables. 7/·036 to 127/·103 Ins. (See also Table XI.)

Current rating for lead-covered cables (with or without armour) insulated with impregnated paper, varnished cambric or impregnated jute, and run :—

(i) Bunched, and enclosed in one troughing or casing (cols. 3 and 4 or col. 7 according to the type and number so run);

(ii) Bunched, and open (cols. 3 and 4 or col. 7, according to the type and number so run).

Conductor.		Not more than :—Two Single-core Cables.				Not more than :—Four Single-core Cables, One or Two Twin Cables, or One Concentric Cable.		
		Current Rating (subject to Voltage Drop).		Approximate Length in Circuit (*Lead plus Return*) for 1-volt Drop with Current Rating in Col. 3 or Col. 4.		Current Rating (subject to Voltage Drop).	Approximate Length in Circuit for 1-volt Drop with Current Rating in Col. 7.	
Nominal Cross-sectional Area.	Number and Diameter (in.) of Wires.	D.C.	Single-phase A.C.	D.C.	D.C. or Single-phase A.C.	Single-phase or 3-phase A.C.	*Lead plus Return*, for D.C.	*Lead plus Return*, for Single phase A.C.; *Lead only*, for balanced 3-phase
1.	2.	3.	4.	5.	6.	7.	8.	9.
Sq. In.		Amps.	Amps.	Ft.	Ft.	Amps.	Amps.	Ft.
0·007	7/·036	33	33	23	23	26	29	29
0·01	7/·044	50	50	23	23	40	29	29
0·0145	7/·052	67	67	24	24	54	30	30
0·0225	7/·064	89	89	27	27	71	34	34
0·03	19/·044	103	103	30	30	82	37	37
0·04	19/·052	123	123	35	35	93	43	43
0·06	19/·064	160	160	40	39	128	50	50
0·1	19/·083	229	229	48	44	183	59	58
0·15	37/·072	295	295	54	47	236	67	62
0·2	37/·083	354	354	60	48	283	75	63
0·3	37/·103	460	460	71	48	368	89	63
0·4	61/·093	555	555	79	44	444	99	58
0·5	61/·103	646	646	84	40	516	105	53
0·75	91/·103	884	803	91	34	—	—	—
1·0	127/·103	1116	952	101	31	—	—	—

Note.—Table X applies to cables employed in the wiring of buildings, but does not apply to every condition under which cables may be used.

Table X refers to situations where the ambient air temperature does not exceed 90° F. (32·2° C.). Where the ambient air temperature is abnormally high the current ratings given in Table X shall be multiplied, and the lengths for 1-volt drop divided, by the factor :—

Ambient air temperature.	95° F.	100° F.	105° F.	110° F.	115° F.	120° F.
Factor	0·96	0·92	0·88	0·84	0·79	0·74
Ambient air temperature.	125° F.	130° F.	135° F.	140° F.	145° F.	150° F.
Factor	0·69	0·63	0·57	0·51	0·43	0·35

The permissible voltage drop is dealt with in Regulation 304, p. 1433.

* Excluding (for use with alternating current) such single-core armoured cables as are prohibited under Regulation 308, p. 1433.

Table XI.—Paper Lead-covered Cables. 7/·036 to 61/·103 Ins. (See also Table X.)

Current rating (subject to voltage drop), for lead-covered cables* (with or without armour) insulated with impregnated paper, varnished cambric, or impregnated jute, and run :—

(i) Bunched, and enclosed in one troughing or casing (col. 3 or col. 6 according to the type and number so run);

(ii) Bunched, and open (col. 3 or col. 6 according to the type and number so run).

Conductor.		Not more than Six Single-core Cables, or Three Twin Cables, or One Three-core or Four-core Cable, or Two Concentric Cables.			Not more than Ten Single-core Cables, or Five Twin Cables, or Two Three-core or Four-core Cables, or Three Concentric Cables.		
		Current Rating (subject to Voltage Drop).	Approximate Length in Circuit for 1-volt Drop with Current Rating in Col. 3.		Current Rating (subject to Voltage Drop).	Approximate Length in Circuit for 1-volt Drop with Current Rating in Col. 6.	
Nominal Cross-sectional Area.	Number and Diameter (in.) of Wires.	D.C., or Single-phase or 3-phase A.C.	*Lead plus Return*, for D.C.	*Lead plus Return*, for Single-phase A.C.; *Lead only*, for balanced 3-phase A.C.	D.C., or Single-phase or 3-phase A.C.	*Lead plus Return*, for D.C.	*Lead plus Return*, for Single-phase A.C.; *Lead only*, for balanced 3-phase A.C.
1.	2.	3.	4.	5.	6.	7.	8.
Sq. In.		Amps.	Ft.	Ft.	Amps.	Ft.	Ft.
0·007	7/·036	23	33	33	20	38	38
0·01	7/·044	35	33	33	30	38	38
0·0145	7/·052	47	34	34	40	40	40
0·0225	7/·064	62	38	38	53	45	45
0·03	19/·044	72	42	42	62	49	49
0·04	19/·052	86	50	50	74	57	57
0·06	19/·064	112	58	58	96	67	67
0·1	19/·083	160	68	66	137	80	78
0·15	37/·072	206	78	71	177	90	82
0·2	37/·083	248	86	72	—	—	—
0·3	37/·103	322	101	70	—	—	—
0·4	61/·093	388	113	65	—	—	—
0·5	61/·103	452	120	59	—	—	—

Note.—Table XI applies to cables employed in the wiring of buildings, but does not apply to every condition under which cables may be used.

Table XI refers to situations where the ambient air temperature does not exceed 90° F. (32·2° C.). Where the ambient air temperature is abnormally high, see note to Table X.

The permissible voltage drop is dealt with in Regulation 304, p. 1433.

* Excluding (for use with alternating current) such single-core armoured cables as are prohibited under Regulation 308, p. 1433.

TABLE XII.—PAPER LEAD-COVERED CABLES ON CLEATS. 19/·083 TO 127/·103 INS.

Current rating (subject to voltage drop) for single-core, unarmoured, lead-covered cables, insulated with impregnated paper, varnished cambric, or impregnated jute, and run open on cleats as defined below.

Conductor.		Current Rating (subject to voltage drop) for Cables run under the conditions defined below.			Approximate Length in Circuit for 1-volt Drop.		
Nominal Cross-sectional Area.	Number and Diameter (in.) of Wires.	D.C.	Single-phase A.C.	Three-phase A.C.	*Lead plus Return*, for D.C., with Current Rating in Col. 3.	*Lead plus Return*, for Single-phase A.C., with Current Rating in Col. 4.	*Lead only*, for balanced 3-phase A.C., with Current Rating in Col. 5.
1.	2.	3.	4.	5.	6.	7.	8.
Sq. In.		Amps.	Amps.	Amps.	Ft.	Ft.	Ft.
0·1	19/·083	262	262	261	42	37	37
0·15	37/·072	340	339	336	47	39	39
0·2	37/·083	412	410	405	52	40	39
0·3	37/·103	530	523	513	61	37	39
0·4	61/·093	630	613	583	70	31	33
0·5	61/·103	739	690	655	73	30	30
0·75	91/·103	952	856	788	84	28	29
1·0	127/·103	1188	1005	898	95	26	29

NOTE.—Table XII applies to cables employed in the wiring of buildings, but does not apply to every condition under which cables may be used.

Table XII applies to two or three cables run spaced as shown below. Where four or more cables are so spaced the current ratings are reduced to 90 per cent. of those set out in col. 3 or col. 4 above for direct-current or alternating-current (either single-phase or three-phase) loading respectively.

For two or three smaller cables (7/·036 to 19/·064 ins. inclusive) so spaced the current ratings are those given in col. 3 or col. 4 of Table X, and for four or more such smaller cables the current ratings are 90 per cent. of those given in col. 3 or col. 4 of Table X, for direct-current or alternating-current (either single-phase or three-phase) loading respectively.

Table XII refers to situations where the ambient air temperature does not exceed 90° F. (32·2° C.). Where the ambient air temperature is abnormally high, see note to Table X.

The lower limit set to the size of conductor by the permissible voltage drop is dealt with in Regulation 304, p. 1433.

NOTE TO TABLES VIII, IX AND XII. CABLES RUN UNDER DEFINED CONDITIONS.

The current ratings and corresponding lengths (approximate) in circuit for 1 volt drop set out in Tables VIII, IX and XII, apply to cables run under the conditions defined below :—

(1) The circuit comprises two single-core cables carrying direct-current or single-phase alternating current, or three single-core cables carrying three-phase alternating current.

(2) Where the cables are lead-covered, the lead sheaths are electrically bonded together, at each end of the cable run, with bonds of negligible resistance.

(3) The cables are remote from iron, steel or ferro-concrete.

(4) The cables are supported horizontally one above the other on cleats on a vertical wall, and are separated from one another and from the wall by the following distances :—

Nominal Cross-sectional Area of Conductor.	Approximate Vertical Distance between Cable Centres.	Approximate Horizontal Distance of Cable Centres from Wall.
1.	2.	3.
Sq. Ins.		
0·1 to 0·3 inclusive .	Twice the diameter of the finished cable	1½ ins.
0·4 „ 1·0 „ .	3½ ins.	2½ „

EARTH CONTINUITY CONDUCTORS.

Size of earth continuity conductor in metal-sheathed and tough-rubber-protected cables.

Size of Current-carrying Conductor.		Sizes of Wires forming Earth Continuity Conductor.			
		Flat Twin and Flat Three-Core Metal Sheathed Cables.		Flat Twin Tough-rubber-protected Cables.	
Nominal Cross-sectional Area.	Number and Diameter (in.) of Wires.	Approximate Cross-sectional Area of Earth Continuity Conductor.	Number and Diameter (in.) of Wires.	Approximate Cross-sectional Area of Earth Continuity Conductor.	Number and Diameter (in.) of Wires.
1.	2.	3.	4.	5.	6.
Sq. in.		Sq. in.		Sq. In.	
0·0015	1/·044	0·001	1/·036	0·0015	1/·044
0·002	3/·029	0·001	1/·036	0·0015	1/·044
0·003	3/·036	0·0015	1/·044	0·0015	1/·044
0·0045	7/·029	0·0015	1/·044	0·003	3/·036
0·007	7/·036	0·0015	1/·044	0·0045	7/·029
0·01	7/·044	0·002	1/·052	0·007	7/·036
0·0145	7/·052	0·003	1/·064	0·01	7/·044
0·0225	7/·064	0·004	1/·072	0·0145	7/·052

B.S. 480—1942. PAPER-INSULATED CABLES FOR ELECTRICITY SUPPLY.

This specifies cables for operation at voltages between cores up to 22 kV., including thickness of insulation and sheath. If armoured, the armour is to be bedded on to a layer of two compounded paper tapes, with suitable fibrous materials. The armouring shall be either galvanised steel wires or two layers of tape. The serving over armouring shall consist of two layers of compounded paper with suitable fibrous material.

Tests :—

(1) *Measurement of Thickness of Insulations* shall be sum of the measured thicknesses of the individual papers removed from the cable. The thickness of the sheath shall be measured on a ring cut from the sample.

(2) *Bending Test.*—Sample of length 60 times overall diameter bent round a drum 12 times overall diameter, then bent opposite way round, then this operation carried out twice more. The cable shall then withstand the voltage test.

(3) *Voltage Test.*—In the works, with an A.C. supply of value stated in the tables. When laid and jointed, either on A.C. or D.C. test. The voltage shall be applied gradually and maintained for 15 minutes, between conductors and between each conductor and sheath.

Identification of Cores.—Cores shall be numbered on the outer layer of paper, O standing for a neutral and other numbers, 1, 2, 3, etc., indicating live conductors.

TABLE I.—660-VOLT PAPER-INSULATED CABLES.
(Centre Point Earthed only.) Primary Sizes only.

DIMENSIONS.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Nominal Area of Conductors.	Thickness of Insulation.		Single-core Cables.			Twin (Belted) Cables.			Three-core (Belted) Cables.*			Four-core (Belted) Cables.*		
	D.	E.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.
Sq. In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In	In.	In.
0·0225	0·07	0·055	0·06	0·42	0·47	0·06	0·54	0·63	†0·06 0·06	†0·73 0·61	†0·79 0·69	†0·06 0·06	†0·79 0·70	†0·86 0·78
0·04	0·07	0·055	0·06	0·49	0·53	0·06	0·62	0·72	0·06	0·71	0·81	0·06	0·83	0·92
0·06	0·07	0·055	0·06	0·55	0·59	0·06	0·70	0·81	0·06	0·81	0·91	0·06	0·95	1·05
0·10	0·07	0·055	0·06	0·65	0·69	0·06	0·81	0·94	0·06	0·96	1·07	0·07	1·16	1·26
0·20	0·07	0·055	0·07	0·83	0·88	0·07	1·04	1·19	0·07	1·24	1·37	0·08	1·51	1·62
0·30	0·08	0·06	0·07	0·98	1·03	0·07	1·23	1·42	0·08	1·50	1·65	0·10	1·84	2·00

* All cores of equal area. † Circular conductors.

D is thickness of insulation between conductors in a twin or multicore cable.
E „ „ „ „ „ any conductor and sheath.

1. The voltage test shall be as follows:
(a) At works: Between conductors: 3,500 volts A.C. Between any conductor and the sheath (including special-purposes conductor, if any): 2,000 volts A.C. } for 15 minutes.
(b) When laid and jointed: Between conductors. 2,000 volts A.C. or 3,000 volts D.C. Between any conductor and the sheath. 1,150 volts A.C. or 1,750 volts D.C. for 15 minutes.

2. The cables specified in Table I are suitable for the requirements of 650/750-volt Railway Electrification Systems, and are suitable for withstanding the variation of voltage met with in this class of service; in particular, the rise in voltage occurring at no-load, which may be 25 per cent. in excess of the nominal voltage of 750.

In all the Tables Below

D is the thickness of the insulation between conductors in a 3-core cable.

E is the thickness of the insulation between any one conductor and the sheath in a 3-core cable or between the conductor and sheath in a single-core cable.

Table II.—3,300-volt Paper-Insulated Cables.

Centre point earthed only.)

(Up to 0·3 Sq. In. Nominal Area.)

Dimensions.

1	2	3	4	5	6	7	8	9	10
Nominal area of Con-ductors.	Thickness of Insulation		Single-core Cables.			Three-core (Belted) Cables.			Nominal area of of Con-ductors.
	D.	E.	Thick-ness of Sheath.	Diam. over Sheath. Min.	Diam. over Sheath. Max.	Thick-ness of Sheath.	Diam. over Sheath. Min.	Diam. over Sheath. Max.	
Sq. In.	In.	In.	In.	In.	In.	In.	In.	In.	Sq. In.
0·0225	0·11	0·09	0·06	0·49	0·55	0·06 0·06	*0·84 0·72	*0·92 0·83	0·022 5
0·04	0·11	0·09	0·06	0·56	0·62	0·06	0·83	0·94	0·04
0·06	0·11	0·09	0·06	0·62	0·68	0·06	0·93	1·05	0·06
0·01	0·11	0·09	0·06	0·72	0·78	0·07	1·09	1·23	0·1
0·15	0·11	0·09	0·07	0·82	0·88	0·07	1·23	1·38	0·15
0·20	0·11	0·09	0·07	0·90	0·96	0·08	1·37	1·53	0·2
0·25	0·11	0·09	0·07	0·97	1·03	0·08	1·48	1·64	0·25
0·30	0·11	0·09	0·08	1·06	1·12	0·09	1·61	1·78	0·03

* Circular conductors.

Voltage Tests.

The voltage test shall be as follows:

(a) At works: Between conductors: 10,000 volts A.C. } for 15 minutes.
Between any conductor and the sheath: 5,800 volts A.C. } for 15 minutes.

(b) When laid and jointed:

Between conductors.	Between any conductor and the sheath.	
6,000 volts A.C., or 9,000 volts D.C.	3,500 volts A.C., or 5,000 volts D.C.	for 15 minutes.

TABLE III.—6,600-VOLT PAPER-INSULATED CABLES.

(Centre point earthed only.

DIMENSIONS.

1	2	3	4	5	6	7	8	9
	Thickness of Insulation.		Single-core Cables.			Three-core (Belted) Cables.		
Nominal Area of Conductors.	D.	E.	Thickness of Sheath.	Diam. over Sheath. Min.	Diam. over Sheath. Max.	Thickness of Sheath.	Diam. over Sheath. Min.	Diam. over Sheath. Max.
Sq. In.	In.	In.	In.	In.	In.	In.	In.	In.
0·0225	0·15	0·12	0·06	0·55	0·61	0·06 0·06	*0·95 0·83	*1·03 0·94
0·04	0·15	0·12	0·06	0·62	0·68	0·06	0·93	0·05
0·06	0·15	0·12	0·06	0·68	0·74	0·07	1·05	1·17
0·10	0·15	0·12	0·07	0·80	0·86	0·07	1·19	1·33
0·15	0·15	0·12	0·07	0·88	0·94	0·08	1·35	1·50
0·20	0·15	0·12	0·07	0·96	1·02	0·08	1·47	1·63
0·25	0·15	0·12	0·08	1·05	1·11	0·09	1·61	1·77
0·30	0·15	0·12	0·08	1·12	1·18	0·09	1·71	1·89

* Circular conductors.

The voltage test shall be as follows:

(*a*) At works: Between conductors: 16,000 volts A.C. } for 15 minutes.
Between any conductor and the sheath: 9,200 volts A.C. }

(*b*) When laid and jointed:

Between conductors.	Between any conductor and the sheath.	
12,000 volts A.C., or 18 000 volts D.C.	7,000 volts A.C., or 10,500 volts D.C.	for 15 minutes.

TABLE IV.—11,000-VOLT PAPER-INSULATED CABLES.
Centre Point Earthed only.)

DIMENSIONS.

1	2	3	4	5	6	7	8	9	10	11	12	13
Nominal Area of Conductors.	Thickness of Insulation.		Single-core Cables.			3-core (Belted) Cables.			3-core (Screened) Cables.			
	D.	E.	Thickness of Sheath.	Diameter over Sheath.		Thickness of Sheath.	Diameter over Sheath.		Thickness of Insulation Es.	Thickness of Sheath.	Diameter over Sheath.	
				Min.	Max.		Min.	Max.			Min.	Max.
Sq. In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
0·0225	0·21	0·15	0·06	0·61	0·67	0·07	*1·10	*1·18	0·13	0·07	*1·17	*1·27
0·04	0·21	0·15	0·06	0·68	0·74	0·07	1·08	1·23	0·13	0·07	1·15	1·33
0·06	0·21	0·15	0·06	0·74	0·80	0·07	1·17	1·34	0·13	0·07	1·24	1·44
0·10	0·21	0·15	0·07	0·86	0·92	0·08	1·34	1·53	0·13	0·08	1·41	1·62
0·15	0·21	0·15	0·07	0·94	1·00	0·08	1·48	1·68	0·13	0·09	1·57	1·80
0·20	0·21	0·15	0·07	1·02	1·08	0·09	1·62	1·84	0·13	0·09	1·69	1·93
0·25	0·21	0·15	0·08	1·11	1·17	0·09	1·73	1·96	0·12	0·09	1·80	2·06
0·30	0·21	0·15	0·08	1·18	1·24	0·10	1·86	2·13	0·13	0·10	1·93	2·22

* Circular conductors.

The voltage test shall be as follows :

(*a*) At works : Between conductors : 24,000 volts A.C.
Between any conductor and the sheath : 14,000 volts A.C. } for 15 minutes.

(*b*) When laid and jointed :

Between conductors. 20,000 volts A.C., or 30,000 volts D.C.

Between any conductor and the Sheath. 11,500 volts A.C. or 17,500 volts D.C. } for 15 minutes.

TABLE V.—22.000-VOLT PAPER-INSULATED CABLES.
(Centre point Earthed Only.)
DIMENSIONS.

1	2	3	4	5	6	7	8	9	10
	Single-core Cables.				3-core (Belted) Cables.				
Nominal area of Conductors.	Thickness of E Insulation.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.	Thickness of Insulation D.	Thickness of Insulation E.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.
Sq. In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
0·04	0·24	0·10	0·94	1·02	0·34	0·27	0·10	1·52	1·74
0·06	0·24	0·10	1·00	1·08	0·34	0·27	0·10	1·62	1·85
0·10	0·24	0·10	1·10	1·18	0·34	0·27	0·10	1·77	2·01
0·15	0·24	0·10	1·18	1·26	0·34	0·27	0·10	1·90	2·17
0·20	0·24	0·10	1·26	1·34	0·34	0·27	0·10	2·02	2·31
0·25	0·24	0·10	1·33	1·41	0·34	0·27	0·11	2·15	2·45
0·30	0·24	0·10	1·40	1·48	0·34	0·27	0·11	2·26	2·57

The voltage test shall be as follows:

(*a*) At works: Between conductors: 44,000 volts A.C. } for 15 minutes.
Between any conductor and the sheath: 25,500 volts A.C.

(*b*) When laid and jointed:

Between Conductors.	Between any Conductor and the Sheath.	
40,0000 volts A.C. or 60,000 volts D.C.	23,000 volts A.C. or 35,000 volts D.C.	for 15 minutes.

TABLE VI.—22,000-VOLT PAPER-INSULATED CABLES (SCREENED).
(Centre Point Earthed only.)
DIMENSIONS.

1	2	3	4	5	6	7	8	9	10
	3-core Screened.				*3-core S.L. (All Types.)				
Nominal area of Conductors.	Thickness of insulation E.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.	Thickness of Sheath (on Individual cores).	Diameter over Sheath. Min.	Diameter over Sheath. Max.	Diameter over Laid up cores. Min.	Diameter over Laid up cores. Max.
Sq. In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
0·04	0·23	0·10	1·63	1·85	0·07	0·86	0·93	2·00	2·18
0·06	0·23	0·10	1·72	1·96	0·07	0·92	0·99	2·13	2·31
0·10	0·23	0·10	1·87	2·12	0·07	1·02	1·09	2·34	2·52
0·15	0·23	0·10	2·01	2·28	0·08	1·12	1·19	2·57	2·76
0·20	0·23	0·11	2·15	2·44	0·08	1·20	1·27	2·74	2·92
0·25	0·23	0·11	2·26	2·56	0·08	1·27	1·34	2·89	3·08
0·30	0·23	0·12	2·39	2·70	0·09	1·36	1·43	3·08	3·27

* S.L.: Each core separately lead sheathed.

The voltage test shall be as follows:

(*a*) At works: Between any conductor and its sheath: 25,500 volts A.C. for 15 minutes.

(*b*) When laid and jointed: Between any conductor and the sheath: 23,000 volts A.C., or 35,000 volts D.C. for 15 minutes.

TABLE VII.—CABLES SERVED (UNARMOURED).

Minimum Tabulated Diameter of Cable over Sheath.	Serving.		
	Nominal Thickness.	Addition to Diameter	
		Min.	Max.
	In.	In.	In.
All cables . . .	0·100	0·19	0·21

TABLE VIII.—CABLES SINGLE OR DOUBLE WIRE ARMOURED.

Minimum Tabulated Diameter of Cable over Sheath*		National Thickness.				Addition to Diameter.					
						S.W.A.		D.W.A.		S.W.A. Left Bare.	
Above	Up to and including.	Bedding.	Diam. of Armouring Wire.	Separator for D.W.A. Cables.	Serving.	Min.	Max.	Min.	Max.	Min.	Max.
In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
—	0·50	0·100	0·064	0·025	0·100	0·48	0·53	0·65	0·72	0·29	0·32
0·50	1·05	0·100	0·080	0·025	0·100	0·51	0·57	0·71	0·79	0·32	0·36
1·05	2·00	0·100	0·104	0·025	0·100	0·56	0·62	0·81	0·89	0·37	0·41
2·00	—	0·100	0·128	0·025	0·100	0·60	0·67	0·90	0·99	0·41	0·46

* Or minimum tabulated diameter over laid up cores in the case of S.L. type cables.
S.W.A. Single wire armoured. D.W.A. Double wire armoured.

TABLE IX.—CABLES STEEL TAPE ARMOURED AND SERVED.

Minimum Tabulated Diameter of Cable over Sheath.*		Nominal Thickness.			Addition to to Diameter.	
Above.	Up to and Including.	Bedding	Each Steel Tape.	Serving.	Min.	Max.
In.	In.	In.	In.	In.	In.	In.
0·50	0·75	0·100	0·030	0·100	0·47	0·53
0·75	3·00	0·100	0·040	0·100	0·50	0·58
3·00	—	0·100	0·060	0·100	0·58	0·66

* Or minimum tabulated diameter over laid up cores in the case of S.L. type cables.

B.S. 608–1943. VARNISHED CAMBRIC INSULATED CABLES FOR ELECTRICITY SUPPLY.

The specification and tests are generally similar to those for paper-insulated cables, the test voltages being the same and being omitted in the following tables.

TABLE I.—660-VOLT VARNISHED CAMBRIC INSULATED CABLES *

(Centre Point Earthed only.) Primary Sizes only.

DIMENSIONS.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
	Thickness of Insulation.		Single-core Cables.			Twin Cables			Three-core Cables.			Fore-core Cables.		
Nominal area of Conductors.	D.	E.	Thick-ness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.	Thick-ness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.	Thick-ness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.	Thick-ness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.
Sq. In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
0·0225	0·07	0·055	0·06	0·42	0·47	0·06	0·54	0·63	0·06	0·61	0·69	0·06	0·70	0·78
0·04	0·07	0·055	0·06	0·49	0·53	0·06	0·62	0·72	0·06	0·71	0·81	0·06	0·83	0·92
0·06	0·07	0·055	0·06	0·55	0·59	0·06	0·70	0·81	0·06	0·81	0·91	0·06	0·95	1·05
0·10	0·07	0·055	0·06	0·65	0·69	0·06	0·81	0·94	0·06	0·96	1·07	0·07	1·16	1·26
0·20	0·07	0·055	0·07	0·83	0·88	0·07	1·04	1·19	0·07	1·24	1·37	0·08	1·51	1·62
0·30	0·08	0·06	0·07	0·98	1·03	0·07	1·23	1·42	0·08	1·50	1·65	0·10	1·84	2·00

* The cables specified in Table I are suitable for the requirements of 660/750-volt Railway Electrification Systems, and are suitable for withstanding the variation of voltage met with in this class of service; in particular, the rise in voltage occurring at no-load, which may be 25 per cent. in excess of the nominal voltage of 750.

Multicore cables shall have all conductors of equal area.

TABLE II.—3,300-VOLT VARNISHED CAMBRIC INSULATED CABLES.

(Centre Point Earthed only.)

DIMENSIONS.

1	2	3	4	5	6	7	8	9
Nominal Area of Conductors	Thickness of Insulation.		Single-core Cables.			Three-core Cables.		
	D.	E.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.
Sq. In.	In.	In.	In.	In.	In.	In.	In.	In.
0·0225	0·13	0·10	0·06	0·51	0·57	0·06	0·76	0·87
0·04	0·13	0·10	0·06	0·58	0·64	0·06	0·87	0·99
0·06	0·13	0·10	0·06	0·64	0·70	0·06	0·96	1·09
0·10	0·13	0·10	0·06	0·74	0·80	0·07	1·13	1·27
0·20	0·13	0·10	0·07	0·92	0·98	0·08	1·41	1·57
0·30	0·13	0·10	0·08	1·08	1·14	0·09	1·65	1·83

TABLE III.—6,600-VOLT VARNISHED CAMBRIC INSULATED CABLES.

(Centre Point Earthed only.)

DIMENSIONS.

1	2	3	4	5	6	7	8	9
Nominal Area of Conductors	Thickness of Insulation.		Single-core Cables.			Three-core Cables.		
	D.	E.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.
Sq. In.	In.	In.	In.	In.	In.	In.	In.	In.
0·0225	0·18	0·14	0·06	0·59	0·65	0·06	0·90	1·01
0·04	0·18	0·14	0·06	0·66	0·72	0·07	1·03	1·14
0·06	0·18	0·14	0·06	0·72	0·78	0·07	1·12	1·24
0·10	0·18	0·14	0·07	0·84	0·90	0·08	1·29	1·42
0·20	0·18	0·14	0·07	1·00	1·06	0·09	1·57	1·72
0·30	0·18	0·14	0·08	1·16	1·22	0·09	1·79	1·96

TABLE IV.—11,000-VOLT VARNISHED CAMBRIC INSULATED CABLES.

(Centre Point Earthed only.)

DIMENSIONS.

1.	2.	3.	4.	5.	6.	7.	8.	9.
Nominal Area of Conductors.	Thickness of Insulation.		Single-core Cables.			Three-core Cables.		
	D.	E.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.	Thickness of Sheath.	Diameter over Sheath. Min.	Diameter over Sheath. Max.
Sq. In.	In.	In.	In.	In.	In.	In.	In.	In.
0·0225	0·27	0·19	0·060	0·69	0·75	0·07	*1·25	*1·34
0·04	0·27	0·19	0·070	0·78	0·84	0·07	1·23	1·40
0·06	0·27	0·19	0·070	0·84	0·90	0·08	1·34	1·52
0·10	0·27	0·19	0·070	0·94	1·00	0·08	1·49	1·69
0·20	0·27	0·19	0·080	1·12	1·18	0·09	1·77	2·00
0·30	0·27	0·19	0·080	1·26	1·32	0·10	2·01	2·29

* Circular conductors.

Armouring and serving: the particulars are identical with those for paper-insulated cables.

TABLE V.—NOMINAL THICKNESS OF TAPING AND BRAIDING.

Taped and Braided (Non-metal-sheathed Cables).

Minimum calculated Diameter of Cable over Cambric Insulation before Taping and Braiding.		Thickness in the Finished Cable of		Addition to Diameter.	
Above.	Up to and Including.	Taping.	Braiding.	Min.	Max.
In.	In.	In.	In.	In.	In.
—	1·00	0·008	0·03	0·070	0·090
1·00	2·00	0·008	0·05	0·110	0·130
2·00	—	0·008	0·06	0·130	0·150

TABLE VI.—SINGLE-CORE TAPED AND BRAIDED CABLES.

Centre Point Earthed only.)

DIMENSIONS.

1	2	3	4	5	6	7	8	9	10	11	12	13
	660 Volt.			3,300 Volt.			6,600 Volt.			11,000 Volt.		
Nominal Area of Conductors.	Thickness of Insulation E.	Diameter over Compounded Braiding. Min.	Diameter over Compounded Braiding. Max.	Thickness of Insulation E.	Diameter over Compounded Braiding. Min.	Diameter over Compounded Braiding. Max.	Thickness of Insulation E.	Diameter over Compounded Braiding. Min.	Diameter over Compounded Braiding. Max.	Thickness of Insulation E.	Diameter over Compounded Braiding. Min.	Diameter over Compounded Braiding. Max.
Sq. In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
0·007	0·055	0·290	0·330	—	—	—	—	—	—	—	—	—
0·0145	0·055	0·340	0·380	—	—	—	—	—	—	—	—	—
0·0225	0·055	0·370	0·420	0·100	0·460	0·520	0·140	0·540	0·600	0·190	0·640	0·700
0·04	0·055	0·440	0·485	0·100	0·530	0·590	0·140	0·610	0·670	0·190	0·710	0·770
0·06	0·055	0·500	0·540	0·100	0·590	0·650	0·140	0·670	0·730	0·190	0·770	0·830
0·10	0·055	0·595	0·640	0·100	0·685	0·745	0·140	0·765	0·825	0·190	0·865	0·925
0·20	0·055	0·760	0·805	0·100	0·850	0·910	0·140	0·930	0·990	0·190	1·030	1·130
0·30	0·060	0·910	0·960	0·100	0·990	1·050	0·140	1·110	1·170	0·190	1·210	1·270

The thickness of dielectric for multicore taped and braided (non-metal-sheathed) cables shall be the same as for the equivalent metal sheathed cable in Tables I to IV.

ALUMINIUM CONDUCTORS.*

Conductivity.

The conductivity of aluminium depends upon the purity, and, to a less extent, upon the amount of work put upon the metal in its evolution from billet to the finished wire or bar. The British standard of resistance for hard copper is 2 per cent. higher than that of soft, and this difference also applies to hard and soft aluminium. The conductivity of all materials is normally compared with that of the International Standard for annealed copper, and on this basis hard-drawn aluminium will have a minimum conductivity of 60 per cent. Hard-drawn copper will have a conductivity of about 98 per cent. and hence compared with hard-drawn copper, hard-drawn aluminium will have a conductivity of 61·3 per cent.

Weight.

The specific gravity of aluminium wire is about 2·703. As the specific gravity of copper wire is about 8·89, the relative weight of a given volume of each metal is in the ratio of 8·89 to 2·71 = 3·3. In other words, a copper wire of any given size and length will weigh 3·3 times as much as an aluminium wire of the same size and length. Aluminium, therefore, having a conductivity of 61 per cent. of that of copper, will have a sectional area 1·642 times as great as a copper conductor of the same resistance and length, and the ratio of the weights will be 3·3 to 1·642 = 2·01; that is to say, a copper conductor of any given resistance and length will weigh approximately twice as much as an aluminium conductor of the same resistance and length.

The ratios are therefore as follows:—

	Copper.	Aluminium.
Conductance for equal section	1	0·61
Section for equal conductance	1	1·642
Diameter for equal conductance	1	1·28
Weight for equal bulk	3·3	1
Weight for equal conductance	2·0	1

Tensile Strength.

Although the tensile strength of aluminium in the annealed state is low, the strength of aluminium in the form of wire is greatly increased owing to the work put upon it during rolling and drawing. The strength can be increased more than 100 per cent. without making the elongation too small for engineering purposes.

Durability.

Aluminium has withstood the test of varying and severe climatic conditions. Lines have been in operation for over twenty years in exposed positions on the sea coast which show little signs of deterioration. The metal is less affected by sulphurous and sulphuric fumes than copper, and shows less deterioration in the presence of any of the commoner acids. On the other hand, aluminium is attacked by strong alkalies, and should not be used in such circumstances without efficient painting.

Increase of Resistance with Temperature.

The coefficient of increase of resistance with temperature varies somewhat with the extent and nature of the impurities present and the temperature range over which it is taken. At 20° C. the value for aluminium may be taken at 0·00407 per ° C., at 15·6 ° C. in accordance with *B.S. Specification*, No. 215–1934.

Coefficient of Linear Expansion.

The coefficient of linear expansion being somewhat greater with aluminium than with copper, due allowance should be made for this when installing long lines of solid feeders or bars.

Action of Acids.

Aluminium is not acted upon by any of the common acids in a cold state, with the exception of hydrochloric acid, and withstands the effects of sea-water better than either copper or iron; but galvanic action is set up between aluminium and any of the common metals in the presence of moisture, and for this reason, if iron and copper joints must be used, they should be thoroughly protected from moisture by taping or painting with bituminous paint. It is found that galvanised iron parts can be used with aluminium owing to their closer proximity in the electro-chemical series, and the hardware employed with aluminium overhead lines is usually of galvanised malleable iron.

* See also B.S. No. 215—1934, 'Hard-Drawn Aluminium and Steel-Cored Aluminium Conductors for Overhead Power Transmission Purposes.'

STANDARD STRANDED ALUMINIUM CONDUCTORS (B.S. 215—1934)

Standard Nominal Copper Area.	Resistance of Standard Hard-drawn Copper Conductor at 60° F.		Stranding and Wire Diameter of Aluminium Conductor (inches).	Calculated Equivalent Area of Aluminium.	Approximate Weight		Resistance at 60° F. When corrected to Standard Weight.		Resistance at 20° C. When corrected to Standard Weight.		Approximate Ultimate Strength of Conductor.	Standard Nominal Copper Area.
	Ohms per 1,000 yds.	Ohms per mile.			per 1,000 yds.	per mile.	Ohms per 1,000 yds.	Ohms per mile.	Ohms per 1,000 yds.	Ohms per mile.		
Sq. in.				Sq. in.	lb.	lb.					lb.	Sq. in.
0·02	1·236	2·175	3/·118	0·03240	117	206	1·222	2·151	1·245	2·191	860	0·02
0·025	0·9887	1·740	3/·132	0·04052	146	257	0·9768	1·719	0·9947	1·751	995	0·025
0·03	0·8237	1·450	3/·144	0·04825	174	306	0·8208	1·445	0·8360	1·471	1185	0·03
0·04	0·6177	1·087	7/·110	0·06579	236	416	0·6017	1·0659	0·6127	1·078	1725	0·04
0·05	0·4943	0·8700	7/·122	0·08093	291	512	0·4891	0·8607	0·4981	0·8767	2040	0·05
0·06	0·4118	0·7248	7/·134	0·09762	351	618	0·4054	0·7135	0·4129	0·7268	2370	0·06
0·07	0·3530	0·6213	7/·144	0·1128	405	713	0·3511	0·6180	0·3576	0·6294	2735	0·07
0·075	0·3294	0·5798	7/·149	0·1207	434	764	0·3279	0·5770	0·3339	0·5877	2930	0·075
0·08	0·3089	0·5437	7/·154	0·1289	463	816	0·3069	0·5402	0·3126	0·5501	3065	0·08
0·09	0·2745	0·4831	7/·164	0·1462	526	925	0·2707	0·4764	0·2757	0·4853	3480	0·09
0·10	0·2469	0·4345	7/·173	0·1628	585	1029	0·2432	0·4281	0·2477	0·4360	3790	0·10
0·125	0·1976	0·3477	7/·193	0·2026	728	1280	0·1954	0·3439	0·1990	0·3503	4620	0·125
0·15	0·1656	0·2914	7/·211	0·2421	870	1531	0·1635	0·2878	0·1665	0·2931	5405	0·15
0·175	0·1408	0·2478	19/·139	0·2836	1030	1813	0·1395	0·2456	0·1421	0·2501	6775	0·175
0·20	0·1224	0·2155	19/·149	0·3261	1183	2083	0·1214	0·2137	0·1236	0·2176	7785	0·20
0·225	0·1096	0·1928	19/·157	0·3620	1314	2313	0·1094	0·1925	0·1114	0·1961	8470	0·225
0·25	0·09861	0·1736	19/·166	0·4046	1469	2585	0·09781	0·1722	0·09963	0·1754	9470	0·25
0·30	0·08126	0·1430	19/·183	0·4918	1785	3142	0·08049	0·1417	0·08198	0·1443	11160	0·30
0·35	0·06978	0·1228	19/·197	0·5699	2070	3643	0·06947	0·1223	0·07075	0·1245	12790	0·35
0·40	0·06111	0·1076	19/·211	0·6539	2373	4176	0·06054	0·1065	0·06166	0·1085	14360	0·40
0·45	0·05428	0·09553	37/·161	0·7370	2707	4764	0·05373	0·09456	0·05469	0·09626	16610	0·45
0·50	0·04919	0·08657	37/·168	0·8026	2947	5187	0·04933	0·08683	0·05025	0·08843	18090	0·50
0·60	0·04175	0·07348	37/·183	0·9521	3497	6154	0·04157	0·07317	0·04235	0·07435	20800	0·60
0·75	0·03297	0·05803	37/·206	1·2066	4432	7801	0·03282	0·05776	0·03342	0·05883	25800	0·75

STANDARD STEEL-CORED ALUMINIUM CONDUCTORS (B.S. No. 215—1934).

Standard Nominal Copper Area.	Resistance of Standard Hard-drawn Copper Conductor at 60° F.		Stranding and Wire Diameter (inches).		Calculated Equivalent Area of Aluminium	Approximate Total Weight		Resistance at 60° F. When corrected to Standard Weight.		Resistance at 20° C. When corrected to Standard Weight.		Approximate Ultimate Strength of Conductor.	Standard Nominal Copper Area.
	Ohms per 1,000 yds.	Ohms per mile.	Aluminium.	Steel.		per 1,000 yds.	per mile.	Ohms per 1,000 yds.	Ohms per mile.	Ohms per 1,000 yds.	Ohms per mile.		
Sq. in.					Sq. in.	lb.	lb.					lb.	Sq. in.
0·025	0·9887	1·740	6/·0930	1/·0930	0·04024	214	377	0·9839	1·732	1·002	1·763	2160	0·025
0·03	0·8237	1·450	6/·102	1/·102	0·04840	258	454	0·8179	1·404	0·8329	1·466	2545	0·03
0·04	0·6177	1·087	6/·118	1/·118	0·06480	345	607	0·6111	1·076	0·6224	1·095	3410	0·04
0·05	0·4943	0·8700	6/·132	1/·132	0·08103	432	760	0·4884	0·8596	0·4973	0·8753	4105	0·05
0·06	0·4118	0·7248	6/·144	1/·144	0·09065	514	905	0·4104	0·7232	0·4180	0·7735	4885	0·06
0·07	0·3530	0·6213	6/·157	1/·157	0·1147	611	1075	0·3452	0·6076	0·3516	0·6189	5735	0·07
0·075	0·3294	0·5798	6/·161	1/·161	0·1206	643	1131	0·3283	0·5779	0·3343	0·5883	6035	0·075
0·08	0·3089	0·5437	6/·166	1/·166	0·1282	683	1202	0·3088	0·5434	0·3145	0·5535	6415	0·08
0·09	0·2745	0·4831	6/·177	1/·177	0·1458	777	1367	0·2716	0·4781	0·2767	0·4870	7220	0·09
0·10	0·2469	0·4345	6/·186	7/·0620	0·1609	797	1403	0·2460	0·4329	0·2505	0·4409	7385	0·10
			7/·173	7/·0760	0·1625	912	1605	0·2437	0·4289	0·2482	0·4368	9270	
0·125	0·1976	0·3477	*26/·100	7/·0750	0·1992	1053	1854	0·1987	0·3497	0·2023	0·3561	10660	0·125
			30/·0930	7/·0930	0·1989	1222	2151	0·1991	0·3503	0·2027	0·3568	12870	
0·15	0·1656	0·2914	30/·102	7/·102	0·2392	1470	2588	0·1655	0·2912	0·1685	0·2966	15240	0·15
0·175	0·1408	0·2478	30/·110	7/·110	0·2782	1710	3009	0·1423	0·2504	0·1449	0·2550	17720	0·175
0·20	0·1224	0·2155	30/·118	7/·118	0·3203	1967	3463	0·1236	0·2167	0·1259	0·2216	20390	0·20
0·225	0·1096	0·1928	30/·125	7/·125	0·3592	2208	3886	0·1102	0·1939	0·1122	0·1975	22520	0·225
0·25	0·09861	0·1736	30/·132	7/·132	0·4005	2462	4333	0·09881	0·1739	0·1006	0·1771	24710	0·25
0·30	0·08126	0·1430	30/·146	7/·146	0·4901	3012	5301	0·08078	0·1422	0·08224	0·1448	30230	0·30
0·35	0·06978	0·1228	30/·157	7/·157	0·5668	3483	6131	0·06985	0·1229	0·07114	0·1252	34590	0·35
0·40	0·06111	0·1076	30/·168	7/·168	0·6491	3988	7019	0·06100	0·1074	0·06213	0·1093	39600	0·40
0·45	0·05428	0·09553	30/·177	7/·177	0·7205	4427	7792	0·05495	0·09672	0·05598	0·09852	43600	0·45
0·50	0·04919	0·08657	54/·139	7/·139	0·7994	4042	7114	0·04951	0·08713	0·05042	0·08874	36300	0·50
0·60	0·04175	0·07348	54/·152	7/·152	0·9565	4834	8507	0·04140	0·07287	0·04216	0·07421	42860	0·60
0·75	0·03297	0·05803	54/·170	7/·170	1·1963	6046	10641	0·03309	0·05824	0·03370	0·05931	53630	0·75

* The 26 aluminium/7 steel is the recommended construction for the 0·125 sq. in. conductor.

PHYSICAL PROPERTIES OF ALUMINIUM AND COPPER.

Atomic weight of aluminium (oxygen = 16) = 26·97 (*Int. Atomic Wt. Comm.*, 1925).

Coefficient of linear expansion per °C. = 23×10^{-6} (*B.S., No.* 215–1934.)

" " " " °F. = $12 \cdot 78 \times 10^{-6}$

Specific gravity, rolled or drawn = 2·703 (*B.S., No* 215-1934).

With regard to the tensile strength of H.D. wires, the minimum wire diameter given should be 0·093 ins. (not 0·0935 ins.). See *B.S.*, No. 215–1934.

The modulus of elasticity of H.D. aluminium conductors = $9 \cdot 9 \times 10^{6}$ lbs. per sq. in.

The modulus of elasticity of H.D. aluminium stranded conductor = $9 \cdot 6 \times 10^{-6}$ lbs. per sq. in.

Coefficient of increase of resistance with temperature at 15·6° C. (60° F.) = ·00407 per ° C.

Coefficient of increase of resistance with temperature at 60° F. (15·6° C.) % 0·00226 per ° F.

Weight per 1,000 yds. per sq. in. of cross section = 3515 lbs.

Standard specific resistance in microhms per in.^{3} at 20° C. of H.D. aluminium = 1·1199.

All the above specific resistance figures are standard values.

Capacitance and Inductance.

Aluminium being a non-magnetic material, the inductance or capacitance of the line can only be affected by the increased diameter. This increased diameter may decrease the inductance and increase the capacitance by some 5 per cent., but as all transmission lines possess more inductance than capacitance, this difference is in favour of aluminium, tending to improve the power factor of the system and to reduce the pressure drop due to inductance.

Heating Effect.

Aluminium, having a larger surface than copper of equal conductance, is more easily able to get rid of its heat, and when transmitting a given current with a given I^2R loss, aluminium will remain the cooler. This is a considerable advantage when installing electric apparatus in a hot atmosphere, or where current-carrying capacity is of more importance than the permissible pressure drop. In any case aluminium will have a higher overload capacity, from a heating point of view, than copper of equal resistance. In consequence of this aluminium is being largely used for busbars and solid rod battery connections. Erection on site is facilitated by the ease with which bends and joints can be made.

Corona Losses.—With a high voltage overhead line a serious loss of energy may occur in the form of a luminous discharge due to the breakdown of the air surrounding the conductors. The point at which corona begins is a function of the conductor diameter, and the use of aluminium or, better still, steel-cored aluminium, both of which involve a larger diameter than that of the equivalent copper, will safely enable a higher working voltage to be employed.

ALUMINIUM CABLE, STEEL-CORED.

After the introduction of aluminium for electrical conductors it became apparent that if a conductor could be obtained, combining the advantages of lightness of aluminium with the strength of high-grade steel now obtainable on the market, the highest efficiency would result. Cables of this character were made up and have now been in service in such a variety of localities and conditions that their efficiency has been thoroughly demonstrated.

The cables consist of a steel core about which the aluminium is stranded. The steel for the smaller size cables consists of a single strand, while for the larger sizes several strands are used. These are composed of patent steel, having an ultimate strength of 160,000 lbs. per square inch to 200,000 lbs. per square inch, according to the diameter, an elastic limit of 70 per cent. of the ultimate strength, and an elongation of 4 to 6 per cent. on 10 inches. The wires are heavily galvanised, and after the aluminium strands are in place over the steel, around which they fit closely by reason of the stress resulting from the load on the cable, the steel is to a large extent protected from corrosive action. This protection is due to the almost complete exclusion of moisture from the steel core by the aluminium envelope, and further on account of the insulating oxide film on the aluminium, which tends to prevent the flow of small local electric currents. Another feature which adds to this desirable condition is the fact that zinc and aluminium are closer together in the electro-chemical series than any other two metals in common use.

Hard-Drawn Copper Conductors for Overhead Lines.

(B.S. No. 125—1947.) (*Abbreviated.*)

STANDARD SOLID CONDUCTORS.

Standard Nominal Area. Sq. In.	Diameter, Inches.			Weight per 1,000 Yds. Lb.			Weight per Mile. Lb.			Resistance per 1,000 yds. at 60° F.		Resistance per mile at 60° F.		Ultimate Tensile.		Minimum Elongation on 10 in. %
	Standard.	Max.	Min.	Standard.	Max.	Min.	Standard.	Max.	Min.	Standard.	Max.	Standard.	Max.	Strength. Lb.	Stress Tons per Sq. In.	
0·003217†	0·064	0·0646	0·06336	37·20	37·94	36·45	65·46	66·77	64·15	7·683	7·760	13·52	13·66	213	29·5	—
0·00850†	0·104	0·1050	0·1030	98·22	100·2	96·25	172·9	176·3	169·4	2·908	2·937	5·118	5·169	550	28·9	—
0·01057†	0·116	0·1172	0·1148	122·2	124·6	119·7	215·1	219·4	210·8	2·337	2·360	4·112	4·154	676	28·6	—
0·01287*	0·128	0·1293	0·1267	143·8	151·8	145·8	261·9	267·1	256·6	1·919	1·938	3·377	3·410	816	28·3	—
0·01348	0·131	0·1323	0·1297	155·8	159·0	152·7	274·3	279·8	268·8	1·832	1·850	3·223	3·256	852	28·2	—
0·01453	0·136	0·1374	0·1346	168·0	171·3	164·6	295·6	301·5	289·7	1·699	1·716	2·990	3·020	912	28·0	—
0·01697	0·147	0·1485	0·1455	196·2	200·2	192·3	345·4	352·3	338·5	1·451	1·468	2·559	5·584	1056	27·8	—
0·02011*	0·160	0·1616	0·1584	232·5	237·1	227·8	409·1	417·3	401·0	1·227	1·239	2·159	2·181	1237	27·5	—
0·02164	0·166	0·1677	0·1643	250·2	255·2	245·2	440·4	449·2	431·6	1·140	1·151	2·006	2·026	1324	27·3	—
0·025*	0·178	0·1798	0·1762	287·7	293·5	282·0	506·4	516·5	496·3	0·9909	1·001	1·744	1·761	1515	27·1	—
0·02926	0·193	0·1949	0·1911	338·3	345·0	331·5	595·3	607·2	583·4	0·8425	0·8510	1·483	1·498	1749	26·7	—
0·03631	0·215	0·2171	0·2128	419·8	428·2	411·4	738·8	753·6	724·0	0·6787	0·6855	1·195	1·206	2142	26·3	1·85
0·05*	0·252	0·2545	0·2495	576·7	588·2	565·1	101·5	103·5	994·6	0·4937	0·4986	0·8689	0·8776	2875	25·7	2·00
0·109*	0·372	0·3757	0·3683	125·7	128·2	123·2	221·2	225·6	216·7	0·2261	0·2283	0·3979	0·4019	5707	23·4	2·47

* Sizes recommended for use as solid conductors.

† The minimum sizes of single conductors permitted by the Electricity Commissioners under the overhead lines regulations are 0·01287 for service lines and 0·0201 for other lines. The conductors marked with an obelus (†) in the above Table are for use only in making up stranded conductors.

STANDARD STRANDED CONDUCTORS.

(*Abbreviated.*)

Standard Nominal Area Sq. In.	Strands and Diameter.	Approximate over-all Diameter. Ins.	Weight per 1,000 Yds. Lb.			Weight per Mile Lb.			Resistance per 1,000 Yds. at 66° F.		Resistance per Mile at 60° F.		Approximate Ultimate Strength of Conductor. Lb.
			Standard.	Max.	Min.	Standard.	Max.	Min.	Standard.	Max.	Standard.	Max.	
0·022	7/·064	0·192	262·6	267·9	257·3	462·0	471·2	452·8	1·106	1·117	1·947	1·966	1372
0·025	3/·104	0·224	297·0	302·9	291·1	522·8	533·3	512·3	0·9771	0·9869	1·720	1·737	1518
0·050	3/·147	0·317	593·3	605·2	581·4	104·4	106·5	102·3	0·4885	0·4934	0·8598	0·8684	2915
0·058	7/·104	0·312	693·2	707·1	679·3	122·0	124·4	119·6	0·4188	0·4230	0·7370	0·7443	3542
0·075	7/·116	0·348	862·5	879·8	845·3	151·8	154·8	148·8	0·3365	0·3399	0·5921	0·5980	4353
0·100	7/·136	0·408	118·6	121·0	116·2	208·6	212·8	204·4	0·2447	0·2471	0·4306	0·4349	5873
0·150	7/·166	0·498	176·6	180·1	173·1	310·8	317·0	304·6	0·1642	0·1658	0·2889	9·2918	8527
0·200	7/·193	0·579	238·8	243·6	234·0	420·2	428·6	411·8	0·1213	0·1225	0·2136	0·2157	11260
0·200	19/·116	0·580	235·1	239·8	230·4	413·9	422·2	405·6	0·1245	0·1257	0·2191	0·2213	11560
0·250	7/·215	0·645	296·3	302·2	290·4	521·4	531·8	511·0	0·09773	0·09871	0·1721	0·1738	13790
0·250	19/·131	0·655	299·8	305·8	293·8	527·8	538·4	517·2	0·09761	0·09859	0·1717	0·1734	14570

BRAIDED COPPER OVERHEAD LINES—B.S. No. 446—1932.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
Standard Nominal Area.	Calculated Area.	Number and Diameter of Wires comprising Conductor.	Diameter of Single Wires, inches.		Overall Diameter of Conductor.	Overall Diameter of Finished Cable, inches.			Weight per 1,000 yd. (bare conductor). lb.			Weight per mile. (bare conductor). lb.			Resistance at 60° F. when corrected to standard weight. Ohms per 1,000 yd.		Ohms per mile.	
Sq. in.	Sq. in.		Max.	Min.	Inches.	Single Braid.	Double Braid.	Triple Braid.	Standard.	Max.	Min.	Standard.	Max.	Min.	Standard.	Max.	Standard.	Max.
0·001	0·001018	1/·036	0·0364	0·0356	0·036	0·076	0·116	0·156	11·8	12·0	11·5	20·7	21·1	20·3	24·29	24·53	42·76	43·19
0·0015	0·001521	1/·044	0·0444	0·0436	0·044	0·084	0·124	0·164	17·6	17·9	17·2	30·9	31·6	30·3	16·26	16·42	28·62	28·91
0·002	0·001943	3/·029	0·0293	0·0287	0·062	0·102	0·142	0·182	23·4	23·8	22·9	41·1	41·9	40·3	12·73	12·86	22·40	22·62
0·003	0·002994	3/·036	0·0364	0·0356	0·078	0·118	0·158	0·198	36·0	36·7	35·3	63·4	64·6	62·1	8·260	8·340	14·54	14·69
0·003	0·003217	1/·064	0·0646	0·0634	0·064	0·104	0·144	0·184	37·2	37·9	36·4	65·5	66·8	64·1	7·683	7·760	13·52	13·66
0·0045	0·004546	7/·029	0·0293	0·0287	0·087	0·127	0·167	0·207	54·4	55·4	53·3	97·7	97·6	93·8	5·440	5·494	9·574	9·670
0·007	0·007005	7/·036	0·0364	0·0356	0·108	0·148	0·188	0·228	83·8	85·5	82·1	117·5	150·4	144·5	3·530	3·565	6·212	6·274
0·01	0·01046	7/·044	0·0444	0·0436	0·132	0·172	0·212	0·252	125·2	127·7	122·7	220·3	224·8	215·9	2·363	2·387	4·158	4·200
0·0145	0·01462	7/·052	0·0525	0·0515	0·156	0·196	0·236	0·276	174·8	178·3	171·3	307·7	318·8	301·5	1·691	1·708	2·976	3·006
0·0225	0·02214	7/·064	0·0646	0·0634	0·192	0·232	0·272	0·312	264·8	270·1	259·5	466·1	475·4	456·8	1·116	1·127	1·965	1·985
0·03	0·02840	19/·044	0·0444	0·0436	0·220	0·280	0·340	0·400	340·4	347·1	333·5	599·0	611·0	587·0	0·8720	0·8807	1·535	1·550
0·037	0·03724	7/·083	0·0838	0·0822	0·249	0·309	0·369	0·429	445·4	454·3	445·2	783·9	799·6	768·2	0·6634	0·6700	1·168	1·180
0·04	0·03960	19/·052	0·0525	0·0515	0·260	0·320	0·380	0·440	475·4	484·9	465·8	836·7	853·4	819·9	0·6241	0·6303	1·098	1·109
0·06	0·05999	19/·064	0·0646	0·0634	0·320	0·380	0·440	0·500	720·1	734·5	705·7	1267	1293	1242	0·4120	0·4161	0·7252	0·7325
0·075	0·07592	19/·072	0·0727	0·0713	0·360	0·420	0·480	0·540	911·4	929·6	893·1	1604	1636	1572	0·3255	0·3288	0·5729	0·5786
0·1	0·1009	19/·083	0·0838	0·0822	0·415	0·475	0·535	0·595	1211	1235	1187	2131	2174	2089	0·2449	0·2473	0·4310	0·4353
0·12	0·1168	37/·064	0·0646	0·0634	0·448	0·508	0·568	0·628	1403	1431	1375	2469	2519	2420	0·2117	0·2138	0·3726	0·3763
0·15	0·1478	37/·072	0·0727	0·0713	0·504	0·564	0·624	0·684	1776	1811	1740	3125	3187	3063	0·1672	0·1689	0·2943	0·2972
0·2	0·1964	37/·083	0·0838	0·0822	0·581	0·641	0·701	0·761	2360	2407	2312	4153	4236	4070	0·1258	0·1271	0·2214	0·2236
0·25	0·2465	37/·093	0·0939	0·0921	0·651	0·711	0·771	0·831	2962	3022	2903	5214	[illegible]	5022	0·1002	0·1012	0·1764	0·1782
0·3	0·3024	37/·103	0·104	0·162	0·721	0·781	0·841	0·901	3634	3707	3561	6396	6524	6268	0·08168	[illegible]	0·1438	0·1452
0·4	0·4064	61/·093	0·0939	0·0921	0·837	0·897	0·957	1·017	4885	4983	4787	8598	8770	8426	0·06090	0·06141	0·1070	0·1031
0·5	0·4985	61/·103	0·104	0·102	0·927	0·987	1·047	1·107	5992	6112	5872	10546	10757	10336	0·04956	0·05006	0·08722	0·08809
0·6	0·6082	91/·093	0·0939	0·0921	1·023	1·083	1·143	1·203	7286	7373	7084	12828	12384	12571	0·04076	0·04117	0·07174	0·07246
0·75	0·7435	91/·103	0·104	0·102	1·133	1·193	1·253	1·313	8940	9119	8761	15735	16050	15420	0·03322	0·03355	0·05847	0·05905
0·85	0·8459	127/·093	0·0339	0·0921	1·209	1·269	1·329	1·389	10173	10376	9969	17904	18262	17546	0·02921	0·02950	0·05140	0·05191
1·0	1·0376	127/·103	0·104	0·102	1·339	1·399	1·459	1·519	12478	12727	12228	21961	22400	21522	0·02381	0·02405	0·04190	0·04232

The following table shows at a glance the superiority in mechanical characteristics of steel-cored conductors over all-aluminium, or copper of equivalent conductance.

Size.	—	Aluminium.	Steel-cored Aluminium.	Copper.
0·025 sq. ins. copper area	Weight	1·0	1·48	2·04
	Diameter	1·0	0·98	0·79
	Ultimate load	1·0	2·19	1·52
0·10 sq. ins. copper area	Weight	1·0	1·37	2·04
	Diameter	1·0	1·07	0·79
	Ultimate load	1·0	1·95	1·55
0·40 sq. ins. copper area	Weight	1·0	1·68	2·04
	Diameter	1·0	1·11	0·79
	Ultimate load	1·0	2·58	1·58

Physical Properties of Steel-Cored Aluminium Conductors.

The strength of a steel-cored aluminium conductor is somewhat less than the sum of the strengths of the component wires, owing to the fact that the aluminium wires will break before the steel wire reaches its ultimate stress. The strength of the composite conductor may be taken as (*a*) 98 per cent. of the sum of the strengths of the aluminium wires, plus 89 per cent. of the sum of the strengths of the steel wires when taken from the stranded conductor and tested, or (*b*) 98 per cent. of the sum of the strengths of the aluminium wires, plus 85 per cent. of the sum of the strengths of the steel wires based on the strengths of the component wires before stranding, *i.e.* in the coil (*B.S. Specification*, No. 215–1934).

The stress is not distributed uniformly over the different wires, and the stress in the steel will normally be about three times the stress in the aluminium.

The coefficient of expansion and modulus of elasticity of the composite conductor are calculated upon the assumption that there will be no relative movement between the aluminium and steel wires. This is found to be the case in practice, and calculations based on 'virtual' values of the constants as so determined are found to conform with sensible accuracy to observed values. The constants depend upon the ratio of steel section to aluminium section in the construction, and the following values apply to standard forms:

No. of Wires.	Coefficient of Linear Expansion per ° F.	Modulus of Elasticity. Lbs. per sq. in.
7	$10 \cdot 55 \times 10^{-6}$	$12 \cdot 49 \times 10^{6}$
37	$10 \cdot 02 \times 10^{-6}$	$13 \cdot 31 \times 10^{6}$
61	$10 \cdot 90 \times 10^{-6}$	$11 \cdot 98 \times 10^{6}$

Conductivity.

The conductivity of a steel-cored aluminium conductor is taken as being that of the aluminium portion.

National Grid Transmission Lines.

The standard line conductor of the National 'Grid' Transmission System in Great Britain consists of a central core of 7 strands of galvanised steel wire, each 0·11 in. diam., surrounded by 30 strands of aluminium wire of the same diameter. The overall diameter is 0·77 in., equivalent in conductivity to a copper conductor of 0·175 sq. in. The carrying capacity of each 3-phase circuit is normally 50,000 kVA., and the current at 132 kV. is 219 amps.; but the conductor is suitable for currents up to 450 amps. and in practice loads up to 90 MW are commonly transmitted over a distance of 100 miles. The critical corona voltage for this diameter conductor under normal atmospheric conditions is 184 kV. The earth conductor has 7 steel and 12 aluminium strands, all 0·11 in. diam. The normal span length is 900 ft. At the Firth of Forth crossing the span is 3,050 ft., and the conductors consist of 19 strands of galvanised steel each 0·125 in. diam., and 18 aluminium strands of the same diameter.

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Aluminium Overhead Lines.

Span Length.

With short spans where a large cross section is required, all-aluminium conductors have a sag differing but little from that of copper, while in the case of a small cross section steel-cored aluminium may be installed with a smaller sag and lower poles. In the case of long spans steel-cored aluminium is also employed, but in this case the span length may be increased. This leads to a reduction of capital outlay and upkeep expenses owing to the reduced number of insulators.

SAG IN STILL AIR AT 120° F. OF VARIOUS CONDUCTORS ERECTED IN ACCORDANCE WITH THE BRITISH H.T. REGULATIONS.

Span Length (Ft.).		100	200	300	400	500	600	700	800	900	1000
Conductors having the conductivity of 0·05 sq. in. copper	Steel-cored Aluminium .	0·25	1·31	3·8	7·62	—	—	—	—	—	—
	Copper . .	0·46	2·4	6·1	11·37	—	—	—	—	—	—
0·1 copper	Steel-cored Aluminium .	—	—	2·72	5·18	8·46	12·55	—	—	—	—
	Copper . .	—	—	3·94	7·16	11·35	16·5	—	—	—	—
0·15 copper	Steel-cored Aluminium .	—	—	—	—	—	7·10	9·91	13·2	17·0	21·3
	Copper . .	—	—	—	—	—	14·0	19·0	24·7	31·3	38·5

Loading conditions.— { 8 lbs. per ft. wind. ⅜ in. radial thickness ice. Factor of safety = 2.

Poles for Transmission Lines.

B.S. No. 513—1933 refers to European Larch Poles.
B.S. No. 607—1916 refers to Reinforced Concrete Poles.

Cables for High Voltage.

'H' and 'SL' Cables.—For voltages above 22 kV. the use of the so-called 'H' (Hochstädter) and 'SL' separate (Lead Sheathed) type cables has become general. In the H-type each core has a thin metal-foil sheathing, while the SL construction provides a separate thin lead sheath for each core. This arrangement ensures a radial electric field and obviates tangential electric stresses which caused serious troubles with belted cables. The sheaths acquire earth potential by contact, so that the fillers carry no electric stress.

SO-Type Cables.—Cables of the 'SO' type have been designed to improve the mechanical and thermal properties of high-voltage cables, the internal form being roughly triangular corresponding with the three conductors. The insulation of the conductors remains unchanged, but the filler area is much reduced, resulting in a shortening of the periphery and a reduction in the lead sheath and armouring.

Oil-Filled Cables.—Three-conductor cables, with paper insulation impregnated with highly viscous oil-resin compound, are satisfactory up to about 33 kV., but have given considerable trouble when used for higher voltages. This is due to the combined action of two processes: (1) fluid dielectrics experience mechanical forces tending to move them away from regions of high electric field density towards the sheathing; (2) the viscosity of impregnating compounds decreases with rise in temperature. Both lead to the formation of gas voids in the insulation of the cable.

These difficulties have been overcome in the oil-filled cable. Single-conductor cables are generally used, although 3-conductor designs are possible. The conductors are wound with graded paper, the impregnating medium being a low-viscosity mineral oil. The system is arranged so that the expansion of the oil due to temperature rise is accommodated by means of external reservoirs which return the oil to the cable under a small hydrostatic pressure when the temperature falls.

Oil-filled cables at 132 kV. are in operation between Kidbrooke and Deptford, Wimbledon and Battersea, in London, for connection to the national grid network.

More recently cables have been developed in which the formation of voids is prevented by the maintenance of the cable under external gas pressure of about 200 lb. per sq. in. either by placing the cable in a steel pipe or by using two lead sheaths, with the high-pressure gas between them. Experimental lengths of different designs of 132 kV. three-core cable have been installed in many parts of Britain.

ENAMELLED COPPER WIRE.

British Standard, No. 156—1943, gives the following dimensions and resistances of enamelled copper wire for instruments and apparatus.

1	2	3	4	5	6
Nominal Diameter of Bare Wire.	Maximum Overall Diameters of Enamelled Wire.		Resistance per 1,000 Yds.		
	Normal Covering.	Thick Covering.	Standard.	Maximum.	Minimum.
	Maximum.	Maximum.			
In.	In.	In.	Ohms.	Ohms.	Ohms.
0·0020	0·0025	0·0027	7642	8788	6872·1
0·0022	0·0027	0·0029	6316	7263·4	5724·4
0·0024	0·0030	0·0033	5307	5944	4841·6
0·0028	0·0034	0·0038	3899	4367	3594·1
0·0032	0·0039	0·0043	2985	3343	2773·1
0·0036	0·0044	0·0048	2359	2595	2193·0
0·0040	0·0048	0·0052	1910·5	2101·6	1785·7
0·0044	0·0052	0·0057	1578·9	1705·2	1482·3
0·0048	0·0056	0·0061	1326·7	1432·8	1250·2
0·0050	0·0058	0·0063	1223·0	1320·8	1154·1
0·0052	0·0061	0·0066	1130·5	1220·9	1068·7
0·0056	0·0065	0·0070	974·7	1052·7	923·87
0·0060	0·0069	0·0075	849·1	900·0	806·70
0·0064	0·0073	0·0079	746·3	791·08	710·50
0·0068	0·0078	0·0084	661·1	700·80	630·52
0·0072	0·0082	0·0088	589·7	625·08	563·33
0·0076	0·0086	0·0092	529·2	556·70	506·26
0·0080	0·0090	0·0096	477·6	502·43	456·97
0·0084	0·0095	0·0101	433·2	455·70	415·45
0·0088	0·0099	0·0105	394·7	415·22	378·93
0·0092	0·0103	0·0110	361·2	380·0	347·11
0·0100	0·0112	0·0119	305·7	320·4	294·58
0·0108	0·0121	0·0128	262·1	274·7	252·31
0·0116	0·0129	0·0137	227·2	238·1	218·71
0·0124	0·0138	0·0147	198·80	208·34	191·38
0·0136	0·0151	0·0160	165·27	172·71	159·10
0·0148	0·0164	0·0173	139·55	145·83	134·34
0·0164	0·0181	0·0191	113·65	118·76	109·41
0·018	0·0198	0·0208	94·35	98·50	90·826
0·020	0·0219	0·0230	76·42	79·78	73·566
0·022	0·0240	0·0251	63·16	65·94	60·801
0·024	0·0261	0·0272	53·07	55·41	51·088
0·028	0·0302	0·0314	38·99	40·67	37·534
0·032	0·0343	0·0356	29·85	31·13	28·735
0·036	0·0381	0·0398	23·59	24·60	22·709
0·040	0·0425	0·0440	19·105	19·907	18·391
0·048	0·0507	0·0521	13·267	13·824	12·772
0·056	0·0589	0·0608	9·747	10·156	9·3830
0·064	0·0671	0·0693	7·463	7·769	7·1843
0·072	0·0753	0·0777	5·897	6·139	5·6768
0·076	0·0794	0·0820	5·292	5·509	5·0943
0·080	0·0835	0·0862	4·776	4·972	4·5976
0·084	0·0876	0·0904	4·332	4·510	4·1702
0·092	0·0958	0·0988	3·612	3·753	3·4771
0·100	0·1040	0·1071	3·057	3·176	2·9428
0·104	0·1080	0·1112	2·826	2·936	2·7205
0·116	0·1204	0·1237	2·272	2·361	2·1871
0·128	0·1328	0·1362	1·8657	1·9385	1·7960

Tests are prescribed as follows:—

Mechanical.—The enamel must not be easily removed from the wire with the thumbnail after winding on a 1-in. mandrel. Below 0·01 in. diameter, a 10-in. sample shall be extended to 11 ins. and baked for one hour at 130° C. without cracking the enamel; above that diameter a sample shall be wound round a mandrel without cracking the enamel under microscope.

Chemical.—The enamel shall not be dissolved when immersed for one hour in absolute alcohol or dilute sulphuric acid (sp. gr. 1·25); and the samples shall pass the mechanical tests after immersion as above.

Electrical.—Test samples twisted together shall withstand an arc test at a pressure varying from 150 volts (R.M.S.) up to 0·0032 in. to 1,200 volts above 0·029 in. diameter. A sample shall pass specified pinhole tests.

In the trade enamelled wire with a 'thick' covering of enamel is sometimes known as 'double-enamelled' wire.

Approximate Sizes of Fuse-elements composed of Tinned Copper Wire or Standard Alloy* Wire for use in Semi-enclosed British Standard Fuses.

Current Rating.	Tinned Copper Wire.		Standard Alloy* Wire.	
	Diameter (in.)	S.W.G.	Diameter (in.)	S.W.G.
1.	2.	3.	4.	5.
amps.				
1·8	—	—	0·0164	27
3·0	0·006	38	0·024	23
5·0	0·0084	35	0·032	21
8·5	0·0124	30	—	—
10·0	0·0136	29	—	—
15·0	0·020	25	—	—
17	0·022	24	—	—
20	0·024	23	—	—
24	0·028	22	—	—
30	0·032	21	—	—
37	0·040	19	—	—
46	0·048	18	—	—
53	0·048	18	—	—
60	0·056	17	—	—
64	0·056	17	—	—
83	0·072	15	—	—
100	0·080	14	—	—

Note.—The current ratings given in this table refer to the normal maximum current of the circuit and do not refer to the overload at which the fuse will operate.

The values of the currents given in above table are approximately those necessary to comply with British Standard No. 88. Where fuses are known to conform to this Specification the size stated by the manufacturer on the case of the fuse should be adhered to in preference to that given above, if the fuse is loaded to its full capacity.

Insulators for Overhead Lines.

British Standard for Porcelain and Toughened Glass Insulators for Overhead Lines. (3·3 kV. and Upwards). No. 137—1941.

(a) *Porcelain Insulators.*

The porcelain must be white, sound, completely vitrified, with insulation independent of the glazing, which must cover all exposed parts. An arbitrary rating number is assigned to all insulators which pass the associated tests mentioned below. Table I shows the rating numbers and test voltages. The porcelain must not engage directly with hard metal such as a screw and must be

* The term 'Standard Alloy' refers to the tin-lead alloy (63 per cent. tin, 37 per cent. lead). I.E.E. Wiring Regulations—Eleventh Ed.

unaffected by temperature stresses. Every insulator must have manufacturer's name, month and year of manufacture printed on it before firing. The following tests are applied :—

(1) 50 per cent. Dry Impulse Flash-over Test. The dry and clean insulator shall be subjected to an impulse voltage rising to its maximum in 1 micro-second and falling to half its maximum in 50 micro-seconds, the '50 per cent.' signifying that 50 per cent. of the impulses cause flash-over. After not less than 20 impulses the insulator shall be undamaged.

(2) One-minute Dry Test and Dry Flash-over Test. The voltage shall be gradually raised to that of column 4 of Table I and maintained for one minute, then gradually increased till flash-over occurs. The insulator shall then be flashed-over four times. The flash-over value shall not be less than that in column 2.

(3) One-minute Rain Test and Rain Flash-over Test. As in (2) above, but the one-minute test shall be that of column 5, and the flash-over voltage that of column 3, with artificial rain of 0·12 in. per minute at 45° to the vertical.

(4) Puncture Test. The insulator shall not puncture at the test voltages of column 6 or 7 at power frequency or at three times the dry 50 per cent. flash-over voltage in an impulse test.

In addition routine mechanical tests (20 per cent. over specified load) and porosity, galvanising and routine electrical tests are specified.

(b) *Toughened Glass Insulators.*

As for porcelain insulators, but with a thermal shock test to prove that the glass will not crack due to sudden heating.

The Choice of Insulators.

Table II gives the insulator rating numbers recommended for use in clean atmospheric conditions for a range of service voltages. The temperature must not exceed 40° C. and the altitude must be less than 3,300 ft. for these values. For polluted atmospheres larger rating numbers are required.

TABLE I.—STANDARD INSULATOR RATING NUMBERS AND TEST-VOLTAGES.

(R.M.S. Values : 50 Cycles per Second.)

1	2	3	4	5	6	
					Puncture Test-Voltage.	
Insulator Rating Number.	Minimum Dry Flash-over Voltage.	Minimum Wet Flash-over Voltage.	One-Minute Dry Test-Voltage.	One-Minute Rain Test-Voltage.	Pin Insulator or Line Post Insulator.	String Insulator Unit.
	kV.	kV.	kV.	kV.	kV.	kV.
16	38	17	36	16	68	1·3 times the actual dry flash-over voltage of the unit.
22	44	23	42	22	80	
30	53	32	50	30	95	
50	74	53	70	50	130	
70	95	74	90	70	170	
90	116	95	110	90	210	
110	137	116	130	110	250	
130	160	137	152	130	290	
170	210	180	200	170		
210	255	220	240	210		
250	300	265	285	250		
310	370	325	350	310		
410	475	430	450	410		
500	590	525	560	500		
600	710	630	670	600		
700	820	735	780	700		

NOTE.—The tabulated voltages are minimum voltages and are not subject to any tolerances They relate to tests under standard armospheric conditions, or to test figures to which corrections have been applied.

TABLE II.—INSULATOR RATING NUMBERS RECOMMENDED FOR USE IN CLEAN ATMOSPHERIC CONDITIONS.

Working Voltage of System. Declared Voltage (3-phase System).	Working Voltage of System. Approximate Voltage to Earth.	Insulator Minimum Rating Number.
kV.	kV.	
3·3	1·9	16
6·6	3·8	22
11	6·3	30
22	12·7	50
33	19	70
44	25	90
55	32	110
66	38	130
88	50	170
110	63	210
132	76	250
165	95	310
220	127	410
275	160	500
330	190	600
385	22	700

Porcelain Insulators for National Grid Lines.

For the national 132 kV. transmission scheme, insulators of the cap and pin type with ball and socket fittings have been selected. Each suspension string consists of 9 units and for tension insulators 10 units. The factor of safety is not less than 2½ when the insulator is supporting its maximum working load (4,000 lbs. on suspension and 8,000 lbs. on tension insulators) with full normal voltage applied to it. The minimum flash-over voltage of a unit is 85 kV. dry and 60 kV. wet. The minimum puncture voltage is 145 kV. per unit. The complete strings with all fittings have minimum dry flash-over voltages of 395 kV. and 450 kV. for suspension and tension types respectively, and 385 kV. and 440 kV. wet flash-over voltages. Since the normal voltage to earth is 76 kV., this gives a nominal factor of safety of 5. Each chain of insulators has a guard ring at the conductor end and arcing horns at the earthed end with the object of equalising the voltage distribution over the units and preventing damage to clamps, conductors, and insulator when a flash-over occurs. Two insulator strings in parallel are used at tension points.

Measurement of High Voltage.

Rules for the measurement of voltage with sphere gaps are given in B.S. No. 358—1939.

Standard Rubber Cables.

Rubber insulated cables were first standardised by the Cable Makers Association 1900. C.M.A. Standard cables had one layer of pure rubber adjacent to the tinned conductor and two layers of superimposed vulcanised rubber. Up to 250 volts, these cables were supplied in the 600-megohm and the 2,500-megohm grade. A cheaper quality 'Nonazo' of the 600-megohm class was also manufactured.

New Standard Cables were introduced in 1937. In them the pure rubber layer has been dropped and the standard dielectric for the C.M.A. grade is now a vulcanised rubber compound applied in three layers. The 'Nonazo' class has similar compounds, but not of such high quality.

The classification by megohms is discontinued, the cables being classified by the voltage they are designed for.

Sizes.—The maximum size in 250-volt, C.M.A. grade is 19/0·064 (0·06 sq. in.). Larger sizes are made only for 660-volt and higher-voltage grades.

In the 'Nonazo' class, the maximum for single-core cables is 19/·064, and in the case of twin and 3-core, flat, lead-covered and tough-rubber-sheathed the maximum size is 7/·064.

Flexible Cords.—These have three-layer vulcanised rubber dielectric. Unkinkable twin-and 3-core domestic flexible cords are standardised.

Cables for Installation Work.

As far as electrical installation work, and particularly conduit work, is concerned, the type of cable with which the contractor has mostly to deal is that known as a rubber insulated cable. The real properties of rubber insulated cables are more or less obscure, and the question as to what constitutes a good cable is almost impossible to answer with any degree of accuracy. To give some indication as to the quality of the rubber compound covering, a careful chemical test is required, and this combined with a stretching test will give a distinct clue as to the possible durability of the material.

GOVERNMENT DEPARTMENT SPECIFICATION FOR RUBBER-COVERED CABLES J, K AND L CLASSES FOR POWER AND LIGHT.

Designation.—Cables shall be designated by the nominal sectional area of the conductor and a letter prefix as follows: 'J Low' and 'J Low Twin,' 'J Medium' and 'J Medium Twin' for 250-volt and 660-volt respectively for V.I.R. taped and braided. 'K' and 'K Twin' for V.I.R. taped, and lead sheathed (660-volt) and 'L' and 'L Twin' for V.I.R. taped, lead-sheathed and armoured (660 volts).

Material.—The conductor shall comply in all respects with B.S. Specification No. 7—1939 and be free from all mechanical defects.

Tinning.—Each wire shall be tinned uniformly with pure tin free from lead in a bath free from contamination with copper. Before stranding, a sample shall be bent into a loop of radius not less than 12 and not more than 15 times its diameter, cleaned to remove grease and immersed for 1 minute in hydrochloric acid specific gravity 1·088 at 60° F., washed and examined under a lens for blackening. Average number of times of immersion to produce visible blackening shall not be less than 8 times for wire 0·029 to 0·048 inches diameter, 7 times for diameters above 0·048 up to 0·0808, and 5 times for diameters over 0·0808 inches. A further sample to be wrapped round a rod of diameter four times that of the wire and submitted to the above for one cycle.

Jointing.—Not more than one joint to be allowed in any wire forming 1000 yards of conductor nor a joint to be within 12 inches of another in same layer, nor in conductor after stranding. Joints shall be brazed, silver soldered or electrically welded.

Stranding.—To be uniform throughout length. 3-stranded wires to have a left-hand helical lay, 7 strand to have straight centre wire with six left-hand helically laid wires, the stranding for other numbers of wires to have alternate left-hand and right-hand lays.

Resistance of Conductors.—To conform to B.S. specification No. 7—1946. Compensation for the laying up of twisted twin cables to be agreed between contractor and inspector.

Dielectric.—Conductor shall be covered with one layer of British plantation pure rubber applied helically or longitudinally, then a covering and jacket of vulcanised rubber consisting of pure rubber mixed with sulphur, non-hydroscopic mineral matter and wax if required. The dielectric shall contain not less than 45 per cent. by weight of pure rubber. After vulcanisation the acetone extract shall not exceed 5 per cent. by weight, of which rubber-resins are not greater than 2 per cent. and free sulphur 0·75 per cent. Maximum amount of organic matter in residue after acetone extraction not to exceed 1·0 per cent. of total dielectric. Sulphur content in rubber not to be less than 1·25 or greater than 1·75 per cent. of total dielectric. No re-manufactured rubber or rubber substitutes to be used. Cable to be rejected if blackening is shown on pure rubber or tin, or if pure rubber adheres to conductor.

Thickness.—To be in accordance with B.S. specification No. 7—1946.

Taping and Vulcanising.—To consist of helical binding of fine cotton tape (red and black or blue for twin cables) of specified thickness, rubber proofed. The whole to be vulcanised into homogeneous body, after which rubber layers shall be inseparable but tape removable without tearing rubber.

Mechanical Tests.—After removal of the conductor, an outer covering a length of 2 inches marked on the dielectric shall be maintained stretched to 6 inches for 24 hours at an ordinary temperature not exceeding 65° F. After release, the length shall not be greater than 2¼ inches, no signs having been shown of the rubber cracking or elongating irregularly. Tensile strength not to be less than 800 lb. per sq. in.

Heat Tests.—Separate samples after removal of conductor shall withstand without material deterioration a dry heat of 270° F. for 1 hr. and a moist heat of 320° F. for 3 hrs. After three days interval samples to be stretched as above from 2 ins. to 5 ins. for 6 hrs., the distance becoming not greater than 2¼ ins. on release.

Electrical Tests.—(*a*) High voltage test as B.S. Specification No. 7—1946 after vulcanising and before outer protective coverings are put on. (*b*) Insulation resistance shall not be less than that given in tables and to be measured immediately after test (*a*). Cable to be immersed in water and measurement made with not less than 500-volt direct current after 1 minute negative electrification. The electrification must proceed regularly.

Twinning.—For circular twin, the complete cores shall be twisted with a right-hand lay, wormed with tanned jute and wrapped to approximate circular section with rubber-proofed cotton tape. For flat twin, the two cores side by side are tape-wrapped to form symmetrical cable.

Mechanical Protection.—The specification gives particulars for braiding, identification cotton, lead sheathing, bedding and serving, galvanised steel wire and steel tape armouring, tests on finished cables and packing, together with tabulated data and a temperature correction table for insulation resistance.

The foregoing details give some indication of the main features which are to be looked for in a high-class rubber insulated cable. As regards the conductors, the conductivity question is one of considerable importance, but it should be mentioned that trouble in this direction very seldom arises, because there is little difficulty in maintaining a standard purity in electrolytically refined copper. The phrase 'to show no sign of sulphide of copper after vulcanising' refers to the appearance of the wire when it is stripped of its insulating covering. It should then present a clean, bright appearance, free from blackening.

The building up of the rubber dielectric in three layers represents the usual English practice, and the wording of the specification is designed to exclude the use of adulterants and deleterious substances, these being practically synonymous so far as the life of the rubber insulation is concerned.

The specifying of a minimum percentage of rubber in addition practically leaves no room for anything but legitimate compounding ingredients, the choice of which is left to the experience and skill of the cable manufacturer.

There is, perhaps, no industry so greatly dependent on detailed experience and skill in applying that experience as that of the manufacture of rubber, and when electrical conditions are involved this is doubly true.

Generally speaking, therefore, where considerations of quality and durability are paramount it will always be best to purchase rubber-insulated cables, either to a specification such as that instanced above, or to agreed standards such as those of the Cable Makers' Association.

The provision of suitable outer coverings for rubber-insulated wires and cables which are exposed to abnormal conditions is a point which should not be overlooked. Conversely, in a given situation, a certain method of installing may be unsuitable. For instance, condensation effects, which occur in tubes and pipes, are notoriously bad for rubber cables, and in such situations either the method of installing should be varied or special coverings, such as a bitumen sheath, should be applied to the cables. Many cable makers are prepared to advise as to the best type of covering and method of running or fixing cables under abnormal surrounding conditions.

A second quality of vulcanised rubber cable has been placed on the market to meet the demand of those users who are satisfied with a cheaper grade of cable. Such cable, known as 'non-Association' cable, is quite satisfactory when properly manufactured, and may be safely used in cases where only a moderate length of life combined with good mechanical properties, is required, and where the conditions do not require a permanent maintenance of high elasticity of the dielectric.

Revised Specification for Cab-Tyre Sheathed Cable.*

The following specification for this class of cable is based on the British Admiralty Specification Tests for Cab Tyre Sheathing :—

The cab-tyre sheathing shall contain not less than 35 per cent. and not more than 40 per cent. of first-grade plantation rubber or other rubber of not less good quality.

The surface of the sheathing shall not be treated or dressed without the consent of the Admiralty Overseer.

The acetone extract of the vulcanised sheathing shall not exceed 5 per cent. calculated on the weight of the complete sheathing as manufactured.

Bending Test.—A test length of the complete cable containing the conductor shall be bent round a hard rod having the same diameter as the external diameter of the cable to form a loop. The unbent portions of the test-piece shall be bound together so that the two parts touch one another closely throughout their length from their ends up to the beginning of the loop.

In cables of large size when this method is not practicable the test-piece shall be bent in a loop so that the maximum extension of the external surface at the curvature at the bend reaches 50 per cent.

The sample in either case shall be left untouched for a period of 48 hours and at the termination of this time shall show no signs of cracking.

Immersion Tests. (*a*) *Chemical.*—Test-pieces 6 ins. long shall be prepared, and after the conductor and dielectric have been removed they shall be dried at a temperature of 100° C. for one hour in an air oven ; they shall then be cooled in a desiccator for two hours, and at the end of this period shall be weighed. The ends of the test-pieces shall be plugged with rubber bungs and sealed with paraffin wax for a length of half an inch from each end. The test-pieces thus prepared shall be completely immersed in a vertical position in the following solutions for a

* Referred to in I.E.E. Rules as 'tough rubber compound' covering.

period of 48 hours at normal temperature; they shall then be removed and washed in running water for a period of two hours, afterwards being wiped dry and the wax and plugs removed. The samples shall then be dried in an air oven for one hour at a temperature of 100° C. and shall be cooled for two hours in a desiccator; they shall then be reweighed. The difference in weights before and after immersion shall be calculated as grammes per 100 sq. ins. of surface exposed to the solution.

(1) *Acid Test.*--The solution shall be hydrochloric acid having a specific gravity of 1·045. The figure calculated as stated above shall not exceed 1·0.

(2) *Alkali Test.*—The solution shall be sodium hydrate having a specific gravity of 1·116. The figure calculated as stated above shall not exceed 1·5.

(*b*) *Oil Test.*—A test-piece shall be prepared 6 ins. long; one end shall be sealed so as efficiently to prevent access of oil to the interior. The test pieces shall be immersed in pure rape oil of the best quality to such a depth that not less than 2 ins. of sheathing is exposed to the action of the oil. The immersion shall be continued for six hours, the oil during that time being maintained at a temperature of 100° C. After immersion the sample shall be removed and rolled dry between blotting paper; the seal shall then be removed and the sample re-weighed.

The difference in weights shall be calculated as grammes per 100 sq. ins. of surface exposed and the figures so obtained shall not exceed 15.

The cable should be generally in accordance with **St. Helens Cable & Rubber Company's** original Patent No. 3998/1911.

Cab-Tyre Sheathed Cables are now made in 'Nonazo' (non-Association) quality, as well as in C.M.A. grade.

WIRING FITTINGS.

The following B.S. specifications have been issued in connection with steel conduits, plugs and sockets, lampholders and other fittings:—

- No. 31—1940, Steel Conduits and Fittings.
- „ 74—1937, Charging Plugs and Sockets for Battery Vehicles.
- „ 196—1930, Reversible Two-pin Plugs and Sockets.
- „ 279—1932, Flameproof Plug and Socket.
- „ 372—1930, Side-entry Wall Plugs and Sockets.
- „ 52—1941, Bayonet Lampholders.
- „ 98—1934, Edison-type Lampholders.
- „ 67—1938, Ceiling Roses.
- „ 91—1930, Cable Soldering Sockets.
- „ 94—1920, Watertight Glands for Cables.
- „ 97—1926, Watertight Fittings for Lamps.
- „ 542—1941, Cable Glands and Sealing Boxes for Mines.
- „ 546—1934, Two-pole and Earthing-pin Plugs and Socket Outlets.
- „ 562—1934, Reversible Connections for Portable Appliances.
- „ 617—1942, Identification of Conduits, Cables, etc.
- „ 1363—1947, Fused-plugs and Shuttered Socket-Outlets.

Conduits and Fittings for Electrical Installations.

Conduits, with the exception of the seamless or drawn varieties, are generally made from specially selected steel strip, varying in gauge according to the diameter of the tube. The strips are rolled to a circular section of exact external and internal diameter, by a method which ensures freedom from internal roughness and an even surface to facilitate the drawing of the wires into the conduit, thus preventing injury to their insulation. Exact sizes are essential in order to obtain a tight junction between the conduit and fittings with both socket and screwed junctions.

The conduits are made in two different types, viz. Light Gauge, for socket and continuity junctions, and Heavy Gauge for screwed junctions.

Conduits are usually supplied in two standard finishes, as follows:—

(1) Stove enamelled (black); (2) Galvanised.

For all ordinary installation work in the interior of buildings, enamelled conduits will be found satisfactory in working.

The enamel employed should be flexible, acid-resisting, insulating, and durable.

The conduits must be well cleaned before being enamelled, and subsequently stored at an even temperature until the enamel is set, so that it is not liable to flake or break away when the conduit is bent. The enamel should resist any chemical action which is likely to be found in the plaster of buildings.

Where conduits are subjected to excessive chemical action, especially when buried in cinder-fill concrete, quick-drying chemical plasters, and the like, a more efficient protection is required, and this is afforded by galvanising.

British Standard for Steel Conduits for Electrical Wiring.

(No. 31—1940.) (*Abstracts.*)

Steel conduits shall be either close-joint, brazed, welded or solid drawn as ordered, and shall be of mild steel and free from burrs and internal roughness. The conduit fittings shall be made of steel or malleable iron. All steel shall have a tensile strength of not less than 18 tons nor more than 24 tons per square inch of section, and an elongation of not less than 15 per cent. in a length of 8 inches. Malleable castings shall be well annealed and free from internal projections.

The conduit shall be manufactured in straight lengths of ten to fifteen feet.

Table of Standard Dimensions of Steel Conduits.

Conduit outside Diameter in ins.		½	⅝	¾	1	1¼	1½	2	2½
Threads per inch		18	18	16	16	16	14	14	14
Depth of thread (ins.)		·0356	·0356	·0400	·0400	·0400	·0457	·0457	·0457
Maximum length of thread on ends (ins.)		7/16	½	9/16	11/16	¾	13/16	15/16	1 1/16
Minimum length of thread on ends (ins.)		⅜	7/16	½	⅝	11/16	¾	⅞	1
Nom. thickness—Class A (plain) (ins.)		·040	·040	·048	·048	·056	·064	·064	·072
Min. thickness—Class A (plain) (ins.)		·036	·036	·044	·044	·052	·060	·060	·068
Nom. thickness—Class B (screwed) (ins.)		·056	·064	·072	·072	·072	·080	·092	·092
Min. thickness—Class B (screwed) (ins.)		·052	·060	·068	·068	·068	·076	·088	·088
Calculated weight per 100 ft., in lb., unenamelled and not including couplers	Class A (plain)	20	26	37	50	73	100	135	191
	Class B (screwed)	27	39	53	73	93	124	192	242

Table of Capacity of Conduits.*

(Type B, Screwed, B.S. No. 31, for the Drawing-in of V.I.R. Braided Cables.)

Size of Conduit (ins.)			½	⅝	¾	1	1¼	1½	2	2½
Internal Diameter (ins.)			0·388	0·498	0·606	0·856	1·106	1·34	1·816	2·316
Nominal Area of Conductor. Sq. In.	Number and Diameter (In.) of Wires.	Approximate Overall Diameter. Inch.	Maximum Number of Cables.							
0·0015	1/·044	0·150	2	4	6	10	14	—	—	—
0·002	3/·029	0·180	—	3	5	10	14	—	—	—
0·003	3/·036	0·200	—	2	4	8	12	—	—	—
0·0045	7/·029	0·210	—	2	4	6	10	—	—	—
0·007	7/·036	0·235	—	—	2	5	8	—	—	—
0·01	7/·044	0·270	—	—	—	4	7	—	—	—
0·0145	7/·052	0·300	—	—	—	3	5	6	—	—
0·0225	7/·064	0·345	—	—	—	2	4	6	—	—
0·03	19/·044	0·380	—	—	—	—	3	5	7	8
0·04	19/·052	0·425	—	—	—	—	2	4	6	7
0·06	19/·064	0·500	—	—	—	—	—	3	5	6
0·1	19/·083	0·630	—	—	—	—	—	—	3	4
0·15	37/·072	0·750	—	—	—	—	—	—	2	2

These capacities apply to runs of conduit which deflect from the straight by an angle of more than 15°.

* From I.E.E. Rules (Eleventh Ed.).

For all fittings made out of *tube*, the minimum thickness of the fittings shall be equal to that of the corresponding conduit with which they are used. In no case shall 'close joint' tube be used for these fittings.

Two classes of steel conduit for electrical wiring are recognised as Standard :—

Class A.—Plain.
Class B.—Screwed.

Class A consists of light gauge conduit of the thickness and dimensions given in the Table below. Class A conduit is either close-joint, brazed, welded or solid drawn. The coupler joining the lengths of tubing is a sleeve, and neither the ends of the conduit nor the coupler are screwed.

In the close-joint tubes, the edges of the steel strip, though brought tightly together in the process of manufacture, are not metallically joined in any way.

Class B consists of heavy gauge conduit of the thicknesses and dimensions given in the Table. Class B conduit is either brazed, welded or solid drawn. Both ends of the conduit are screwed.

Unless otherwise specified with the order, all conduits and fittings shall be stove enamelled jet black or galvanised by the hot process both inside and out.

Electricity Act, 1947.

The purpose of this Act was to co-ordinate under public ownership the electricity supply industry of Britain, in place of the previous arrangements where there were the Central Electricity Board, the North of Scotland Hydro-Electric Board, 550 company and local authority undertakings and ten joint authorities or boards. A *British Electricity Authority* has been established to control the generation and bulk transmission of power and fourteen *Area Electricity Boards* to distribute electricity to consumers. The *North of Scotland Hydro-Electric Board*, in its area, carries out both functions.

The B.E.A. consists of a chairman and four to six other members, the chairman of four of the Area Boards taken in rotation and the chairman of the North of Scotland Hydro-Electric Board. The B.E.A. has power, in addition to its main functions, to manufacture and sell or hire electrical plant and fittings. It also co-ordinates the activities of the Area Boards and exercises some general control over their policy.

The Area Boards buy in bulk from the B.E.A. (or from each other) and distribute to consumers. For administrative purposes, and to maintain local touch with local events, they have each split their territory into sub-areas, corresponding more or less at present with the areas of the previous undertakings. Within each Area Board's area a Divisional Controller represents the B.E.A., the divisions of the B.E.A. corresponding to the areas of the Area Boards. The Area Boards may sell, but not manufacture, and repair or maintain electrical fittings (but *not* plant).

The B.E.A. is empowered to standardise systems of supply and types of electrical fittings, and to standardise and simplify systems of charging for supplies.

Each Area Board must submit an annual report to the B.E.A. and the B.E.A. must submit an annual report to the Minister of Fuel and Power. Consultative councils are being established in each Area, each consisting of twenty to thirty persons, of whom half to three-fifths will be nominated by the Minister of Fuel and Power from persons suggested by local authorities and the remainder will represent other consumer interests. The Electricity Commissioners have been abolished.

The tariff for bulk supplies from the B.E.A. to the Area Boards is, at present, £3 10*s*. per kW. of maximum demand at each point of supply, plus 0·335*d*. per kWh., with a fuel cost variation of 0·0007*d*. per kWh. for each 1*d*. by which the price of coal (based on a calorific value of 11,000 B.Th.U. per lb.) varies from 38*s*. per ton. There will be a percentage adjustment to the total charge for each Area Board as follows (based on the different fuel prices in different Areas).

London	+ 3
South-Eastern	+ 7
Southern	+ 6
South-Western	+ 5
Eastern	+ 1
East Midlands	—
South Wales	+ 10
Merseyside and North Wales	+ 3
Yorkshire	− 2
North-Eastern	− 7
North-Western	—
South-East Scotland	− 4
South-West Scotland	− 8

Ministry of Fuel and Power Regulations in reference to Overhead Electric Lines.

Every application for the consent of the Ministry of Fuel and Power **to the placing of electric lines above ground should be accompanied by the following particulars :—**

1. Where the undertakers are a company, or a local authority supplying outside their own area, evidence of consent of the local authority for the district.

2. A statement showing commercial or other considerations why underground cables should not be used.

3. A brief description of the proposed system, whether by continuous or by alternating current; the working voltage; the kind of wire, whether copper or aluminium; whether solid or stranded; the total sectional area; tensile strength and elongation; average and maximum length of span; minimum height of wires from the ground; name or description of automatic protective device, if any.

4. A statement whether the supply is to form (1) an extension of an existing system of underground cables, or (2) of an existing traction system, or (3) an independent system.

5. An ordnance map on a scale of 6 ins. to the mile, showing the proposed route of the overhead lines and any existing overhead lines. The sheets of these maps must be fastened together.

6. In the case of high and extra high pressure, plans of construction of poles, etc., on a scale of about 1 in. to the ft., or a reference to previously deposited plans where these are identical with the proposed work.

Regulations of the Electricity Commissioners.

ELECTRICITY SUPPLY REGULATIONS—1937.*

REGULATIONS FOR SECURING THE SAFETY OF THE PUBLIC AND FOR INSURING A PROPER AND SUFFICIENT SUPPLY OF ELECTRIC ENERGY.

Lines and Systems for Low and Medium Voltages.

(*Low* voltage means not exceeding 250 volts and *medium* voltage not exceeding 650 volts.)

1. (*a*) All lines and circuits of the undertakers to be tested for insulation resistance after erection and before use.

(*b*) Insulation must withstand either (i) tests prescribed by British Standards Institution, or (ii) test voltage of 500 volts for 15 minutes between conductors and between conductors and earth.

(*c*) After disconnection for alteration or repair, undertakers must ensure that insulation of circuit is in sound condition.

(*d*) Undertakers must record result of every test.

2. Undertakers must maintain insulation of every system so that leakage current shall not exceed one-thousandth part of maximum supply current. Excessive leakage to be remedied without delay.

3. In a medium-voltage supply, the voltage between earth and any conductor shall not exceed 250 volts.

4. A point of any low-voltage A.C. system above 125 volts and of any medium-voltage system shall be connected with earth as follows:—

(*a*) The earth connection shall be made at one point only in each distinct system (unless otherwise approved by the Commissioners), and the insulation shall be efficiently maintained at all other parts.

(*b*) With concentric conductors, the external is the one to be earthed.

(*c*) The earth connection shall be efficiently maintained except when testing or locating a fault.

(*d*) In a D.C. system, an ammeter shall be permanently in the earth connection and a record of the current shall be kept by the undertaker. In a 3-wire, D.C. system, a fusible cut-out or automatic circuit breaker may be in the earth connection in parallel with a resistance of not more than one ohm.

(*e*) In an A.C. system, no impedance, cut-out or breaker shall be in the earth connection, Earth currents made on test must be recorded.

(*f*) A.C. systems connected to earth as above may be electrically interconnected provided (i) each earth connection is bonded to the metal sheathing or armouring of the lines concerned; (ii) the neutral conductors of overhead line systems are of the same material and cross-section as the phase conductors; (iii) generators or transformers not having a mesh-connected winding of low impedance which form part of any system have their neutral points connected to earth.

(*g*) Where an earthed A.C. system provides a low or medium voltage supply to an electrode boiler which is also earthed, the metal work of the boiler shall be bonded to the metallic sheathing or armouring of the lines concerned.

Lines, Systems and Apparatus for High Voltages.

5. Every high-voltage line of the undertakers shall be completed and tested before use, and shall be in their sole charge, unless otherwise approved by the Commissioners.

6. (*a*) All high-voltage lines, circuits and apparatus of the undertakers to be tested for insulation resistance after completion and before use.

(*b*) Insulation must withstand either (i) tests prescribed by British Standards Institution, or (ii) continuous application for 15 minutes between conductors and between conductors and earth either an alternating test voltage of 1½ times working voltage or the working voltage plus 10,000 volts, whichever be the less. With concentric conductors where the outer is to be earthed, the test voltage between outer and earth shall be 1,000 volts. If the neutral conductor of an A.C.

* Explanatory Notes on the 1937 issue of these regulations are published by H.M. Stationery Office. Price 3*d*. The Regulations are now issued by the Ministry of Fuel and Power, which has taken over the duties of the Commissioners.

system is not to be earthed, the working voltage between a phase and earth shall be taken to be the voltage between phases. The duration of test may be reduced to one minute if test voltage is not less than 1½ times working voltage, or equal to working voltage plus 20,000, whichever be the less. Direct voltage may be used if testing voltage be 50 per cent. greater than the prescribed alternating test voltage.

(*c*) After disconnection for alteration or repair, undertakers must ensure, before use, that insulation is sound.

(*d*) Results of every test must be recorded by undertakers.

7. (*a*) Every line and all associated apparatus to be provided with an automatic device to ensure immediate and safe discharge of energy in event of excess leakage.

(*b*) Lines, other than overhead and station lines, to be enclosed in metal sheathing, electrically continuous and effectively earthed.

(*c*) In event of failure of insulation between a conductor and the metal sheathing, the impedance shall be such that with full supply voltage the fault current shall not be less than twice the value at which the automatic devices operate.

8. A point of every high-voltage system shall be earthed thus: (i) the earth connection shall be made at one point only in each distinct system, unless otherwise approved, and the insulation shall be maintained at all other points; (ii) with concentric conductors, the external shall be the one earthed.

If such a high-voltage earthed system is used to supply an electrode boiler which is also earthed, then (i) the metal work of the boiler shall be bonded to the metallic sheathing or armouring of the lines concerned; and (ii) the high-voltage supply to the boiler shall be controlled by a suitable automatic circuit-breaker set to trip when the current unbalance in the phases continues to exceed 10 per cent., or in exceptional cases 15 per cent., of the rated current of the boiler.

Transformation and Control of Energy at High Voltage.

9. Undertakers may transform or convert high-voltage energy in sub or switch stations (including outdoor), in completely enclosed street boxes above ground, in street boxes under ground, in fire-resisting casings on consumers' premises, or on or near the supports of overhead lines; provided that:

(*a*) Such stations be preferably above ground, but ventilated and drained where below ground.

(*b*) Outdoor stations shall be protected by fencing not less than 8 ft. high or other means to prevent access to lines and apparatus by unauthorised persons.

(*c*) Underground boxes containing transformers shall not contain switches or other apparatus, which must be separately housed, preferably above ground.

(*d*) Fire-resisting casings on consumer's premises shall completely enclose lines and apparatus to prevent access by unauthorised persons, and shall be labelled with appropriate danger notice and with undertaker's name and address.

10. (*a*) Street boxes or fire-resisting casings on consumer's premises shall have doors or covers that cannot be opened without key or a special appliance. When open, it shall not be possible to come into contact with metal at high voltage.

(*b*) The conductors and apparatus unless completely enclosed and connected with the system by armoured lines) on the supports of overhead lines or adjacent to the lines shall have no live metal less than 15 ft. from the ground or any place accessible by an unauthorised person, or less than 15 ft. from any platform on which an authorised person may stand. Provision shall be made to prevent unauthorised climbing.

11. Where energy is transformed, provision shall be made—preferably by earthing a point on the lower-voltage system—to prevent danger from this system becoming charged from the high-voltage system.

Electric Lines and Apparatus (General).

12. All lines to be in accordance with appropriate current specification of British Standards Institution.

13. Every circuit of undertakers and all apparatus associated therewith shall be protected against excess energy by a fusible cut-out or other automatic circuit breaker. In concentric conductors no such device shall be inserted in any earthed external conductor.

14. Where an electric line crosses or is near to any pipe, line or other metal, precautions must be taken by undertaker to prevent the latter becoming alive. Any metal work associated with lines (except serving as conductor) to be earthed where necessary to prevent danger.

15. Overhead lines to be erected and maintained in accordance with Commissioners' Regulations.

16. (*a*) In delivering energy, undertakers shall take precautions to avoid risk of causing fire.

(*b*) Where fire risk to a building is involved, and there is more than 2,000 gallons of oil in any one transformer or switch tank, provision must be made for removing any oil which may escape from the tanks; precautions must be taken to prevent spread of fire from ignition of oil, also to extinguish fire. Spare oil must not be stored in the station.

17. Regarding lines not completely enclosed in a continuous metallic earthed sheathing:

(*a*) In cases where prior to these Regulations lines were in use which are insulated or protected *in situ* by composition or bituminous material: (i) such lines shall be frequently tested and inspected, the results being recorded by the undertakers; (ii) any pipe or circuit in which the line is placed must be sealed at point of entry into any street box to prevent flow of gas from or into the pipe or circuit or into or from the street box.

(b) In future no lines insulated or protected *in situ* by any composition or material known to be liable to produce noxious or explosive gases on excessive heating shall be installed.

18. Conduits, pipes, casings, street boxes, etc., used as receptacles for lines or apparatus shall be durable, and where placed under carriageways of ample strength.

19. In addition to Regs. 9 and 10, street boxes must comply with the following:

(a) Street boxes shall not contain gas pipes, and reasonable means shall be taken to prevent influx of water or gas.

(b) Lines forming part of different systems and passing through the same street box shall be readily distinguishable and supported and protected to minimise risk of damage to or from adjacent lines.

(c) All street boxes to be regularly inspected for presence of gas, and if any influx or accumulation is discovered the undertakers shall give immediate notice to the relevant gas authority. Where a person can enter the box, provision must be made (i) that any gas shall escape before the person enters, (ii) for prevention of danger from sparking.

(d) In this Regulation, street box includes underground sub or switch stations.

20. Where access to a line is obtained through underground ways not regularly inspected, no person shall enter until the undertakers have made tests for presence of noxious or explosive gases, and expelled same if present.

21. The supply system shall be separated into sections, and where necessary provided with circuit breakers or fusible cut-outs so as to restrict reasonably the area of supply affected by a failure.

Connection with and Supply to Consumer's Premises.

22. Service lines entering consumer's premises below ground level must not permit influx of gas at point of entry.

23. The separate conductors shall be permanently marked by colouration or labels near the supply terminals to indicate clearly polarity or the neutral and live phase conductors.

24. For protection against excess energy, a fusible cut-out or other automatic circuit-breaker, completely enclosed in a locked or sealed fireproof receptacle, shall be inserted by undertakers in each service line in a suitable position close to supply terminals.

No such device shall be inserted in any neutral conductor at a point common to more than one pair of conductors of a 3-wire or multi-phase system. With high-voltage supply, provision must be made (a) to isolate the cut-out or breaker from the service line, and (b) that the consumer can cut off all voltage from the supply terminals without risk of danger.

25. Lines and apparatus on consumer's premises and belonging to or under control of undertakers must be maintained by undertakers in a safe condition and protected to prevent leakage to any adjacent metal. Lines and apparatus which comply with the I.E.E. Regulations for the Electrical Equipment of Buildings are deemed to satisfy this Regulation. Lines and apparatus on consumer's side of supply terminals and forming whole or part of consumer's installation shall be subject to any agreement made between undertakers and consumer with respect to hire or hire-purchase. Nothing in this Regulation shall relieve the owner or occupier as consumer from any obligation imposed by Home Office or Mines Department Electricity Regulations.

26. The undertakers shall not permanently connect a consumer's installation with their lines unless they are reasonably satisfied that such connection would not cause a leakage from consumer's installation exceeding 1/10,000th part of the maximum current to be supplied to said installation.

27. (a) Undertakers shall no be compelled to give a supply unless satisfied in respect of consumer's installation:—(i) that all conductors and apparatus are sufficient in size and power for the intended purposes, and are constructed, installed and protected reasonably to prevent danger; (ii) that every distinct circuit is protected by a fusible cut-out or other automatic circuit breaker, suitably rated and constructed and readily renewable; (iii) that every motor is controlled by a switch for starting and stopping, and placed to be easily worked by the person in charge of the motor. Any consumer's installation complying with the I.E.E. Regulations for the Electrical Equipment of Buildings shall be deemed to satisfy this Regulation.

(b) The above provisions of this Regulation shall not apply to cases were the provisions of the Home Office or Mines Department Electricity Regulations are applicable.

These last two provisions apply also to Regulations 28, 29 and 31.

28. (a) A supply of energy at low voltage from more than one pair of conductors of a 3-wire or multi-phase system at medium voltage shall not be given unless the total rating of consumer's installation exceeds 8 kW., and then only if necessary to comply with Regulation 34.

(b) Undertakers shall not in any case be compelled to give a low-voltage supply to any consumer from more than one pair of conductors unless satisfied: (i) that the supply terminals are arranged in separate pairs so as to avoid danger of shock at medium voltage; (ii) that the consumer's wiring connected to the separate pairs of supply terminals is kept separate and distinct, or complies with Regulation 29; (iii) that in any room containing the different pairs of conductors, any apparatus connected to one pair is fixed at least 6 ft. away from any apparatus conected to the other pair; or, alternatively, that the metal framework of all such apparatus is earthed, and that any flexible conductor attached to portable apparatus is protected against mechanical damage. Any metallic covering of conductor shall be earthed, but this shall not constitute the only means of earthing the metal framework.

29. (a) Undertakers shall not be compelled to give a medium-voltage supply unless satisfied:—(i) that all metalwork associated with consumer's installation, unless serving as a conductor, is

earthed; (ii) that all consumer's wiring is completely enclosed in electrically continuous metal sheathing, or, alternatively, installed and protected to prevent danger; (iii) that each motor or separate piece of apparatus is controlled by a cut-off switch readily accessible to the person in charge, and arranged so that it can cut off all voltage from the motor, apparatus or associated device.

30. (*a*) A high-voltage supply shall not be given unless: (i) all high-voltage conductors and apparatus on consumer's premises are inaccessible to consumer, and all operations in connection with same are carried out by undertakers by arrangement with consumer; (ii) the consumer gives a written guarantee to undertaker that an authorised person will be in charge of consumer's high-voltage installation when in use, and that the latter will be efficiently maintained, and if required to the undertaker's satisfaction, and that instructions regarding treatment of persons suffering from electric shock will be affixed in consumer's premises.

(*b*) High-voltage energy shall not be supplied for water-heating in which a live element is in contact with the water without the consent of the Commissioners.

(*c*) Undertakers are not compelled to give a high-voltage supply unless they are reasonably satisfied: (i) that no metal at high voltage is exposed so that it can be touched; (ii) all high-voltage conductors other than overhead lines are completely enclosed in electrically-continuous metal protected against mechanical damage; (iii) all metalwork associated with consumer's installation, except conductors, is earthed; (iv) that the supply to each motor or separate piece of apparatus be controlled by a cut-off switch readily accessible to the person in charge, and connected so that all voltage can be cut off from the motor, apparatus or associated device; (v) that all high-voltage windings within reach are protected to prevent damage; (vi) that the low-voltage side of transformers be safeguarded—preferably by earthing—against becoming accidentally charged at the higher voltage; (vii) that unless the whole high-voltage installation can be made dead at the same time for cleaning or other work, it must be sectionalised and the sections must be divided or screened from one another so that work can be done safely on one section without danger from another; (viii) that adequate working space is provided in front of any switchboard (except low voltage) and at places where live conductors can be exposed; (ix) that means be provided to prevent access to any part of high-voltage installation by an unauthorised person.

(*d*) Paragraphs (*a*) and (*c*) do not apply where Home Office or Mines Department Electricity Regulations are applicable.

(*e*) Undertakers must notify the District Factory Inspector of intention to supply high-voltage energy to premises to which Home Office Regulations apply.

31. (*a*) Undertakers shall not be compelled to supply any consumer who proposes to transform the energy to a high voltage for an outside luminous tube sign unless they are satisfied (i) that cut-off switches on the lower voltage side are provided inside and outside the premises, easily accessible to the person in charge, and connected so as to cut off all high voltage; (ii) that no high-voltage metalwork be exposed so that it can be touched; (iii) that all high-voltage conductors (other than overhead lines and series connecting wires) are completely enclosed in electrically continuous metal protected where necessary against mechanical damage; (iv) that all such enclosing metal is earthed; (v) that all high-voltage windings within reach are protected; (vi) that the low-voltage side is protected, preferably by earthing, from becoming accidentally charged at the higher voltage; (vii) that unless the whole installation can be made dead simultaneously for cleaning or other work, it must be sectionalised so that work can be done on any dead section without danger; (viii) that means are provided to prevent access to any high-voltage part by an unauthorised person.

32. (*a*) Where undertakers after examination have reasons to suppose that a leakage exceeding the amount mentioned in Regulation 26 exists at some part of a consumer's installation, or that the installation or part of it is not in sound condition, the following provisions shall have effect:—(i) where immediate action is justified as a work of emergency in interests of public safety or to avoid undue interference with supply to other consumers, undertakers may discontinue supply forthwith, giving immediate notice in writing to consumer and specifying matter complained of; (ii) in any other case, undertakers may by notice in writing require consumer to permit inspection and test between 9 A.M. and 6 P.M. If consumer refuses facilities, or if after inspection and testing, the undertaker's officer reports excessive leakage or defective installation, the undertaker may in writing specify the complaint and require the consumer to remedy the same within a specified reasonable period. Should the consumer fail to show that the matter has been remedied, the undertaker may on expiration of said period, but subject as hereinafter provided, discontinue the supply, giving immediate notice to consumer; (iii) if any question arises between consumer and undertaker regarding any matter to be remedied, either party shall take immediate steps to have the question settled as provided for by Regulation 33; (iv) in exercising powers in (ii), undertakers shall not discontinue supply pending settlement referred to in (iii), and shall in no case discontinue supply to the whole of consumer's installation, where it is feasible for them to discontinue the part complained of, provided nothing shall prevent undertakers exercising powers in (i); (v) where in pursuance of this Regulation the supply has been discontinued, it shall not be recommenced until undertakers are satisfied, or until it has been decided as provided in Regulation 33.

(*b*) This Regulation shall extend as far as practicable to any lines and apparatus on consumer's premises and belonging to undertakers or under their control within the meaning of Regulation 25; and in case of discontinuance the undertakers shall forthwith remedy any defect,

subject to the terms of any agreement between undertakers and consumer regarding hire or hire-purchase.

(c) This Regulation as regards order or condition shall not apply where the Home Office or Mines Department Electricity Regulations are applicable.

33. (a) Where undertakers decline to give a supply, or to recommence after discontinuance, they shall serve on the consumer a notice in writing stating reasons.

(b) Any question arising between a consumer and undertakers under (a) or under Regulation 32 shall be determined by an Electric Inspector appointed by the Electricity Commissioners on the application of either party, and the inspector shall determine by which of the parties the costs shall be paid.

(c) Either party may appeal against the inspector's report to the Electricity Commissioners, whose decision shall be final and binding on both parties.

34. (a) Before commencing a supply, the undertakers shall declare to the consumer: (i) type of current, whether direct or alternating; (ii) in case of A.C., the number of phases and the constant frequency; (iii) the constant voltage of the supply.

(b) The declared type of current, number of phases, frequency and voltage shall be constantly maintained, subject to a permissible frequency variation of $\pm 2\frac{1}{2}$ per cent. and a permissible voltage variation of $\pm$ 6 per cent. These shall not be departed from without the Commissioners' consent.

(c) Public notice of any application of undertakers to the Commissioners for any alteration in type of current, number of phases, frequency or voltage must be made in accordance with the consumers' requirements.

35. From the time when undertakers begin a supply through any distributing main, they shall maintain a supply sufficient for all consumers entitled to be supplied. Supply shall be maintained without change of polarity in D.C. and without change of neutral in A.C. For testing or any other purposes connected with the efficient working of the undertaking, the supply may be discontinued by the undertakers for such period as may be necessary subject (except in cases of emergency) to not less than 24 hours' notice to all consumers likely to be affected. In event of any consumer objecting, the supply shall not be discontinued except with consent of the Commissioners. Provided also that the polarity in case of D.C., and neutral in case of A.C., may be changed with the like consent.

36. Undertakers shall keep printed copies of these Regulations and supply a copy thereof to any person demanding same at a price not exceeding that paid by themselves.

37. Inspections, examinations and tests may be made by authorised servants of the Commissioners.

38. (a) Undertakers must report to Commissioners any accident as soon as possible—failure to do so renders undertakers liable to a penalty of 20*l*. for each default.

(b) The Commissioners may appoint an inspector or other person to hold an inquiry into any accident affecting the safety of the public.

39. Relates to penalties imposed on undertakers failing to comply with these Regulations.

Regulations of Electricity Commissioners for Overhead Lines.*
(El.C. 53—1947.)

General Regulations.

1. Conductors to be of copper, aluminium, or other approved material.

2. Conductors to conform to B.S. specification for elongation, breaking load, and elasticity.

3. Minimum permissible size for copper and other line conductors (other than service lines) must have an actual breaking load of not less than 1,237 lbs. Minimum area and weight per mile for copper, No. 8 S.W.G., 0·0201 sq. ins., 409 lbs. For service lines, minimum breaking load 816 lbs.; minimum area and weight, No. 10 S.W.G., 0·0129 sq. ins., 262 lbs. per mile.

4. Conductors must be rendered inaccessible to persons not having the use of appliances. Regard must be paid to the normal use of his land or premises by the occupier. See 18.

5. Except in the case of contact between broken conductors and earth wire on the same support, precautions must be taken by the undertakers against contact between line conductor and any other overhead wire.

6. Line insulators to be carried on supports of wood, iron, steel, or reinforced concrete. Corrosion of metal work on or below ground surface to be prevented. Wooden supports must be of red fir, except oak or hardwood cross arms, unless otherwise approved.

7. Supports, in conjunction with any struts or stays, shall take all loads without damage or movement in the ground. Strength in direction of line must not be less than one quarter of transverse strength. Factor of safety for iron and steel 2·5, for wood and reinforced concrete 3·5. These factors to be calculated for 22° F. with ice covering specified in 12 or 15, and with a wind pressure of 8 lbs. per sq. ft. on whole of the projected surface, corresponding to a velocity of 50 m.p.h. at right angles to the line. Wind pressure on lee side members of lattice or other compound structures, A and H poles, to be taken as one-half that on windward side, and factor of safety calculated on crippling load of struts and upon elastic limit of tension members

8. Service lines to be connected only at a point of support, and fixed to insulators on consumer's premises. Service lines (other than an earthed neutral conductor) which are accessible from a building must be protected by insulating material or otherwise.

* Now issued by the Ministry of Fuel and Power.

Service lines over carriage ways must not be lower than 20 ft. from the ground except with consent of Commissioners.

9. Adequate protection to be made against danger to linesmen, and from a lower voltage system becoming charged above normal from a higher voltage system when conductors of the different systems are carried on same poles or supports.

10. The whole structure with all appliances must be regularly inspected and maintained.

11. Materials must conform at time of erection to B.S. specifications and to the Post Office Technical Instructions of Aerial Lines.

Specific Regulations.

A. For pressures not exceeding 650 volts D.C. and 325 volts A.C.

12. Factor of safety of line conductors, 2, based on breaking load at 22° F., with $\frac{3}{16}$ in. radial thickness of ice and simultaneous wind pressure as in 7.

13. Minimum height of any wire (except service) at 122° F., not to be less than 19 ft. across a public road or 17 ft. in other positions. In places inaccessible to vehicular traffic, 15 ft. Minimum height of a service line across or along a carriage-way, except with permission, 19 and 17 ft. respectively.

14. Where pressure to earth exceeds 250 volts D.C. or 125 volts A.C., precautions shall be taken to prevent danger
- (*a*) from a broken conductor; by a neutral or earthed conductor suitably disposed or other approved method:
- (*b*) from leakage; in cases of metal poles, by provision of earthed wire connecting the poles, or by supporting the insulators on insulated metallic frame connected to neutral conductor. With wooden poles by a bonding wire connected to supporting metal of all insulators and terminating at lowest part of supporting metal work. Other means may be approved.

Where lighting or uninsulated conductors are run down wooden poles to within 10 ft. of the ground, precautions against leakage must be as for metallic poles. Unearthed stay wires must be insulated by insulator placed not less than 10 ft. above ground.

B. For pressures exceeding 650 volts D.C. and 325 volts A.C.

15. Factor of safety of conductors as in 12, except that for $\frac{3}{16}$ in. ice, read $\frac{3}{8}$ in.

16. Minimum heights of conductors above ground at temperatures below 122° F are 20 ft. for pressures not exceeding 66,000 volts, 21 ft. exceeding 66,000 and below 110,000 volts, 22 ft. 110,000 to 165,000 volts, and 23 ft. above 165,000 volts. Height from the ground of any earth wire must not be less than the minimum prescribed in Regulation 13.

17. Means must be taken to render broken conductors dead. All other metal work to be earthed by continuous earth wire earthed at four equidistant points per mile; alternatively metal shall be effectively earthed at each support. When earth contact is made by a line conductor, leakage current must be at least twice that required to operate safety devices.

18. At road, canal, or railway crossings, in addition to 16, there shall be provided (*a*) duplicate line insulators with automatic earthing of a fallen conductor or (*b*) duplicate insulators tied at intervals not exceeding 5 ft.

Where line is erected along or within 50 ft. of road, canal, or railway, (*a*) duplicate insulators or (*b*) automatic earthing for fallen conductor shall be provided. In the case of a line crossing over any other overhead wire, (*a*) duplicate line insulators with automatic earthing of a fallen conductor, (*b*) duplicate insulators tied at intervals not exceeding 5 ft.; together with arcing horns or rings for line conductors above 650 volts. Other means may be approved.

19. Supports must be consecutively numbered and carry a permanent danger notice.

20. Anti-climbing devices must be fixed to all supports other than single wooden poles, to all single wooden poles carrying transformers, fuses or switchgear and to stay wires arranged near each other.

21. Lines may be in accordance with B.S. 1320—1946, which has priority over the Electricity Supply Regulations, 1937. This Specification deals with a design of light line for rural electrification prepared by the Electrical Research Association.

An explanatory memorandum El.C.53A is issued.

The Electricity Commissioners have also issued a number of relaxations of their Regulations to facilitate rural electrification. These deal with lines of 8 S.W.G. to 0·04 sq. in. section. Under M 2854 of 24th September, 1937, such lines may be designed for a wind-loading of 8 lb. per sq. ft. with $\frac{3}{16}$ in. radial thickness of ice. The minimum diameter of the pole 5 ft. from the butt is to be $7\frac{1}{2}$ ins. Under A 950/820 of 16th September, 1942, lines may be designed for a wind load of 16 lb. per sq. ft. on the bare conductor, with a factor of safety of $2\frac{1}{2}$ on the conductors and $3\frac{1}{2}$ on the poles ($2\frac{1}{2}$ on the poles for very light lines), and with a minimum ground clearance of 17 ft. for 11 kV. working.

The most favourable regulations may be chosen by those erecting a line.

Regulations Relating to Extra High Pressure. (El.C. 13—1920.)

The following *additional* regulations relating to *extra high pressure* have been issued by the Commissioners.

A. *Regulations for Securing the Safety of the Public.*

3. An extra-high-pressure main before being brought into use shall be tested in position to withstand for half an hour continuously a pressure of 10,000 volts in excess of the working voltage, when this exceeds 10,000 volts, or twice the working pressure when this is less than 10,000 volts. Results of tests must be recorded.

4. Suitable fuse or circuit breaker must be provided for every main, but not in earthed outer conductor of concentric mains.

8. Where leakage from 3-phase mains is prevented by a copper strip under the lead sheath of cable, this shall not be less that 0·016 in. thick, and where steel wires are thus employed outside a lead sheath, each shall not be less than 0·1 in. in diameter. A lead sheath must be efficiently earthed and not less than 0·1 in. thick.

9. Single-phase supply mains to consist of two concentric conductors or separate conductors. Outers of concentrics to be earthed at one point: with separate conductors, leakage to be prevented as with 3-phase mains.

Street Mains.

11. These mains must be enclosed in strong metal casing where passing through street box with other mains; boxes must not contain any other undertaker's electric mains, nor water or gas pipes. Telephone wires belonging to the undertakers are allowed.

Sub-stations.

12. Must be in sole or joint occupation of the undertakers or authorised distributor.

13. When constructed below streets, must contain no switches or apparatus other than transformers.

14. The transforming apparatus must be arranged to avoid danger of excessive voltage on connected mains.

15. Precautions must be taken against fire risks.

17. Gives right of entry and examination to the Commissioners.

Connection of Circuits with Earth.

18. Earth connection to be made at only one point on a circuit, viz. generating station, sub-station, or transformer. Insulation must be maintained elsewhere.

19. When neutral star point of each distinct 3-phase circuit is earthed through a resistance, this must be low enough to ensure the action of the cut-out in the mains.

Separate electrostatic voltmeters must be connected in the generating station between each distinct circuit and earth to act as leakage indicators. Faulty circuit insulation must be immediately restored.

20. Penalties are prescribed against default which are not to affect liability of undertakers in respect of damages caused by default.

B. *Regulations for Ensuring a Proper and Sufficient Supply of Electrical Energy.*

1. Undertakers must maintain sufficient and constant supply from the main to authorised distributors and consumers.

2. Maximum load on a main not to exceed 1,000 kW. without consent of Commissioners, unless provision is made for alternative supply in the event of breakdown.

3. Minimum pressure to be declared. Pressure at authorised distributors' terminals must not be less than this minimum, and not exceed it by more than 12½ per cent. Alteration of minimum to be allowed only by consent of Commissioners after one month's public notice of such application.

4. Frequency to be declared 50 or 25 cycles per second; 2½ per cent. variation allowed. Consent and notice of alteration as in 3.

5. Penalties are prescribed for default.

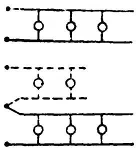
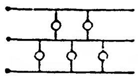
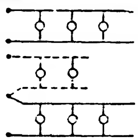
These regulations are in addition to any regulations relating to mines and factories made by any Act, and also include regulations 9, 16, 18, 20, and 27 of El.C. 38, summarised above.

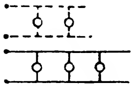
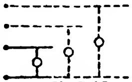
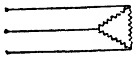
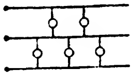
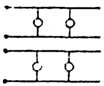
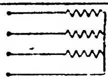
The Electrical Equipment of Buildings.

Regulations Issued by the Institution of Electrical Engineers.

The Eleventh Edition was issued in 1939, and amended in 1943 and 1945. A Supplement, altering some of the Regulations, was issued in 1946. A Twelfth Edition is now being prepared to take into account the use of ring circuits and of discharge lighting, and other matters. The abstract (p. 1430) is the latest available.

CONTROL OF THE SUPPLY FROM AN EXTERNAL SOURCE.

Nature of Consumer's Wiring and of the Supply			Main Switchgear to Control Consumer's Wiring			
			Type of Circuit Breaker (linked) to be installed		Alternative to Circuit Breaker	
Wiring from Consumer's Terminals	Diagram showing Nature of Supply and Consumer's System of Wiring	Earth Connection of Supply System (See NOTE after Regulation 116)	No. of Poles	Overload Trip	Switch (linked)	Cut-Out†
1.	2.	3.	4.	5.	6.	7.
D.C. 2-wire	D.C. 2-wire supply D.C. 3-wire supply 	None, or through a resistor, cr otherwise	2*	In each conductor	2-pole	In each conductor‡
D.C. 3-wire	D.C. 3-wire supply 	None, or through a resistor, or otherwise	2¶ or 3	In each outer	2-pole¶ or 3-pole	In each outer conductor only
A.C. 2-wire	A.C. single-phase 2-wire supply A.C. single-phase 3-wire supply 	None, or at one point only	2*	In each conductor	2-pole	In each conductor
		At more than one point (see Regulation 113)	2	In non-earthed conductor	2-pole	In non-earthed conductor only

A.C. 2-wire (*cont.*)	A.C. two-phase 4-wire supply		None, or at one point only	2*	In each conductor	2-pole	In each conductor
	A.C. three-phase 4-wire supply		At more than one point (see Regulation 113)	2	In non-earthed conductor	2-pole	In non-earthed conductor only
A.C. 3-wire	A.C. three-phase 3-wire supply		At one point or more, but neutral not used	3	In at least two phases	3-pole	In each conductor
	A.C. single-phase 3-wire, or two-phase 3-wire supply		None, or at one or more than one point	2¶ or 3	In each outer, or phase conductor	2-pole¶ or 3-pole	In each outer, or phase conductor only
A.C. 4-wire	A.C. two-phase 4-wire supply		None, or at one or more than one point	4	In each phase conductor	4-pole	In each phase conductor
	A.C. three-phase 4-wire supply		None, or at one or more than one point	3** or 4	In each phase conductor	3-pole** or 4-pole	In each phase conductor only

* Or single-pole overload circuit breaker in one conductor and a single-pole switch and cut-out in the other conductor.

† Wherever separate cut-outs and linked switches are specified above they may be replaced by linked fuse-switches and where a 'splitter unit' is used it is permitted under the provisions of Regulation 106 to omit the cut-outs specified above.

‡ Except in earthed concentric wiring, in which case the cut-out shall be inserted only in the internal (non-earthed) conductor (see Regulation 114).

¶ One pole in each outer or phase conductor, if a 2-pole circuit breaker or 2-pole switch is employed.

** One pole in each phase conductor, if a 3-pole circuit breaker or 3-pole switch is employed.

Abstracted by permission. (*Abbreviated.*)

SUPPLY VOLTAGE.

These regulations provide for wiring based on the following voltages of supply :—
Medium voltage, i.e. above 250 volts and below 650 volts.
Low voltage, i.e. above 30 volts A.C. or 115 volts D.C., but not exceeding 250 volts.
Extra low voltage, i.e. below 30 volts A.C. or 115 volts D.C.
High voltage, exceeding 650 volts.

GENERAL.

These Regulations are intended to be applicable to voltages not exceeding 650 volts.

1. Good workmanship is essential for compliance with Regulations.

2. Where a contract specifies compliance with Regulations, it shall not cover work outside the contract.

3. No addition to an existing installation shall be made without ascertaining that the current-carrying capacity is adequate.

4. The design and construction of all electrical apparatus, including cables, shall comply with the requirements of Section 13.

5. Special forms of construction shall be adopted where there is risk of fire or explosion.

6. A notice shall be fixed near the main distribution fuse-board calling attention to the necessity for periodical inspection and testing of the installation.

SECTION 1.—CONTROL AND DISTRIBUTION OF SUPPLY.

101. *Main Switchgear.*—Every installation obtaining a supply shall be adequately controlled by the switchgear specified on pp. 1428–9.

102. The main switchgear shall be easily accessible to consumer.

103. If supply authority's switchgear is under consumer's control, it need not be duplicated by him.

104. With *ironclad switchgear*, supplying two or more circuits, means must be provided for isolating busbars for supply.

105. *Omission of Main Cut-outs.*—Service cut-outs used for one consumer only may be used to control consumer's wiring, replacing cut-outs in col. 7, p. 1428. This is recommended where the supply authority's fuses are of the quick-acting (H.R.C.) type. The largest fuse in the installation should have a rating not greater than one-third of that of the authority's fuse.

106. Where the distribution fuse-board is combined with the main switch, *i.e.* as a 'splitter unit,' the cut-outs specified in col. 7, p. 1428, may be omitted provided that :—

(*a*) The distribution fuse-board comprises not more than three single-pole or double-pole ways, each of not more than 15 amperes rating and labelled for the rating of the circuit it controls.

(*b*) The total lighting load does not exceed 5 amperes, nor the total load 30 amperes.

(*c*) The fuse-board and main switch are enclosed in a rigid case of metal, or of non-conducting, non-absorbent, incombustible material—the latter only if protected against or not liable to mechanical damage.

(*d*) The switch can be operated from outside the case and its handle does not pass through an unprotected slot.

107. (*a*) *Open-type Switchboards* shall be placed only in dry situations and in well-ventilated rooms—if used near batteries, access of acid fumes must be prevented.

(*b*) Switchboards in damp situations, or where exposed to inflammable dust or gas, must be of enclosed type. (See Regulation 1301 (*m*)).

108. *Distribution Fuse-boards* shall comply with Regulations 1302 and 1303, provided that :—

(*a*) If sunk into a wall with adjacent material not entirely incombustible, the case be incombustible.

(*b*) If exposed to weather or to very moist atmosphere, the case is weatherproof, and is either provided with cable glands or bushings or adapted to receive screwed conduit.

109. *Connection of Distribution Boards.*—Every distribution fuse-board shall be directly connected to one of the following :—

(*a*) The main switchgear controlling the supply.

(*b*) A separate way on a larger distribution fuse-board or switchboard.

(*c*) A circuit feeding several distribution fuse-boards, either by looping into their busbars or in the form of a ring main. Such circuit shall itself be connected either to one way of a switchboard, or, through switchgear, direct to the source of supply.

110. *Protection against Excess Current.*—Except as prohibited under Regulations 112 to 114 and 1204, every circuit and sub-circuit shall be protected on each pole by a cut-out or circuit-breaker.

111. A fuse, non-linked switch or circuit-breaker shall not be inserted in —

(*a*) The middle wire of a D.C. or 1-phase, 3-wire circuit.

(*b*) The common return of a 2-phase, 3-wire circuit.

(*c*) The neutral of a 3-phase, 4-wire circuit.

NOTE.—The terms middle wire, common return and neutral do not apply to a conductor of a 2-wire circuit derived from a 3- or 4-wire circuit.

112. Where an A.C. supply system is, with approval, connected to earth at more than one point, or where a supply is taken from a transformer on consumer's premises and has one pole permanently and solidly connected to earth at more than one point, a cut-out, non-linked switch or circuit-breaker shall not be inserted in an installation in that pole which is earthed.

NOTE.—This includes 2-wire circuits connected to a 3- or a 4-wire system earthed as described.

113. A fuse, non-linked switch or circuit-breaker shall not be inserted in the external conductor of an earthed concentric system. (See Regulation 412.)

114. An *isolating link* may be provided for testing. The link shall have at least as large current-carrying capacity as the conductor it isolates, be securely fixed by bolts or screws, and used only when other conductors are disconnected.

115. In a two-wire installation or circuit all single-pole switches shall be fitted in the same conductor throughout, which shall be that connected to an outer or phase conductor or to the non-earthed conductor of the supply.

116. (*a*) Except as provided in 201 (*c*) and 704 (*c*) every cable and flexible cord shall have a current rating not less than that of the fuse protecting the circuit in which it is connected.

(*b*) When the cross-section of a conductor is reduced, the circuit shall be protected locally by a fuse unless the rating of the fuse protecting the circuit does not exceed the rating of the smaller conductor. This does not apply to cases where, for technical reasons the reduction in cross-section must be made (*e.g.* shunt coils of contactors, etc.).

(*c*) Where a branch conductor is tapped from a main conductor having a current rating greater than 150 amperes and it is not possible to place the local fuse or circuit-breaker at the tapping point, the length of branch to the fuse, etc., shall not exceed 50 ft., the branch shall have a rating not less than one-quarter that of the main conductor, the branch shall be protected against mechanical damage, precautions shall be taken to reduce fire risk if the insulation of the branch be inflammable, the rupturing capacity of the fuses in the main and the branch shall be related to the estimated short-circuit current and the main conductor shall be protected by a circuit-breaker incorporating earth-leakage protection operating when the leakage exceeds 50 per cent. of the rating of the smallest branch.

117. Terminals between which medium voltage exists shall not be installed except in rooms accessible only to authorised persons, or shall be not less than 6 ft. apart, or shall be enclosed in earthed metal. It shall be possible to open fuse-boards, etc., without necessarily exposing terminals, etc., between which there may be medium voltage.

SECTION 2.—ARRANGEMENT OF FINAL SUB-CIRCUITS.

201. (*a*) Every *final sub-circuit* shall be connected to a separate way on a distribution fuse-board. Where there is only one final sub-circuit, it may be directly connected to the main switchgear.

(*b*) A final sub-circuit having a rating not exceeding 15 amperes may supply an unlimited number of points, provided that their aggregate rating does not exceed that of the cable, that there shall be not less than one final sub-circuit for lighting (apart from socket outlets) for each 1,000 sq. ft. of floor area and that the protection of flexible cords complies with 202.

(*c*) A final sub-circuit having a rating exceeding 15 amperes shall not supply more than one point except (i) for cookers complying with B.S. No. 438 ; (ii) a final sub-circuit not less than 0·0045 sq. in. (7/·029) and protected by a fuse or fuses not exceeding 20 amperes may serve two 13 ampere sockets ; (iii) similarly a 0·007 (7/·036) protected by a fuse or fuses not exceeding 30 amperes may supply four 13 ampere sockets ; (iv) a 0·0045 sq. in. (7/·029) ring circuit both ends of which are brought into a fuse not exceeding 30 amperes may serve up to ten 13 ampere sockets (an unlimited number for a small house or flat of less than 1,000 sq. ft. floor area ; (v) spurs may be taken from the ring provided they are not less in section than the ring.

(*d*) Where grouped lampholders (*e.g.* in cornice or panel lighting, electric signs, etc.) are connected to a final sub-circuit without flexible cords, more than 10 lampholders may be connected to a final sub-circuit provided that the maximum working current does not exceed 10 amperes, and that any electric sign is controlled either by a fuse on each pole and a multi-pole linked switch or by a multi-pole circuit-breaker.

202. (*a*) The rating of the fuse or circuit-breaker protecting a circuit shall not exceed the current rating of the cable.

(*b*) Where a fuse is fitted in a plug, the rating of the fuse shall not exceed that of the flexible cord attached to the plug.

(*c*) A final sub-circuit supplying one point only shall have a rating not less than that of the point.

(*d*) Where a diversity factor is applicable (see 303), a final sub-circuit supplying a number of points shall (except for (*f*) below) have a rating not less than that of the largest point or less than two-thirds the aggregate rating of the points, whichever is the larger.

(*e*) Every 15 ampere point shall be taken to require 15 amperes, every 5 ampere point 5 amperes, every 2½ ampere point at least ½ ampere and every lampholder at least 60 watts.

(*f*) In buildings other than dwellings the cables may be smaller than those in (*d*) above provided that the installation is specified by a competent electrical engineer and is in the continuous charge of a qualified person.

(*g*) If a fuse or circuit-breaker rating be replaced by one of larger rating, every flexible cord attached to the sub-circuit shall be increased accordingly.

(h) Fitting wire shall be used only for the internal wiring of fittings.

(i) Except for 203 (a), the size of every lampholder shall depend on the rating of the fuse protecting it.

FINAL SUB-CIRCUITS.

(See Regulation 202.)

NOTE.—Where a final sub-circuit is protected by a circuit-breaker or fuse-link adjusted or rated in accordance with col. 1 or col. 2 respectively, then the smallest cables and flexible cords that may be used in any part of such sub-circuit without further protection (see Regulations 202 and 203) are shown in cols. 4 and 5, and appropriate sizes of lampholders and socket-outlets in cols. 6 and 7 respectively.

Circuit-Breaker or Fuse-Link.*			Minimum Size of Cable other than Flexible Cord.‖	Minimum Size of Flexible Cord.‖	Appropriate Sizes of Lampholders.**	Appropriate Sizes of Socket-Outlets (B.S.I. Rating).
Adjustment of Circuit-Breaker.†	Current Rating of Fuse-Link.‡	Size of Fuse Wire (Tinned Copper).§	Number and Diameter (in.) of Wires.	Number and Diameter (in.) of Wires.		
1.	2.	3.	4.	5.	6.	7.
Amps.	Amps.	S.W.G.				Amps.
Up to 4	3	38	1/·044	14/·0076	B.15, B.22, E.14††, E.27, and E.40	2 and 5
4·1 „ 6	—	—	1/·044	23/·0076	B.22, E.27, and E.40	2 and 5
6·1 „ 10	5	35	1/·044	40/·0076	B.22, E.27, and E.40	2 and 5
10·1 „ 20	10	29	3/·036	70/·0076	B.22, E.27, and E.40	5
20·1 „ 30	15	25	7/·029	110/·0076	E.27 and E.40	5 and 15

203. (a) Where a pilot lamp is used in connection with and with its wiring is built into a current-using appliance, the fuse or circuit-breaker protecting the appliance shall be taken to be that protecting the pilot lamp.

(b) When the pilot lamp is not built in it shall be supplied through local fuses.

NOTE.—A pilot lamp circuit used in connection with a current-using appliance and connected through local cut-outs is not deemed to be a final sub-circuit.

204. *Lift Circuits.*—(a) Circuits supplying current to lift or hoist motors shall not be included in any twin or multicore trailing cable used in connection with the control and safety devices; and they shall not be connected to a lighting distribution fuse-board unless the maximum current (including starting) of the motor is less than 20 per cent. of the total rating of the fuse-ways, and unless the fuse-way of the motor is clearly labelled.

(b) A twin or multicore trailing cable for a lift or hoist and incorporating any conductor at supply voltage shall not include any conductor of a signalling circuit operated at reduced voltage from the supply system or energised from another source.

*** Attention is drawn to the exception permitted in Regulation 704 for circuit-breakers and fuse-links in motor circuits.**

† The requirements for the current rating and adjustment of circuit-breakers are contained in Regulation 611.

‡ See Regulation 612 (c) and Table on p. 1413 for the current rating of fuse-elements.

§ Standard alloy fuse-links which comply with B.S. No. 88—1939 may be used up to a working current of 5 amperes (see Table on p. 1413).

‖ Conductors of greater cross-sectional area than those indicated in cols. 4 and 5 may be necessary in order to comply with the requirements for voltage drop (see Regulation 304).

**** B.15 = small bayonet lampholder; B.22 = ordinary-size bayonet lampholder; E.14 = small Edison-type screw lampholder; E.27 = medium Edison-type screw lampholder; E.40 = Goliath Edison-type screw lampholder.**

†† E.14 Edison-type screw lampholders should only be used on circuits not exceeding 130 volts and for apparatus not exceeding 40 watts.

205. *Control of Lighting Fittings.*—Every lighting circuit shall be controlled by a switch or switches, or—subject to Regulation 208—by a readily accessible socket and plug.

206. Every *switch lampholder* shall be provided with further means of control in a readily accessible position in the same room.

207. *Single-pole Switches.*—(*a*) In a 2-wire installation all non-linked, single-pole switches shall be fitted in the same conductor throughout, which shall be the conductor connected to an outer or phase conductor, or the non-earthed conductor of the supply.
(*b*) In a 3- or 4-wire installation every non-linked single-pole switch shall be fitted in a conductor connected to one of the outer or phase conductors of the supply.

208. Where the supply is D.C., each socket-outlet shall be controlled by a switch immediately adjacent thereto or combined therewith.

209. *Control of A.C. Socket-outlets.*—With A.C. supply, a socket need not be controlled by a switch in the final sub-circuit to which it is connected.

NOTE.—Where a socket may be misused by children, it is desirable to instal a type in which the contact tubes cannot remain alive after, or, alternatively, are automatically screened by, the withdrawal of a plug.

210. A sub-circuit supplying an electric sign shall be controlled by a multi-pole linked switch or a multi-pole circuit-breaker.

SECTION 3.—CONDUCTORS AND CABLES (GENERAL REQUIREMENTS).

301. *Cables.*—All conductors of cables, other than the external conductor of earthed concentric wiring, shall comply with Regulations 1304 and 1306 as regards manufacture, and shall be of the standard sizes and constitution set out in Tables in the Regulations for ordinary cables, flexible cables and flexible cords. (See p. 1381 *et seq.*)

302 (*a*). *Carrying-capacity of Cables.*—Every conductor of a cable shall be capable of carrying without the respective ratings in Tables given in the Regulations being exceeded, the maximum current which can flow in it under normal conditions of service,
(*b*) *Bare Conductors.*—Relates to the current-carrying capacity of bare conductors.
(*c*) *Switchboard Connections* shall comply with B.S.S. No. 159.
(*d*) *Motor Circuits.*—For a motor, the maximum current shall be deemed to be the rated full-load current.

303. *Diversity Factor.*—A diversity factor may be applied, provided that in the case of a final sub-circuit Regulations 201 and 202 are not contravened.

304. *Voltage Drop.*—For lighting, the size of conductors shall be such that the drop of voltage from the consumer's terminals to any point does not exceed 1 volt plus 2 per cent. of the declared voltage at the consumer's terminals when conductors are carrying the maximum demand under normal service, except :—
(i) Where the voltage of the consumer's installation is automatically controlled, the voltage drop at the terminals of any lamp or appliance may be such that the voltage is not less than 95 per cent. of the declared voltage.
(ii) In motor circuits the voltage at the terminals of the motor shall not in normal service be less than 92·5 per cent. of the declared voltage.

305. Where a D.C. supply is given to earthed concentric wiring, the p.d. between any two positions in the external conductors shall not exceed : (i) seven volts if the internal conductors are connected to the positive pole of the system ; (ii) 1½ volts if connected to the negative pole of the system.

NOTE.—This is to minimise risk of electrolytic action.

306. *Minimum Size.*—The smallest cross-sectional area of conductor for sub-circuit wiring shall not be less than 0·0015 sq. in., except for fittings wire.

307. *Covering of Cables.*—Flexible cables and cords with a protective braiding of natural or artificial silk or of glacé cotton shall not be used where subject to risk of mechanical damage. Twisted flexible cords may be used only for fixed wiring and fixed lighting fittings (see Regulation 604).

308. *Cables unsuitable for A.C.*—The following types of cables shall not be used for A.C. except as earthed concentric wiring in which the sheath forms one conductor :—
(*a*) Single-core cables armoured with wire or tape of magnetic material, or encased in a sheath of magnetic material.
(*b*). Single-core cables encased in brass, copper or equally hard and incorrodible metal with a conductor cross section greater than 0·1 sq. in.

309. *Cable colours.* (*a*) The middle wire, common return or neutral shall be black. The covering of the earth continuity conductor in a flexible cord shall be green.

(*b*) Where more than one phase or non-earthed conductor is used, the colours set out in 310 and 311 shall be used.

(*c*) Where busbars are distinguished by colour, either the colours shall conform to 310 and 311 or, if they conform to B.S. No. 158 the terminals shall be coloured in accordance with 310 and 311.

NOTE.—Brown may not be used for any purpose, and any green used should easily be distinguished from black or blue.

310. *D.C. Cable Colours.*—For D.C. supply systems, the following colours for cables shall be used :—

(*a*) Two-conductor circuits connected to a 2-wire system—red for positive or switch wire, black for negative.

(*b*) Two-conductor circuits connected to the middle wire and one outer conductor of a 3-wire system—red for outer, black for middle wire.

(*c*) Two- or three-conductor circuits connected to a 3-wire system, except as in (*b*)—red for positive or switch wire, black* for middle wire, white for negative or switch wire.

311. *A.C. Cable Colours.*—For A.C. supply systems, the following colours for cables shall be used :—

(*a*) Two-conductor circuits of a 2-wire system connected to one phase—red for switch wire or one conductor, black for neutral or other conductor.

(*b*) Two- or three-conductor circuits of a 3-wire system connected to one phase, except as in (*a*)—red for one conductor or switch wire, black* for middle wire, white for other conductor or switch wire.

(*c*) Three-conductor circuits connected to a 2-phase, 3-wire system—red for one phase, black* for common return, white for other phase.

(*d*) Three-conductor circuits connected to a 3-phase, 3-wire system—each conductor red, white and blue respectively.

(*e*) Four-conductor circuits connected to a 2-phase, 4-wire system—red for one phase, white for other.

(*f*) Four-conductor circuits connected to a 3-phase 4-wire system—red, white and blue for the three phases and black for neutral.

NOTE.—The covering of an earth continuity wire in a flexible cord or cable shall be green.

312. *Cable Ends, Sockets, Terminals.*—(*a*) The ends of all conductors above 0·01 sq. in. (7/·044 in.) in cross section shall have soldering sockets capable of taking all the strands.

(*b*) Where cable sockets are not used, the exposed ends of stranded V.I.R. cables in damp situations and of all impregnated-paper-insulated cables shall be made solid by soldering.

(*c*) There shall be no appreciable mechanical stress on any socket or terminal.

(*d*) The insulation shall not be removed farther than necessary to permit conductor to fill socket or terminal and soldering. Insulation damaged by heat to be cut away and replaced by suitable insulation at least as thick as original.

(*e*) Braid, lead, or other covering over the insulation, also the tape in contact, shall be cut back at least half an inch from the insulation.

(*f*) In impregnated-paper-insulated cables, the exposed conductor and insulation shall be protected by sealing.

(*g*) Soldering fluxes containing acid or other corrosive substances shall not be used.

313. *Joints and Connections.*—(*a*) Every connection between cables shall be made by soldering or a mechanical connector. It shall be readily accessible and mechanically and electrically sound. (For flexible cords, see Regulation 605.)

(*b*) Soldered joints in V.I.R. cable shall be lapped with rubber to at least thickness of cable insulation, and made moisture-proof with waterproof protecting tape. Joints in cables protected by tough rubber to be enclosed in boxes complying with B.S.S. No. 816—1938, the cable covering to extend inside boxes.

(*c*) Soldered joints in impregnated-paper-insulated cables to be insulated with impregnated tape and enclosed either in wiped lead sleeves or cable sheathings or in joint boxes complying with B.S.S. No. 816—1938, the sleeves or boxes being filled with insulating compound impervious to moisture.

(*d*) Every mechanical connector used for joints to be enclosed in a box complying with B.S.S. No. 816—1338, the cable covering extending inside box. With impregnated-paper-insulated cables, the box to be filled with insulating compound impervious to moisture, or the cable ends protected by compounded tape. (For flexible cords, see Regulation 605.)

(*e*) Soldering fluxes not to contain acid or other corrosive substances.

314. Joints in the external conductor of earthed concentric wiring must not increase the resistance of the conductor.

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* Except in a three-core cable, when it shall be a bright blue.

315.—(a) Cables must not be exposed to drip or accumulation of water or oil or to high temperature unless adequately shielded, or specially designed to withstand same.

NOTE.—Cables for use where acids or alkalis are likely to be present, should have an outer protective covering of thick rubber.

Installing conductors in channels, etc., containing hot pipes is strongly deprecated—where unavoidable, they should be protected from reaching excessive temperatures.

In long vertical channels (as in lift wells), barriers may be desirable at interva's to prevent air at the top attaining an excessive temperature.

When channels, etc., are being formed as runs, incombustible barriers should be provided to prevent spread of fire.

(b) The maximum ambient temperature shall not exceed 115° F. for V.I.R. cables and 150° F. for impregnated paper or jute or varnished cambric.

(c) Electricity supply cables shall not be run in same conduit, groove, etc., as cables of radio, telephone or bell circuits, except where latter conform to the requirements in these Regulations relating to lighting, heating and power circuits. Where pushes for such services are mounted in or on parts of the supply system, they must be protected by rigidly fixed screens.

(d) All cables other than trailing cables in a lift shaft shall (except in special instances such as chemical works) be armoured or enclosed in steel or other hard-metal conduits. If conduits are used, the control and motor leads shall be in separate conduits.

(e) Bends in V.I.R. cable must have inside radius not less than 4 times the cable diameter if unarmoured, nor less than 6 times if (i) lead-sheathed and/or armoured, or (ii) hard-metal-sheathed.

(f) Bends in any impregnated-paper-insulated cable must have inside radius not less than 8 times cable diameter.

(g) With P.V.C. cables precautions must be taken against deformation of the insulation with temperature, and P.V.C. cables shall not be taken into fittings where the temperatures exceeds 135° F.

(h) Where flexible cords are used and the temperature may exceed 135° F., the cords shall be of one of the types specified in 1309.

SECTION 4.—INSTALLING OF CONDUCTORS AND CABLES.

401. *Bare Conductors.*—Relates to bare, taped or painted conductors other than earthing connections and the external conductor of earthed concentric wiring.

402. *Exposed Wiring.*—Braided V.I.R. cables or P.V.C. cables braided or unbraided complying with Regulations 1307 (a) and (b) and 1308 (a) and (b) may be used without further protection of casing, duct or conduit provided that:—

(a) They are open to view throughout their length (except as in clauses (e), (f) and (g) and in particular they are not installed under floors or within partitions (except as in (g)) or buried in plaster.

(b) They are prevented from coming into contact with any other conductor, earthed metal, gas or water pipes.

(c) They are supported on insulators spaced to prevent cables from coming into contact with one another, or with walls, ceilings, etc., or with any fixture. Insulators must have edges that will not damage the braiding.

(d) In damp situations, the supports and fixings of insulators are of non-rusting material.

(e) Where liable to mechanical damage or less than 6 ft. above the floor they are adequately protected.

(f) If passing through floors, walls, etc., they are enclosed in metal or other non-absorbent incombustible conduits with bushed ends. The holes for conduits in walls and floors to be made good with cement, etc., to full thickness. Metal conduits to be earthed.

(g) Where not open to view (in walls, roof spaces, etc.) they shall be protected by casings (Regulation 409), conduits (Regulation 407) or ducts (Regulation 406).

403. *Metal-sheathed and/or armoured cables* complying with Regulation 1306 may be used without the further protection of casing or conduit, provided that:—

(a) They are prevented from coming into contact with gas or water pipes.

(b) If liable to mechanical damage, they are suitably protected.

(c) They are secured by clips, saddles or clamps (excluding driven staples) or (subject to (b)) embedded in plaster. The clips, etc., must not set up electrolytic action with sheathing, etc., nor damage the latter.

NOTE.—Serious corrosion of lead sheathing is likely to result from contact, in presence of moisture, with lime, cement, oak and other woods.

(d) The maximum spacing of clips, etc., for metal-sheathed cables (other than hard-metal-sheathed—see Regulation 1309—and armoured cables) installed where likely to be disturbed comply with the following table:—

Nominal Cross-Sectional Area of Conductor.	Standard Number and Diameter (in.) of Wires forming Conductor.	Maximum Spacing of Clips, etc.	
		Horizontal Runs.	Vertical Runs
Sq. In.		Ins.	Ins.
0·0015	1/·044	9	15
0·002	3/·029	9	15
0·003	3/·036	9	15
0·0045	7/·029	9	15
0·007	7/·036	12	15
0·01	7/·044	12	15
0·0145	7/·052	15	18
0·0225	7/·064	15	18
0·03	19/·044	15	21
0·04	19/·052	18	21
0·06	19/·064	18	21

Note.—The above spacings are also recommended for armoured cables.

(e) Where metal-sheathed and/or armoured cables are not likely to be disturbed (as under floors, etc.) greater distances than in (*d*), but not exceeding 10 ft., are permissible. Where vertical, cables shall be firmly gripped at supports; also special supports shall be provided where excessive pressure is likely.

(*f*) Where exposed to dampness or weather, the supports with fixings referred to in (*c*) must be of non-rusting material.

(*g*) Where cables pass through floors, etc., holes must be made good with cement, etc., to full thickness. Where cables—other than armoured—pass through structural steelwork, holes must be bushed to prevent abrasion

(*h*) All connections under floors, etc., are made in metal boxes complying with Regulation 1322.

(*i*) All metal sheathing, armouring, joint boxes, etc., are earthed in accordance with Regulations 1001 to 1008, and made electrically continuous throughout by soldered joints, or bonding clamps, or by special earthing conductors. The resistance of metal sheathing and/or armouring and of earthing lead shall not exceed 1 ohm between earth electrode and any part of installation.

(*j*) The sheathing or armouring is brought into lighting fittings, to the finished surface of a wall or ceiling, or into a recess lined with incombustible material.

404. *Tough-rubber Protected Cables.*—V.I.R. cables protected with tough rubber, or similar types of P.V.C. cable, and complying with Regulation 1306 may be used without further protection of casing or conduit provided that:—

(*a*) If tough-rubber protected, cables are suitably protected where exposed permanently to direct sunlight. This does not include sunlight which has passed through ordinary window-glass.

(*b*) They are prevented from coming into contact with gas or water pipes or earthed metal.

(*c*) If liable to mechanical damage they are suitably protected.

(*d*) They are run on insulators or other supports (excluding driven staples) or (subject to (*c*)) embedded in plaster. The supports and fixings shall not damage the cable.

(*e*) The maximum spacing of the supports where the cables are likely to be disturbed do not exceed those specified in Regulation 403 (*d*).

(*f*) Where the cables are not likely to be disturbed (*e.g.* under floors) greater distances than in (*e*), but not exceeding 10 ft., are permissible. Where vertical, cables shall be firmly gripped at supports; also special supports shall be used where excessive pressure is likely. (See Regulation 315 (*e*) for bends.)

(*g*) Where exposed to weather or dampness, supports and fixings are of non-rusting material.

(*h*) Where cables pass through floors, etc., holes are made good with cement, etc., to full thickness, and where they pass through structural steelwork holes are bushed to prevent abrasion.

(*i*) All connections under floors or within partitions are made in boxes complying with B.S. 816.

405. *Conduits.*—Any type of cable (other than the high-voltage cables in Regulation 808 for luminous-discharge tubes) complying with Regulation 1306 may be enclosed in conduits provided that:—

(*a*) The conduits of each circuit are erected complete before the cables are drawn in.

Note.—Inspection and draw-boxes should remain accessible.

(*b*) The conduits are adequately supported, and are prevented from coming into contact with gas or water pipes or other electric circuits except where the conduits are associated with gas compressors, pumps, etc., or where the conduits are intentionally bonded to water pipes or in the case of communications circuits where the other circuits comply with these Regulations.

(c) The conduits are protected against mechanical damage.

(d) The number of cables run in one conduit does not exceed the maximum set out in the Table on p. 1419. Draw-boxes shall be provided to make it unnecessary to pull cable round more than 2-90° bends.

(e) Conduit bends must permit compliance with Regulations 315 (e) and (f). In no case must inner radius of bend be less than 2½ times conduit outside diameter. Elbows or tees (unless of inspection type) shall not be used, except at ends of conduits close to fittings.

(f) Where a conduit terminates at or passes through a switch position, the outlet from the conduit is in the form of a box.

(g) All conduit ends are bushed or finished to prevent abrasion of cables, and at fittings are screwed or clamped Factory-made open sockets may be used as outlets, but openings shall not be made in conduits or their fittings to form outlets.

(h) Where conduit passes through a floor, etc., the hole is made good with cement, etc., to full thickness of floor, etc.

(i) All connections under floors, etc., are made in metal boxes, except where non-metallic conduits are used, the boxes may be of non-absorbent, incombustible material.

(j) Where exposed to weather or dampness, supports and fixings shall be of non-rusting material or finish. The conduits, if of metal, must be heavy gauge, welded or solid drawn, and screw-jointed.

NOTE.—Damp situations do not include burial in plaster.

(k) Metal conduits are earthed according to Regulations 1001 to 1009, and are mechanically and electrically continuous at all joints, so that the resistance from earth electrode to any part of installation shall not exceed 1 ohm.

NOTE.—Plain grip sockets do not comply with (j).

(l) Inspection and draw-boxes for metal conduits are in rigid electrical and mechanical connection with same. With heavy-gauge steel conduits this device to be obtained by screwing or clamping both sides.

(m) V.I.R. cables not to be used unless suitably protected where permanently exposed to contact with rust.

(n) D.C. cables may be bunched whatever their polarity. A.C. cables shall be bunched so that outgoing and return cables are in same conduit.

(o) D.C. and A.C. cables may be run in same conduit, except otherwise specified.

(p) The interior of conduits into which taped and compounded unbraided V.I.R. cables are drawn shall be tinned.

406. *Duct Systems.*—Sheet-metal or non-metallic duct systems shall comply with Regulation 405 as far as applicable.

407. Where passing through floors, etc., ducts, etc., shall be closed by barriers to prevent spread of fire.

408. Deals with mineral-insulated copper-sheathed cables.

409. *Wood Casing.*—Any cable complying with Regulation 1307 may be enclosed in wood casing provided that:—

(a) Casing is used in dry situations only, is not buried in plaster or cement, is not fixed in contact with gas or water pipes or immediately below latter, and is not exposed to drip.

(b) The capping is secured by screws.

(c) If casing is part of ornamental woodwork, ready access to cables is provided.

(d) Number of cables bunched in one groove does not exceed that shown in table.

Nominal Cross-Sectional Area and Size of Cables.	Maximum Number of Cables.
Not exceeding 0·007 sq. in. (7/·036 in.)	10
Exceeding 0·007 " " (7/·036 ") / Not exceeding 0·0225 " " (7/·064 ")	6
Exceeding 0·0225 " " (7/·064 ") / Not exceeding 0·1 " " (19/·083 ")	4
Exceeding 0·1 " " (19/·083 ")	3

(e) Size of casing does not exceed that necessary for number of cables permissible.

(f) D.C. and A.C. cables may be run in same groove, unless otherwise specified.

410. *Earthed Concentric Wiring.*—(*a*) This shall only be used where (i) it is connected to secondary side of a transformer or convertor, and insulated from supply system; (ii) it has been approved by Electricity Commissioners for connection to a particular supply system, or (iii) supply is obtained from a private plant.

(*b*) A cut-out, non-linked switch or circuit breaker shall not be inserted in external conductor.

(*c*) The external conductor shall be earthed. With D.C., the external conductor shall, where possible, be negative to the internal conductor.

(*d*) Concentric wiring shall be employed from earthed point to all points for fittings. Wherever the external ceases to surround the internal conductor, the latter shall be separated from the surface on which the fitting is mounted by an incorrodible metal plate or terminal box to which the external conductor is connected.

411. *Medium-voltage Circuits.*—These shall be completely enclosed in strong metal casing, heavy-gauge screwed conduit or armouring; or so installed as to prevent danger.

412. *Bell and Signalling Circuits.*—(*a*) If connected directly to a supply system (*i.e.* not through a transformer) the wiring of these circuits shall conform with the Regulations for lighting, heating and power circuits.

(*b*) Where connected to a supply through a transformer or converter, the primary wiring shall conform with Regulations for lighting, heating and power circuits.

(*c*) The voltage of a bell or signalling circuit that is not metallically connected to a supply system shall not exceed 15 volts, unless the whole of wiring is designed for the operating voltage.

NOTE.—See Regulations 709, 710 and 1001 (*c*).

413. *Flexible-cord Wiring.*—Flexible cords should not be used for fixed wiring, but may be used as temporary extensions (see Regulation 604).

SECTION 5 DEALS WITH TEMPORARY INSTALLATIONS.

SECTION 6.—INSTALLING OF ACCESSORIES AND LIGHTING FITTINGS.

601. All accessories must be capable of carrying, without their rating being exceeded, the maximum current to which they are likely to be subjected.

602. (*a*) *Ceiling roses* must not be used on circuits for more than 250 volts.

(*b*) Not more than 2 flexible cords, each of which shall not have more than 3 conductors, shall be attached to one ceiling rose unless the latter be designed for multiple pendants.

603. (*a*) *Lighting fittings* shall comply with Regulation 614 (damp situations), Regulation 615 (fire) and Regulation 205 (control).

(*b*) *Celluloid* must not be used for shades or candle tubes, nor near a lamp.

(*c*) *Portable Lighting Fittings.*—All exposed metal parts of a portable lighting fitting which have to be earthed in compliance with Regulation 1001 (*b*) shall be connected to the earth electrode through an earthing conductor in the flexible cord. If such cord has a metal armouring, the latter also must be connected to the metal of the fitting and the earthed metal of the plug.

NOTE.—A metal lampholder in a portable fitting may, as an alternative to being earthed, be insulated and shielded from user. (See Regulation 1001 (*a*).)

(*d*) Where passing through ceilings, flexible cords shall be enclosed and end in incombustible tubes and boxes.

(*e*) *Weight Supported by Cords.*—The maximum weight a twin flexible cord shall carry shall be:—

Conductor.	Weight in Lb.
14/·0076	3
23/·0076	5
40/·0076	10

(*f*) The rubber insulation of the cores is not pressed against other cores or against a wall or ceiling, etc.

604. *Temporary extensions* in flexible cord wiring in shops and showrooms shall:—

(*a*) Comply with B.S.S. No. 7—1939.

(*b*) Be used only for final sub-circuits and below 6 amps. and 250 volts.

(*c*) Be open to view or protected in accordance with 603 (*d*).

(*d*) Be prevented from coming into contact with other conductors, gas, water, etc., pipes, or other wiring, and not be below water pipes or exposed to drip.

(*e*) The rubber insulation of a core shall not be compressed against other cores or against walls or ceilings, etc.

605. (*a*) *Connections between cables and flexible cords* or between cords shall be made by mechanical connectors shrouded in incombustible insulating material.

(*b*) Connectors for attaching flexible cords to portable appliances shall comply with Regulation 1328.

(*c*) If connection be made by contact-tubes and pins, separation of pins from the contact-tubes shall disconnect pins from the supply.

606. (a) A lampholder shall not be used on currents over 250 volts except for fused pilot lamps on switchboards.

(b) Every switch lampholder shall be provided with further means of control in same room. (See Regulation 206.)

(c) Where centre-contact bayonet or Edison-type screw-lampholders are used, the outer or screwed contact shall be connected to the middle wire or neutral or earthed conductor of the circuit.

607. *Lampholder Plugs.*—(a) A lampholder plug shall only be inserted in a lampholder controlled by a switch fixed conveniently near or by a readily accessible cord.

(b) A lampholder plug shall not be used where appliance takes over 2 amperes or where exposed metal has to be earthed in compliance with Regulation 1001.

608. *Socket-outlets and Plugs.*—(a) These shall comply with the appropriate B.S. Specification.

(b) Where sunk, means shall be provided to ensure that floor may be washed without damaging insulation. Line metal must not come into contact with floor covering.

(c) Where sunk, a receptacle of incombustible material, oak (English), teak or mahogany shall be provided to take up slack cable behind socket.

(d) A D.C. socket shall be controlled, where required by Regulation 208, by a switch in final sub-circuit.

(e) In earthed concentric wiring, where earthing is necessary under Regulation 1001, the flexible cords shall terminate in non-reversible plug and socket connections.

(f) Where a socket is controlled by a 1-pole switch in final sub-circuit, the switch shall control an outer or phase conductor, or the non-earthed conductor of the circuit.

(g) Terminals E, L and N of a socket for use with a 3-pin plug, shall be connected thus:—

(i) Terminal E (earth) to earthing wire.

(ii) Terminal L (line) to an outer, or phase, or the non-earthed conductor.

(iii) Terminal N (neutral) to the middle wire, common return or neutral.

(h) Terminals L and N of a socket for use with a non-reversible 2-pin plug shall be connected thus:—

(i) Terminal L (line) to an outer, or phase, or non-earthed conductor.

(ii) Terminal N (neutral) to middle wire, common return or neutral.

(i) A plug containing a cut-out shall comply with Regulation 1329, shall be non-reversible and arranged so that cut-out controls an outer, phase, or non-earthed conductor.

(j) The connections to the terminals of a 3-pin plug shall correspond with (g). A 1-pole switch (if any) on appliance shall control an outer, or phase, or non-earthed conductor.

609. (a) Every *socket-outlet adaptor* shall comply with Regulation 1330.

(b) It shall not be sunk below the surface of the wall to which the main socket is fixed.

(c) Where a socket adaptor is connected to a socket to supply one or more appliances of small rating, it shall contain a protecting cut-out complying with Regulation 1331.

(d) A socket adaptor containing a cut-out shall be non-reversible, and the cut-out shall control an outer, phase or non-earthed conductor.

610. *Switches.*—Every switch shall comply with Regulation 1332 and:—

(a) If in a room with a fixed bath it shall comply with Regulation 1002.

(b) If on a wall and surrounded by incombustible material, the case or cover may be of oak (English), teak or mahogany; but if conductors are in metal conduit, the switch shall be in a metal box in electrical continuity with conduit.

(c) It shall not render ineffective Regulations 1332 (c) and (e).

(d) Where subject to mechanical injury, the cover, unless of rigid metal, shall be protected by a guard.

NOTE.—Special switches may be necessary for inductive apparatus.

611. *Circuit breakers.*—A circuit breaker shall operate when a current flows equal to or less than twice the rating of the smallest conductor it protects, provided that this does not apply to a motor circuit installed in accordance with Regulation 704 (c).

612. *Fuses*—(a) Every fuse shall be fitted in accordance with one of the three methods below, and never in a ceiling rose or socket-outlet:—

(i) On a switchboard, unless it protects a pilot lamp or instrument, it shall be on the front.

(ii) In a socket-outlet adaptor or in a plug complying with 1312.

(iii) In a readily accessible place and completely shielded.

(b) A fuse shall not be fitted with an element larger than that for which it is designed.

(c) The rating of the fuse shall not exceed that of the smallest cable in the circuit which it protects, provided that:—

(i) A fuse smaller than 3 amperes need not be inserted in a final sub-circuit (but see 902).

(ii) Where the fuse protects a motor when the starting or accelerating circuit greatly exceeds the full load current, the fuse may be of size given in 704 (c).

(iii) Fuses installed under exemptions permitted in 201 (c) may be of the rating indicated therein.

NOTE.—It is recommended that the supply authority be consulted as to the grade of main fuse installed.

(d) Every fuse shall be marked with its appropriate current rating.

The rating of a fuse shall not exceed that of the smallest cable it protects, subject to :—

(*a*) No fuse rated below 3 amperes need be inserted in any final sub-circuit (subject to Regulation 902).

(*b*) Rating of a flexible cable or cord shall, for this Regulation, be deemed that of a V.I.R. cable of like cross section.

(*c*) For protecting motors with large starting or accelerating currents, the fuses may be of size permitted under Regulation 704 (*c*).

613. *Voltages above* 250 *Volts.*—Where the voltage between any two points in a room exceeds 250 volts :—

(*a*) Appliances with such high voltages shall be fixed not less than 6 ft. apart. Where this is impracticable for switches, they shall be completely enclosed in earthed metal boxes properly labelled.

(*b*) Flexible cables and cords for portable appliances to be protected against damage. Any metal covering to be connected at each end to earthing wire in cable or cord in addition to other earthing devices.

614. *Damp Situation.*—(*a*) In damp situations, weatherproof fittings to be used, and switches to be provided with cable glands or bushings or adapted for screwed conduit.

(*b*) Fittings shall be either in electrical continuity with the sheathing of an earthed system, or be mounted on a block of ash, beech, birch, box, cocos, mahogany, oak (English), teak, or walnut, or upon some non-absorbent insulating material, in addition to its own base.

(*c*) In a room containing a fixed bath, all flexible cords shall be of the tough rubber protected or equally waterproof type.

615. (*a*) Where inflammable dust or gas is likely to be normally present, flameproof-type fittings shall be used.

(*b*) Lamps near to or which might swing into contact with inflammable material shall be suitably guarded.

(*c*) In garages, etc., every fitting other than portable shall—unless flameproof—be fixed at not less than 4 ft. 6 ins. above floor level.

NOTE.—Portable appliances should not be used where inflammable dust or gas is likely to be present.

SECTION 7.—INSTALLING APPLIANCES (INCLUDING MACHINES).

701. *Control of Appliances.*—(*a*) Every portable heating and cooking appliance shall be fed from an accessible socket.

NOTE.—With appliances like irons a red pilot lamp may be desirable if an automatic temperature-limiting device is not fitted.

(*b*) Every fixed heating appliance not fed from a socket shall be controlled by a switch in the final sub-circuit. These shall be fitted either on the appliance or in the room. When appliance is thermostatically controlled, the switch may be at the fuse-board or elsewhere.

(*c*) The load of a thermostatically-controlled heating appliance shall not exceed 5 kilowatts.

(*d*) Where switches are on an appliance, one luminous section of the heating element or a lamp on the appliance shall not be switch-controlled, and it shall serve as an indication whether the flexible cord and socket are live.

(*e*) A cooker not fed from a socket shall be controlled by a switch or push-button away from the appliance but within easy reach. For less than 30 amperes, this control switch shall comply with B.S.S. No. 438. Heat-control switches may, in addition, be on or near the appliance.

702. *Motors, Generators, etc.*—(*a*) Where the cooling air is above the limit permitted by the appropriate B.S. Specification, machines shall be specially constructed, or of the pipe-ventilated, forced or induced-draught type, or connected by ventilating ducts to a cool air supply.

(*b*) Unless enclosed, as provided for in (*d*), every machine shall be placed :—

(i) In a space ventilated to prevent accumulation of inflammable dust or gas.

(ii) In a position not exposed to mechanical injury, or damage from water, oil or steam.

(*c*) No readily combustible material shall be within 12 ins. horizontally or 4 ft. above any non-enclosed-type machine.

(*d*) Where enclosing is necessary to satisfy (*b*) or (*c*), the enclosure shall be of a type approved in B.S. Specifications.

(*e*) Where an open-type machine, mounted on a combustible floor, is liable to a special fire risk, means to prevent oil dripping on to the floor shall be provided.

703. *Motor Controls.*—Every motor shall have starting and stopping switches, conveniently placed for the operator. Motors rated over $\frac{1}{2}$ h.p. shall be provided with :—

(*a*) Means to prevent the motor re-starting after the under-voltage release has operated.

(*b*) A starter or switch for limiting the starting or accelerating current, if this is required by the supply authority.

(*c*) An isolating switch to cut off all supply, including that to the automatic circuit-breaker. (A single isolating switch may be provided for a group of motors where all can be stopped for the inspection of one).

(d) Where the primary isolation is remote, local isolation shall also be provided or arrangements made for locking the primary isolation.

704. *Protection of Motor Circuits*—(a) The maximum normal current in the motor circuit shall be taken to be the rated current of the motor.

(b) The size of the cables for the rotor circuits of slip-ring induction motors shall be suitable of the starting, acceleration and load conditions.

(c) Where starting current greatly exceeds running current, over-current protection must be provided but may include a suitable time-lag device.

705. *Transformers, etc.*—Consumer's transformers, choke coils, resistors (above 60 watts), condensers, rectifiers, etc. (except used as in Sections 8 or 9, or Regulations 709 and 710) shall comply with :—

(a) If not enclosed, it shall not be exposed to water, oil, steam or mechanical damage.

(b) If oil-cooled, placed in a chamber ventilated to outside only. If oil exceeds 50 gallons, escaping oil must not gain access to building.

706. *Separation from Woodwork, etc.*—Combustible material within 24 ins. above, 12 ins. [illegible]low or 6 ins. in any direction from frames containing control resistances and rated at or above 60 Wa[illegible] [illegible] be protected by incombustible material.

707. *Control o[illegible] [illegible]ep-up Transformers.*—A step-up transformer—other than an auto-transformer—must [illegible] [illegible]olled on the consumer's side by a multi-pole (linked) switch for isolating supply.

708. *Auto-transformers.*—[illegible] auto-transform[illegible] [illegible]lying lamps or appliances shall not supply more than 250 volts, nor ear[illegible]ed concentric wiring [illegible] from private plant.

709. *Bells, etc.*—(a) Bells and signalling apparatus connecte[illegible] [illegible]tly to a supply system shall be installed in conformity with these Regulations regarding lighting, h[illegible] [illegible]d power circuits.

(b) Where bells, etc., are connected to supply through transformers, conv[illegible] [illegible]r resistors, the primary side shall conform to these requirements.

(c) The voltage of a bell or similar circuit not metallically connected to a supply system sh[illegible] not exceed 15 volts, unless the apparatus is suitably designed for the operating voltage.

NOTE.—See Regulations 412 and 1001 (c).

710. Bell transformers and similar devices shall conform to Regulation 1342 ; and

(a) Its case (if metal) and core and frame shall be earthed to comply with Section 10.

(b) It shall be mounted on incombustible material and, unless suitably enclosed, not be exposed to water, oil or steam, or risk of mechanical damage.

711. *Electrode Water-Heaters and Boilers.*—Every installation shall comply with :—

(a) It shall not be connected to a D.C. supply.

(b) The shell shall be earthed to the metal sheathing and armouring of the supply system, but shall not be connected to the neutral of the supply.

(c) It shall be protected by a circuit-breaker fitted with earth leakage protection, which shall operate when the earth leakage current exceeds 10 per cent. of the rated current, or 15 per cent. if this is necessary to ensure stable operation, and an inverse-time device may be used to ensure stable operation during momentary leakages.

(d) The circuit-breaker shall be of the multi-pole linked type.

(e) A local as well as a remote circuit-breaker need not be fitted, provided that the remote circuit-breaker can be locked, and that a local push-button and indicator lamps are provided.

(f) Indicator lamps shall be installed wherever the circuit-breaker is not readily visible from the heater or boiler.

(g) At least two over-current releases must be provided for 3-phase apparatus, one is in each live phase wire for 2-phase operation, one in the live phase wire for single-phase operations. Where the consumer's earth resistance exceeds

$$\frac{40}{\text{operating current of circuit-breaker}}$$

the earth-leakage trip shall operate when the potential between earth and the shell of the apparatus exceeds 40 volts.

(h) There must be an ammeter in each phase (or one ammeter with a multiway switch or plug).

712. *Other Appliances.*—Every appliance not otherwise provided for in Regulations 710 to 712 shall be controlled by a readily accessible switch, or, subject to Regulations 208 and 209, by a readily accessible socket and plug.

713. *Fire Risk.*—(a) Where appliances are exposed to inflammable dust or gas, and are not provided for in Regulations 701 to 715 or in (b) below, they shall be of enclosed or flameproof type.

(b) Arc lamps shall not be used where exposed to inflammable dust or gas.

(c) An open inverted arc lamp near combustible material shall be fitted with a metal reflector attached below arc as in Regulation 1336 (c). Where an open arc is essential, the floor beneath

shall be protected from falling particles of carbon. In other positions near combustible material or persons, arc lamp shall be fitted with globe or lantern.

(d) In garages, every appliance other than a portable appliance shall, unless flameproof, be fixed at least 4 ft. 6 ins. above floor level.

SECTION 8.—INSTALLING OF LUMINOUS DISCHARGE TUBES AND ELECTRIC SIGNS.

801. *Luminous-Discharge-Tube Installations.*—In Regulations 803 to 811, the term ' luminous discharge tube ' shall be deemed to apply to any tube, or device, of translucent material, hermetically sealed, for the emission of light from the passage of a current through a gas or vapour contained within it. The term shall not apply to the cathode-glow lamp, or to electric discharge tubes designed to operate below 500 volts. The words ' primary ' and ' secondary ' are used in Regulations 802 to 812 to define respectively the conductors connected to the primary (low-voltage) windings and the secondary (high-voltage) windings of the transformers of luminous-discharge-tube installations.

802. *Fixed Luminous-Discharge-Tube Installations.*—Every fixed luminous-discharge-tube installation shall also comply with (*a*), (*b*), (*e*), (*g*) and (*h*) and Regulations 803 to 811.

(*a*) It shall be so arranged as to prevent possible spreading of fire.

(*b*) It shall be suitably screened; or enclosed and sealed, with a notice ' DANGER. High Voltage ' near the seal.

(*e*) All live parts of an exterior installation shall be protected against the effects of weather.

NOTE:—The installation in a closed market or arcade is deemed to be an exterior installation; but not so in a permanent building used for exhibition purposes.

(*g*) The secondary circuit shall be permanently earthed at the transformer in compliance with Regulations 1005 to 1008, and the core of every transformer shall be earthed.

NOTE.—It is recommended that the whole of the secondary circuit be examined not less frequently than once every three months by a competent engineer.

(*h*) Where the primary circuit is insulated from the supply it shall be permanently earthed at the motor-generator or converter, or at the transformer, in compliance with Regulations 1005 to 1008.

803 (*a*). *Final sub-circuits* which form the primary circuits of luminous-discharge-tube installations shall be reserved solely for such purpose.

(*b*) A separate primary final sub-circuit shall be provided for each transformer of the luminous-discharge-tube installation, except that transformers of less than 250 volt-amperes each may be grouped, provided that the aggregate does not exceed 1,000 volt-amperes.

804. *Switchgear Controlling Final Sub-Circuits.*—(*a*) Each primary final sub-circuit supplying a luminous-discharge-tube installation shall (except as in (*c*)) be controlled by a locked switch operating on all poles, and so controlled that it cannot be put into the ' on ' position until the key has been inserted in the lock, and it shall not be possible to withdraw the key unless the switch is in the ' off ' position. The switch cover shall not be readily opened without the use of the key or a special tool.

(*b*) The ' off ' position shall be clearly marked, and a conspicuous notice shall be placed near the switch, worded as follows: ' Before working on or near electrical discharge-tube installations remove and retain key of locked switch.'

(*c*) Where distribution fuse-boards control exclusively the primary final sub-circuits of luminous-discharge-tube installations, one adjacent switch of the type required in (*a*) may be used to control the busbars of the fuse-boards.

805. *Emergency Switch for Interior Installations.*—An interior installation * shall be provided with suitable adjacent means (which may, if desired, be the locked switch referred to in Regulation 804) for disconnecting all poles of the supply except the neutral in a three-phase four-wire circuit.

NOTE.—It is recommended that the local fire-brigade authority be notified of the position of the switch or circuit breaker controlling an interior installation.

806. ' *Fireman's* ' *Switch.*—For installations on the exterior of a building * a suitable emergency (' fireman's ') switch complying with the following clauses (*a*) to (*f*) shall be provided :—

(*a*) It shall be arranged to open and close the circuit on all poles except the neutral in a three-phase four-wire circuit, and it shall be connected either in the primary supply circuit or, alternatively, where the main supply is direct current, in the direct-current circuit from which the alternating-current supply to the primary circuit is derived.

(*b*) It shall be fixed in a conspicuous position, reasonably accessible to firemen, and, except in the case of an agreement to the contrary with the local fire-brigade authority, at not more than 9 ft. above the ground. It shall be as nearly as possible vertically below the luminous discharge tube or tubes, or alternatively, a notice indicating the position of the switch shall be placed directly below the tube or tubes and a nameplate shall be fixed near the switch so as to render it clearly distinguishable.

* See note to Regulation 802 (*e*) in regard to the term ' exterior installation.'