DESIGN PROBLEMS OF HEATING AND VENTILATION

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By

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AUTHOR'S FOREWORD

THIS book does not pretend to be exhaustive: that would have required several volumes and the repetition of much that is readily available elsewhere. But it seeks to deal with those branches of the subject that have hitherto received little or no attention in English publications. I hope that this fact, together with the many methods of calculation and graphical representation used, will recommend my work as a useful and practical addition to the library.

In spite of the care that has been taken in checking calculations and formulae, errors may have occurred and I should appreciate notification of these for correction in future editions.

A. T. HENLY

'BLYTHEWOOD',
COURT FARM ROAD,
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INTRODUCTION

ALTHOUGH many developments have been made in the application of heating and ventilating equipment in the past twenty years, there have been few alterations in the basic principles upon which heating and ventilating problems are based. There is but little doubt that in spite of this there is room for methods of calculation by means of which time can be saved, and attention has therefore been given to this matter. It is surprising how little of the foreign technical literature is actually employed by the engineer dealing with the many branches of the mechanical equipment of buildings, particularly in view of the matter available in French and German publications.

In numerous instances foreign literature has dealt extensively with points never referred to in English publications, and frequent recourse has been had to these sources of information in the preparation of this work.

Fundamental Problems.

The many fundamental problems which are continually being met with have been dealt with in a large number of text-books, but it is here our intention to avoid the repetition of such matter as far as possible, although references are provided to other works dealing with the points in question. It is rather our aim to present special problems which do not fall essentially within the scope of the student but rather the practising engineer and architect. It is believed that none of the problems dealt with in this work have received attention in other English books, whilst many methods of calculation or applications of known theories are definitely unique.

Graphical Calculation.

The author feels that the graphical representation of formulae and graphical methods of calculation are of great importance, and he has himself proved their practical value. With this in view the author has eliminated wherever possible tables evaluating known formulae and substituted instead graphical methods of representation which he himself employs. One is forced to realize in considering the graphical representation of formulae that in many cases a simple curve is capable of representing many pages of tables, and of the numerous types of curve available for solution of engineering problems the alinement chart or Nomograph is of the greatest value. There is a feeling that the Nomograph or alinement chart is a type of graph developed only in the last few years, but this belief is erroneous, for as far back as 1899 Maurice d'Ocagne produced what is still acknowledged to be the only full treatise† on Nomographs, and at that time he mentioned that the Nomograph was not a new development. It is felt, therefore, that no apologies are required for using almost without exception nomographical treatment throughout this work. It does not fall within our scope to deal with the principles underlying the proportions and construction of a Nomograph for various types of formulae, as there are several works‡ in English referring to this subject.

Although the Nomograph is probably better known as a chart with parallel scales, there are numerous cases where the scales may be inclined or in some instances of curved form. Fortunately, for the majority of formulae employed in heating and ventilating problems the simple type of Nomograph can be used.

We would refer to the usual tables for the sizing of pipes for gravity hot-water heating; these generally occupy several pages and have the disadvantage that they are not visible over the whole of their range without turning pages. Although several authorities prefer to base calculations upon the number of pounds of water flowing through pipes, there are many reasons in support of using instead the number of British Thermal Units (B.T.U.) required to be passed. According to the temperature drop on which the system is to be designed so it would be necessary to have a separate table for this particular temperature drop.

In graphical representation of the formula for the flow of hot water through pipes it would normally be essential either to have a separate Nomograph for each particular temperature drop or to introduce a scale of temperature drop and hence a further reference for every B.T.U. value to be read from the chart. The Nomographical calculator devised by the author combines the functions of the Nomograph and the slide rule in such a way that any scale of values which might be required to be set, as for instance for some particular

† *Traité de Nomographie*, by Maurice d'Ocagne.

‡ See *Line Charts for Engineers*, by W. W. Rose; *The Nomogram*, by Allcock and Jones.

temperature drop in the system, is capable of being adjusted to a fixed position, whilst reference is required to be made to the chart for this particular temperature drop. Thus in Fig. 1 the scale of B.T.U. may be set to any desired temperature drop, remaining in this position for further reference to the chart by the normal method applied to Nomographs. Not only does this type of Nomographic calculator facilitate easy reference to values without a complexity of alinements, but in some cases, as in this particular one, it avoids duplication of charts. The method of use of the author's Nomographic calculators is explained later.

Chapter One

HOT-WATER HEATING

Flow of Water in Pipes.

THE theory of the flow of hot water in pipes has already been amply dealt with elsewhere, so that it will not here be referred to at length. Early investigators into the subject of friction in hot-water pipes were led to believe that the coefficient of friction was not constant over the whole range of velocities met with in practice, and certain limiting velocities were determined to decide which friction coefficient should be employed for the particular velocity range. The use of these different friction coefficients necessarily meant that when plotted in the form of a curve correlating the quantity of water flowing through any particular size of pipe with its frictional resistance, the curve was not of a regular nature. This fact definitely points out the error due to the varying friction coefficient. As a basic formula for determining the friction in hot-water pipes the author has found by practical observation that the formula originally introduced by Meier† fulfils the conditions required in practice.

The formula is as follows:

$$Pf = f\frac{v^{1\cdot86}}{2g} \times \frac{l}{d^{1\cdot25}},$$

where Pf denotes friction loss in feet of water column per foot of pipe,

f = friction coefficient,

v = velocity of flow in feet per second,

g = acceleration due to gravity,

l = length of pipe in feet,

d = diameter of pipe in feet.

A value of the friction coefficient f found to give accurate results in practice is

$$f = 0\cdot0228.$$

This basic expression is not in convenient form for direct calculation of friction, but the nomographic calculator in Fig. 1 is designed to facilitate its practical use. Referring briefly to this chart, the use of which is described facing the chart, it may be seen that it is possible for various commercial pipe sizes to determine friction loss in inches of water column per foot run for various B.T.U. capacities at the temperature drops normally used in practice, whilst a

† *Mechanics of Heating and Ventilating*, by Konrad Meier, U.S.A.

Method of Using Nomographic Calculator for Finding Sizes and Pressure Losses in Pipes for Gravity Hot-Water Circulation. (Fig. 1.)

This special type of nomographic calculator combines the functions of the nomograph and slide rule. It consists of the following scales:

A, an adjustable scale of B.T.U. per hour.

B, a scale of total temperature drop in the system, to which scale A may be set.

C, a scale of pipe diameters.

D, a scale giving pressure loss in pipes in inches of water column per foot run.

E, a scale giving total pressure loss for lengths indicated on scale F.

F, a scale giving total lengths of pipe in feet.

Before this calculator can be employed it is necessary to cut out carefully from the edge of the page the scale A, the cuts being made with a *sharp* knife along the two dotted lines running from top to bottom of the page. *A suitable backing should be placed behind the page before doing this, to avoid damaging other pages.* Further short cuts should be made along the dotted horizontal lines *a—a*, so that scale A may have its ends passed through these two slots *a—a*, enabling it to be adjusted vertically, the arrow on scale A being set to the appropriate value of temperature drop on the fixed scale B.

EXAMPLE 1. The average permissible pressure drop in a pipe carrying 150,000 B.T.U. per hour is 0·001 in. of water column per foot run, the system being designed for a 40° F. temperature drop. What size pipe is required?

Set the arrow on the adjustable scale A to 40° F. on scale B. A straight line through 150,000 B.T.U. on scale A and 0·001 on scale D indicates 4-in. pipe as the nearest diameter on scale C.

EXAMPLE 2. If as example 1, the length of pipe including allowances for bends and similar resistances was 86 ft., what would the total pressure loss in this length amount to?

A straight line through 150,000 B.T.U. on scale A and 4 in. diameter on scale C intersects scale D, and a further straight line from this intersection through 86 ft. on scale F gives on scale E a total pressure loss of 0·066 in.

EXAMPLE 3. A 2-in. pipe carries 100,000 B.T.U. per hour, the system being designed for a 30° F. drop. What is the pressure loss per foot for this pipe, and the total loss in 45 ft.?

Set the arrow on scale A to 30° F. on scale B. A straight line from 100,000 B.T.U. on scale A to 2 in. diameter on scale C intersects scale D at 0·017 in. per ft. A further straight line from this point to 45 ft. on scale F gives the total pressure loss on scale E as 0·76 in.

Note: In using the calculator take special care in setting the adjustable scale A that it is directly in line with the fixed vertical lines at the top and bottom of the scale.

In cases where the total length of pipe is greater or less than given on scale F, take a value in proportion and adjust the total pressure accordingly.

EXAMPLE 4. In example 3, assume the total length of pipe to be 150 ft. Read the value on scale E for 15 ft. on scale F, namely, 0·25 in. The actual total loss would be

$$0{\cdot}25 \times \frac{150}{15} = 2{\cdot}5 \text{ in.}$$

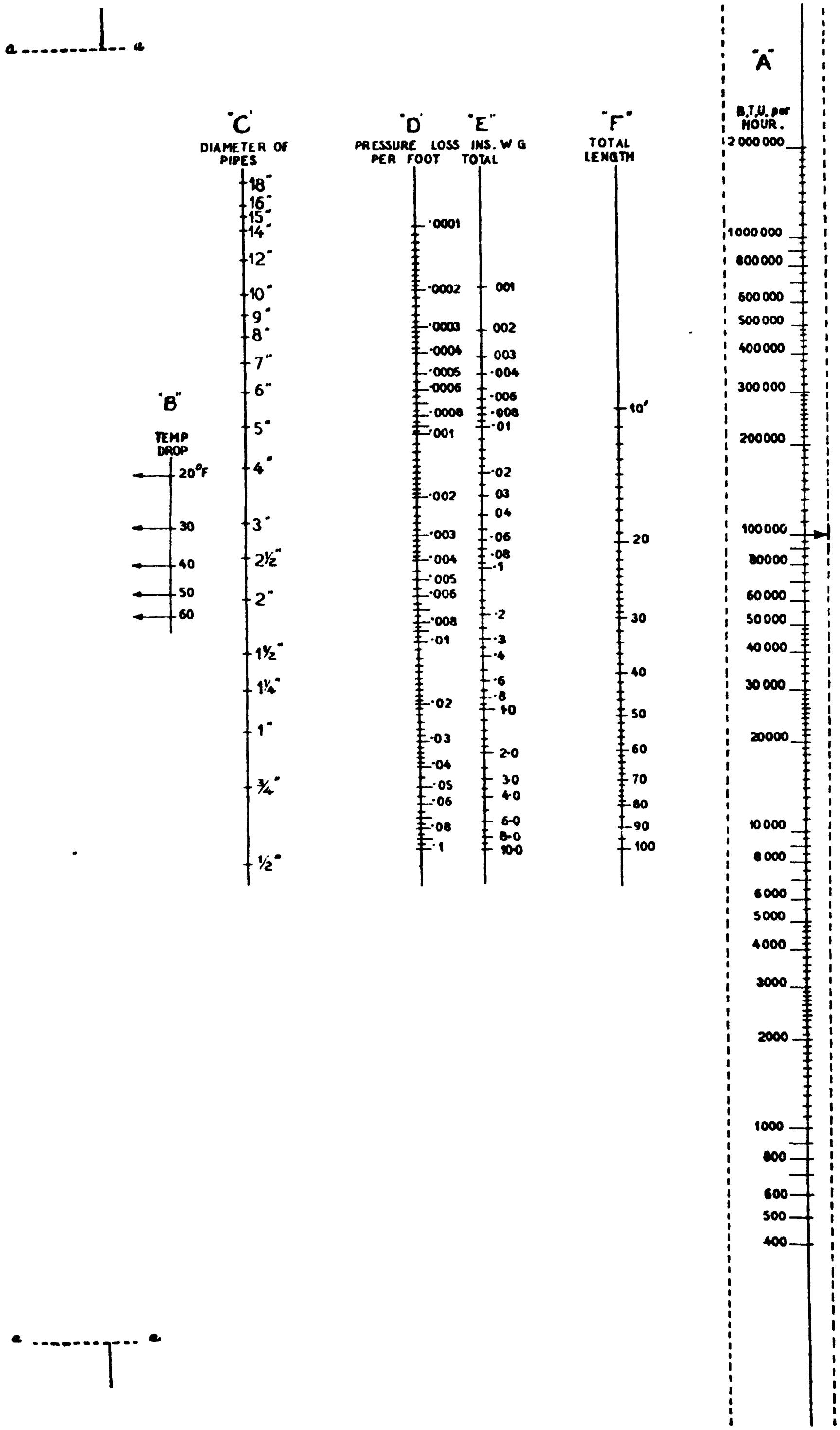


Fig. 1. NOMOGRAPHIC CALCULATOR.
GRAVITY HOT-WATER HEATING-PIPE SIZES

further scale is introduced to facilitate direct reading of the total pressure loss in inches of water column for various lengths of pipe. In using this chart, which is intended only for gravity hot-water heating systems, where total pipe lengths exceeding 100 ft. or less than 10 ft. are to be referred to the chart, the reading may be taken either 10 times as great or 1/10th of the actual, the decimal point being moved accordingly in reading off the total pressure loss.

For example, with a system working at 40° F. temperature drop it is seen that a 3-in. pipe delivering 100,000 B.T.U. per hour has a pressure loss of 0·00146 in. per ft. If this pipe had a total length of 130 ft., which value is not given on the total length scale, the value of 13 ft. should be employed, and by reference to the chart the total pressure drop allowing for moving the decimal point one place to the left is read as 0·19 in. W.G. If, on the other hand, the total length had been 30 ft., direct reference could be made and the total loss for this length would be 0·044 in. W.G. It is felt that the method of using this chart will be easily understood, and in other examples the readings taken from the chart will be employed without reference to the method of obtaining them.

Circulating Pressure.

The available head or pressure causing circulation produced by a difference of temperatures between rising and falling columns is derived from the following formula:†

$$ah = 24h\frac{W_r - W_f}{W_r + W_f},$$

where ah = circulating pressure in inches W.G.,

h = total vertical projection,

W_f = specific weight of water at the mean calculated temperature of the flow column,

W_r = specific weight of water at the mean calculated temperature of the return column.

The nomograph in Fig. 2 enables circulating pressures to be obtained for various temperature conditions, and its use is described on p. 4.

Temperature Drop in Pipes.

Recknagel described‡ in great detail in pipe-sizing examples the facilitation of calculations for temperature drop in hot-water heating pipes by the use of tables of temperature drop. In designing one-pipe

† For derivation see *A Treatise on Central Heating*, by F. Broadhurst Craig.
‡ *Die Berechnung der Warmwasserheizungen*, by H. Recknagel.

heating systems in particular the drop of temperature in the pipes has a deciding influence on the proportions of the system. The general tendency of design if accurate calculation is considered at all is to calculate separately the heat losses from the various sections of pipe, then to determine the drop of temperature as a proportion of the total temperature drop for which the system is to be designed by the ratio of the heat loss in any particular circuit to the total heat carried. The temperature-drop tables avoid much tedious figuring which may well be eliminated.

The loss of heat from exposed heating surfaces may be stated by the following expression:

$$E = Sk(T_s - T_a),$$

where E = total emission in B.T.U. per hour,

S = heating surface in square feet,

k = coefficient of heat transmission in B.T.U. per degree difference of temperature between the water in the pipe and the surrounding air,

T_s = temperature of the pipe surface,

T_a = temperature of surrounding air.

It may also be definitely stated that

$$T = \frac{(T_s - T_a)Sk(T_f - T_r)}{E},$$

where T = drop in temperature of water in degrees per foot run of pipe,

T_f = flow temperature degrees Fahr,

T_r = return temperature degrees Fahr.

Method of Employing Circulating Pressure Nomograph (*Fig.* 2).

It will be observed that this nomograph has the following scales:

A, indicating flow temperatures in degrees Fahr.

B, indicating circulating pressure in inches of water-column for a height of 1 ft., for the differences of temperature shown on scale.

C, which is also in degrees Fahr.

EXAMPLE 1. The flow temperature is 180 F. and return temperature 140° F. What is the circulating pressure for these temperatures?

A straight edge placed at 180° F. on the flow temperature scale A and 180 − 140 = 40° F. on the temperature drop scale C, intersects the circulating pressure scale B to give 0·16 in. W.G.

EXAMPLE 2. What return temperature would be obtained if the desired circulating pressure per foot of height were 0·2 in. W.G., with a flow temperature of 190° F.?

A straight line through 190° F. on scale A and 0·2 in. on scale B gives on C a temperature drop of 46° F. approximately, so that the return temperature will be 190 − 46 = 144° F.

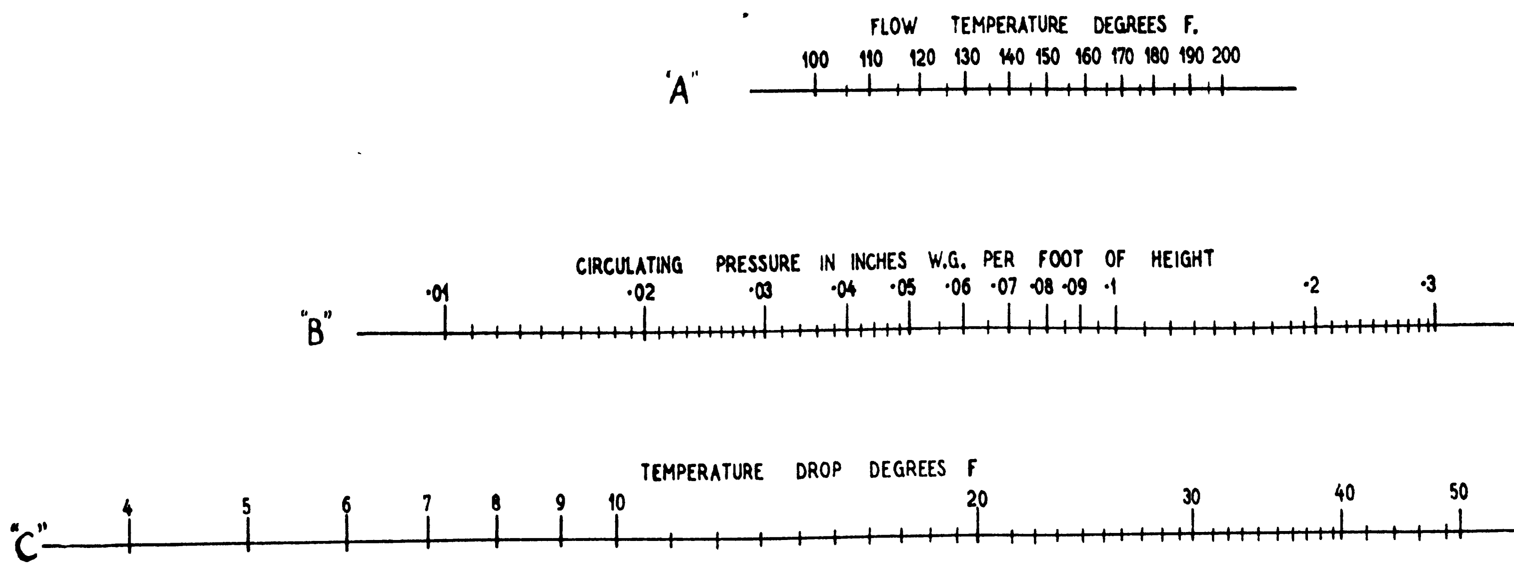


Fig. 2. Nomograph for Circulating Pressure.

This last equation is an algebraic statement that the drop of temperature is directly proportional to the heat loss from the pipe compared with the total B.T.U. which it carries, this being the fundamental factor in all calculations of temperature drop. To facilitate the employment of this expression the nomographic calculator in Fig. 3 may be used.

It has been assumed for all purposes that the temperature of the water in the pipes is 180° F., and that the temperature of the air immediately surrounding a bare pipe exposed on the face of a wall is 70° F. and an insulated pipe on the face of a wall 60° F. For bare pipe, either in wall chases or trenches, the temperature of the air surrounding the pipe is taken as 100° F., whilst for insulated pipes in similar conditions the air temperature is taken as 90° F. In all cases the temperature of the air in a roof space is assumed to be 40° F., whilst the efficiency of insulating material is taken as an average value of 65 per cent.

It will be observed that this nomographic calculator has a fixed scale for the pipe diameters, whilst the B.T.U. scale is adjustable according to the temperature drop in the system and according also to pipe position. Opposite the chart brief instructions are given for its use, so that we shall not refer to this again, mentioning only the values read from the chart. The simplicity of this chart can be realized by comparison with the author's tables† for similar problems.

As an example in the simple use of Fig. 3 we will determine what the water temperature would be at the end of a $\frac{3}{4}$-in. pipe, 50 ft. long, delivering 20,000 B.T.U. per hour, if the pipe is insulated and fixed in a chase, with an initial temperature of water in the pipe of 180° F. and a difference of temperature between flow and return pipes of 40° F.

For this condition, by reference to the chart, we find

$$T = 0{\cdot}032.$$

The total drop of temperature $= 0{\cdot}032 \times 50$

$= 1{\cdot}6.$

The temperature at the end of the pipe $= 180 - 1{\cdot}6$

$= 178{\cdot}4.$

In using this graph it should be remembered that for a B.T.U. value 10 times greater than that given, the value of T would be 1/10th as great, so that if the pipe taken in the above example were

† 'Temperature Drop in Pipes', by A. T. Henly, *The Heat and Vent. Engineer*, Jan. 1928.

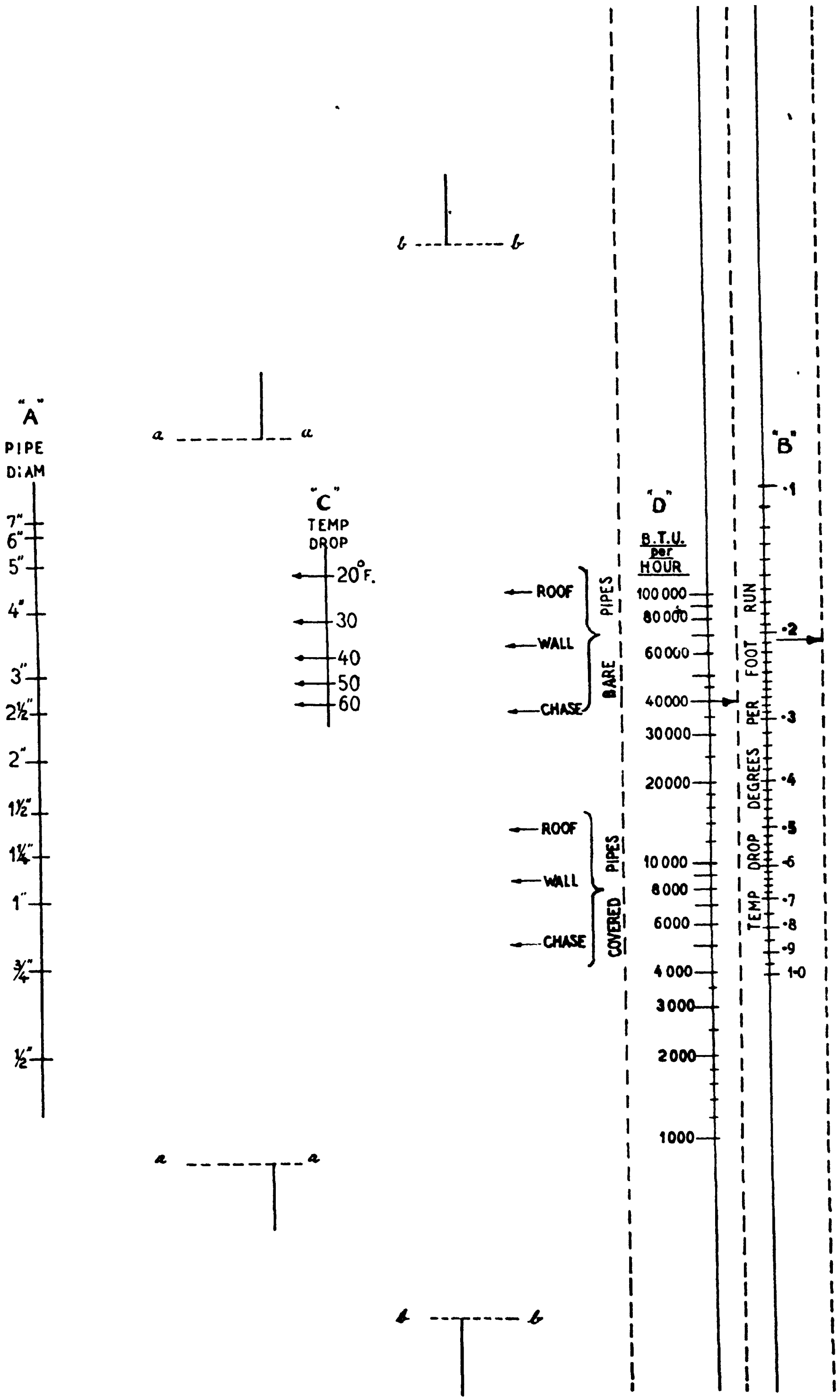


FIG. 3. NOMOGRAPHIC CALCULATOR FOR TEMPERATURE DROP IN PIPES

Method of Using Nomographic Calculator for Finding Temperature drop in Pipes. (Fig. 3.)

This special type of nomographic calculator combines the functions of the nomograph and slide rule. It consists of the following scales:

A, a scale of pipe diameters.

B, an adjustable scale of temperature drop in degrees Fahr. per foot run of pipe.

C, a scale of total temperature drop in the system.

D, an adjustable scale giving B.T.U. per hour, passing through the pipe.

Before this calculator can be employed it is necessary to cut out carefully from the page scales D and B, the cuts being made with a sharp knife along the three dotted lines running from top to bottom of the page. *A suitable backing should be placed behind the page before doing this, to avoid damaging other pages.* Further cuts should be made at the short horizontal lines $a-a$ and $b-b$. The ends of scale D are placed through the cuts at $a-a$, so that scale D can be moved vertically in such a way that the arrow may be set to any desired value of total temperature drop in the system on scale C. Similarly, the ends of scale B are placed through the slots $b-b$, enabling scale B to be adjusted vertically, the arrow being set to the line representing the situation of the pipes and whether insulated or bare.

EXAMPLE 1. An insulated 6-in. pipe, in a chase, carries 20,000 B.T.U. per hour, the total temperature drop in the system being 30° F. What is the temperature drop in this pipe if it is 100 ft. long?

Set the arrow on scale D to 30° F. on scale C and the arrow on scale B to covered pipes in chase. A straight line through 6 in. diameter on scale A and 20,000 B.T.U. on scale D intersects scale B at a temperature drop of 0·121° per foot run, so that the total drop is $0{\cdot}121 \times 100 = 12°$ F. approximately.

EXAMPLE 2. If a 4-in. bare pipe on the wall leaves a boiler at 180° F. and carries 20,000 B.T.U. per hour, the total temperature drop in the system being 40° F., what is the water temperature in the pipe 50 ft. away from the boiler?

Set scale D to 40° F. and scale B to bare pipes on the wall. A straight line through 4 in. on scale A and 20,000 B.T.U. on scale D gives on scale B a temperature drop of 0·53 degrees per foot run. The temperature at the end of 50 ft. would therefore be $180-(50 \times 0{\cdot}53) = 153{\cdot}5°$ F.

Note: In using this calculator, take especial care in setting the adjustable scales that the scale is directly in line with the short fixed vertical lines at the top and bottom of the scales.

In cases where a value for B.T.U. greater than that given on the scale is concerned, as for example 1,000,000 B.T.U., use 100,000 on the scale, dividing the temperature-drop value by $\frac{1{,}000{,}000}{100{,}000} = 10$.

Similarly, if any particular example gives a temperature-drop value outside the scale, choose a B.T.U. value which is some multiple or proportion of the actual B.T.U., adjusting the indicated temperature drop accordingly, always remembering that a greater value of B.T.U. will mean a proportionately less value of temperature drop. That is, if the B.T.U. is taken as $\frac{1}{10}$ of the actual figure, the actual temperature drop will be $\frac{1}{10}$ of that indicated.

passing 200,000 B.T.U. per hour the total temperature drop would be 0·16. The B.T.U. values in the graph are given for the differences of temperature commonly employed of 20° F., 30° F., 40° F., 50° F., and 60° F.

Effect of Temperature Drop on Radiator Temperature.

We will now consider a simple system as in Fig. 4 and investigate the effect of temperature drop in pipes upon the temperature of radiators. In designing most hot-water heating systems the flow temperature is decided as 180° F. and the return as 140° F., the mean temperature being 160° F., which is assumed to be the mean temperature of the radiator. This represents a simple two-pipe system. It will be assumed that the B.T.U. carried by the system = 10,000, diameter of pipes = $\frac{3}{4}$ in., and all pipes are to be considered as insulated and exposed on the face of the wall to a room temperature of 60° F.

From Fig. 3 we find the drop of temperature T = 0·087° F. per foot run. The length of the piping from boiler to radiator is 67 ft. The temperature loss in this length is

$$0{\cdot}087 \times 67 = 5{\cdot}83^\circ \text{F}.$$

The temperature at the entrance to the radiator is

$$180 - 5{\cdot}83 = 174{\cdot}17^\circ \text{F}.$$

If there is to be a 40° difference of temperature between the flow and return, the temperature at point F will be

$$174{\cdot}17 - 40 = 134{\cdot}17.$$

The return pipe is 50 ft. long and from previous figures T = 0·087° F., where the water temperature is 180° F. For the average return temperature of 140° F., T would be corrected proportionately to the respective difference between water and surrounding air temperatures, namely:

$$\frac{(140-60)}{(180-60)} \times 0{\cdot}087 = 0{\cdot}058^\circ \text{F}.$$

There is thus a further temperature drop in the return of 0·058 × 50 = 2·9° F. From these calculations it appears that the water reaches the boiler 48·73° F. lower than it leaves, which is the case, provided that the emission of 10,000 B.T.U. per hour actually occurs, and that no account is taken of heat loss from the pipes when calculating pipe sizes.

The mean temperature of the radiator would be

$$\frac{174{\cdot}17 + 134{\cdot}17}{2} = 154{\cdot}17^\circ \text{F}.,$$

instead of the 160° F. generally assumed.

It is usually not necessary to take into account the drop of temperature in pipes as affecting the amount of radiator surface for a two-pipe system, as 10° F. variation in the mean water temperature would have little effect on the transmission. In the case of a one-pipe

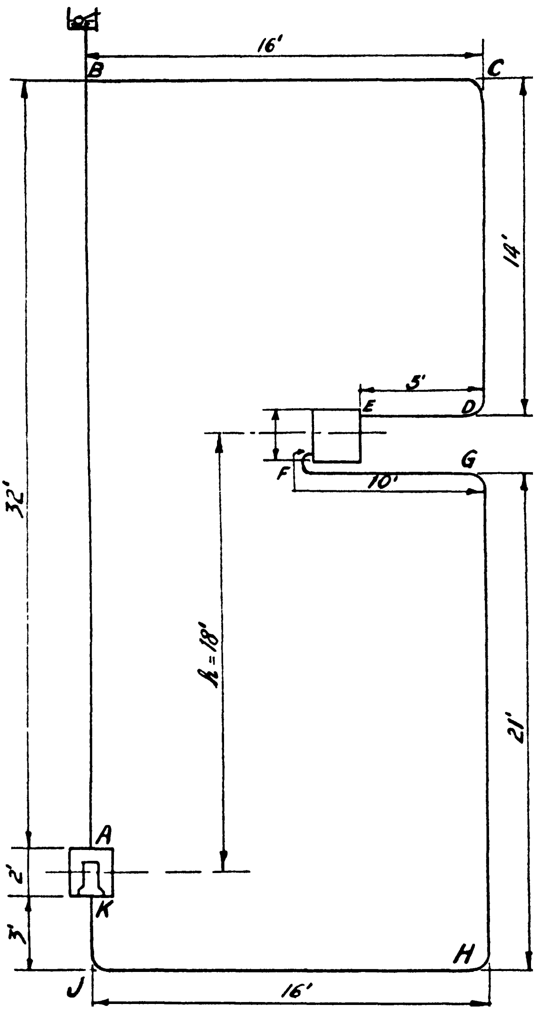


FIG. 4. One-pipe simple system.

system, however, in which the branch-flow pipe to each radiator branches from the main pipe, and back again into the same main pipe, temperature drop must be taken into account.

Up to this point calculations have been based upon the assumption that there is no loss of heat from the pipe connecting to the radiator which, although incorrect of course, serves to illustrate the problem discussed. Strictly, in addition to the emission of 10,000 B.T.U. per hour from the radiator, illustrated in Fig. 4, there is a further definite emission from the pipes, and this quantity, as will be shown later, has an important bearing upon the actual pressure loss in the pipes, due to friction, and also upon the pressure available for circulation.

Before it is possible to determine any numerical values for these latter factors we must first find what the heat loss from the pipes would be. Considering the system in Fig. 4 from the table below, for insulated pipes on the face of a wall we find an average emission from the ¾-in. diameter pipe of 22 B.T.U. per lineal ft. per hour. The total loss for the 67 ft. of flow pipe is thus $67 \times 22 = 1{,}474$ B.T.U. per hour. The return pipe, at the lower mean temperature of 140° F., would have an emission of

$$\frac{(140-60)}{(180-60)} \times 22 = 14{\cdot}5 \text{ B.T.U. per lineal ft.}$$

Table showing average emission of heat by hot-water pipes under various conditions

	Size of pipe and emission in B.T.U. per hour per foot run											
Conditions	½	¾	1	1¼	1½	2	2½	3	4	5	6	7
Bare on face of wall .	50	67	88	105	123	151	182	208	264	316	355	372
Insulated on face of wall .	16	22	28	34	40	49	59	67	85	202	114	120
Bare in chases . . .	36	49	64	76	90	110	132	151	192	230	258	271
Insulated in chases . .	12	16	21	25	30	37	44	50	64	76	86	90
Bare in roof . . .	63	86	112	154	157	192	232	265	335	402	451	600
Insulated in roof . .	20	27	36	43	50	61	74	84	107	128	144	191

It should be noted at this point that the ratio of the difference of temperature of the flow pipe and its surrounding air to the difference of temperature of the return pipe and its surrounding air is a constant, namely, 0·66, which figure we shall in the future use.

The total loss from the 50 ft. of return piping is thus $50 \times 14{\cdot}5 = 725$ B.T.U. per hour.

The total B.T.U. which should be carried by the ¾-in. diameter pipe is

	B.T.U. per hour
Radiator	10,000
Flow pipe	1,474
Return pipe	725
Total	12,199

As the actual pressure consumed in the pipes varies as the velocity $v^{1{\cdot}86}$ and B.T.U. varies directly as the velocity, then P, the pressure, also varies as B.T.U. to the power of 1·86. The pressure consumed in the pipes is thus $\left(\frac{12,199}{10,000}\right)^{1\cdot 86}$ = approximately 1·44 times as great as that determined when neglecting the heat loss from pipes.

The Increase of Circulating Pressure due to Cooling of Pipes.

Although the pressure consumed would be apparently 1·44 times as great as that when pipe-heat losses are neglected, it should be remembered that another factor, namely, the influence of the cooling of the pipes upon the available head, or pressure, is at the same time exercising a balancing effect, until the system finally functions at a very slightly greater difference between the boiler flow and return temperature than that for which the system appears to have been designed. In order to analyse this conception of the problem, we must first find the theoretical temperature at each point in the system. The lengths of the various horizontal and vertical pipes are as follows (from Fig. 4):

AB = 32 ft.	FG = 10 ft.
BC = 16 ,,	GH = 21 ,,
CD = 14 ,,	HJ = 16 ,,
DE = 5 ,,	JK = 3 ,,

The two opposing columns of water, the difference in weight of which causes the circulation, are $JKAB$ as the rising column, and $BCDEFGHJ$ as the falling or return column. The temperature drops at the various points are found as follows:

Pipe	*Length*	*Value of T*	*Temp. drop in length*	*Temperature at various points. Initial temp. at A = 180° F.*
A to B	32 ft.	×0·087	= 2·79°	180 − 2·79 = 177·21° at B
B to C	16 ft.	×0·087	= 1·39°	177·21 − 1·39 = 175·82° at C
C to D	14 ft.	×0·087	= 1·22°	175·82 − 1·22 = 174·60° at D
D to E	5 ft.	×0·087	= 0·44°	174·60 − 0·44 = 174·16° at E
	40° F. drop through radiator			174·16 − 40 = 134·16° at F
F to G	10 ft.	×0·058	= 0·58°	134·16 − 0·58 = 133·58° at G
G to H	21 ft.	×0·058	= 1·22°	133·58 − 1·22 = 132·36° at H
H to J	16 ft.	×0·058	= 0·93°	132·36 − 0·93 = 131·43° at J
J to K	3 ft.	×0·058	= 0·17°	131·43 − 0·17 = 131·26° at K

From here we see that the temperature at K, the boiler return, is 131·26° F. That is to say, instead of 40° F. drop in temperature, there is actually about 49° F. drop. Whether the heat loss of piping is taken into account or not, this is inevitable if the drop of 40° F. through the radiator is maintained.

The next stage is the determination of the available head under the temperature conditions just established. There are two methods in general use for this purpose: either to find the weight of water in all the vertical sections of piping, and then the difference in weight of the rising and falling columns; or alternatively, to employ Recknagel's method, which is to find the mean temperature of each section of the rising and falling columns, multiplying the mean temperature of any particular section by its vertical projection, and dividing the total of these sums by the total vertical projection to obtain a mean temperature for the whole of the column considered. This is done for both rising and falling columns as follows:

Section of piping	*Vertical projection*	*Mean temperature °F.*	*Mean temperature vertical projection.*
A to *B*	32 ft.	$\frac{180+177\cdot21}{2} = 178\cdot60$	5715·2
K to *A*	2 ft.	$\frac{180+131\cdot26}{2} = 155\cdot63$	311·26
J to *K*	3 ft.	$\frac{131\cdot43+131\cdot26}{2} = 131\cdot35$	394·05
			6420·51
Total	37 ft.		
		Mean temperature of rising column $= \frac{6420\cdot51}{37} = 173\cdot5^\circ$ F.	
C to *D*	14 ft.	$\frac{175\cdot82+174\cdot60}{2} = 175\cdot21$	2452·94
E to *F*	2 ft.	$\frac{174\cdot16+134\cdot16}{2} = 154\cdot16$	308·32
G to *H*	21 ft.	$\frac{133\cdot58+132\cdot36}{2} = 132\cdot97$	2792·37
Total	37 ft.		5553·63
		Mean temperature of falling column $= \frac{5553\cdot63}{37} = 149\cdot7^\circ$ F.	

It is then possible to obtain the available head or pressure causing circulation from the nomograph in Fig. 2.

With the new mean flow and return temperatures the available head for a vertical projection of 1 ft. is found as 0·097 in. W.G. The total available head for actual temperature conditions taking into account cooling of water is $37 \times 0\cdot097 = 3\cdot59$ in. of water.

It is not proposed to detail at this stage the calculation of pressure loss in pipes. The ¾-in. circulation to the radiator, if calculations were based upon the load of 10,000 B.T.U. for the radiator alone, would be found by referring to the pipe-sizing chart, Fig. 1, to have a pressure loss of 0·0183 in. per ft. pipe, the total pressure consumed becoming $0\cdot0183 \times (117+15)$, these last two figures representing the actual length of pipe and an allowance for resistance of valves, elbows, etc., respectively. The total consumed pressure for this length is found from the chart as 2·4 in. of water.

For the B.T.U. value of 12,199 for radiator and piping, we have determined the pressure loss to be 1·44 times as great, or $2{\cdot}4 \times 1{\cdot}44 = 3{\cdot}45$ in. of water.

The difference between the actual loss and the calculated available head is almost negligible, being only 4 per cent. of the available pressure. From this calculation we may see very clearly the compensating action in hot-water circulation, for as the pressure consumed increases with the loss of heat from the pipes, so the available head increases, which proves an earlier statement—that the circulation will be satisfactory in practice although the temperature difference between boiler flow and return may be greater than that calculated upon, even if the heat loss from pipes is ignored.

Allowances for Pipe-heat Losses.

Professor Barker in his book† describes at some length what must be an accurate system of pipe sizing. By his method one must first sum up the B.T.U. values of all the radiators and pipes from the farthest radiator back to the boiler, and by dividing the total thus obtained by the required temperature drop, find the total weight of water circulated per hour. This quantity is then proportioned between the various circuits, according to the respective quantities of heat carried, until once again we arrive at the end radiator, with the probability of finding a ridiculous result if a slight error has been made early in the calculations, as the author has often found by his own experience.

Yet another method‡ has been used which eliminates many tedious calculations, namely, that of adding some percentage of the radiator B.T.U. to account for pipe losses. This method also has its disadvantages.

Following upon the last calculation, it is felt that it is not unreasonable to suggest neglecting pipe losses altogether in pipe sizing.

When neglecting pipe losses for sizing purposes the following points should be remembered. If the system is sized for a temperature drop of 30° F., through the radiators, the ultimate difference in temperature between boiler flow and return pipes will be about 40° F. where the loss of heat from the mains is 50 per cent. of that from radiators, and 50° F. where the loss of heat from mains is 110 per cent. of that from radiators.

† *Barker on Heating*, by A. H. Barker, B.Sc., etc.

‡ *A Treatise on Central Heating*, by F. Broadhurst Craig.

If the system is sized for 40° F. drop in temperature and mains lose 40 per cent. of the heat losses from radiators, the ultimate temperature drop will be 50° F. This represents the condition obtained in the average two-pipe system.

The actual temperature drop is not of importance provided that it does not exceed 60° F. In practice the loss of heat from uninsulated pipes compared with radiators is about 100 per cent. for one-pipe systems, and anything from 25 to 75 per cent. for two-pipe systems. This points to the desirability of sizing one-pipe systems on 30° F. drop of temperature through the radiators; two-pipe systems may be sized on 40° F. drop if the pipe loss is not likely to be more than 60 per cent. The balancing of pressure losses in the various branch circuits is by far the most important point to be considered.

Two-pipe rising and falling systems with radiators situated above the centre line of the boiler have been dealt with at length in many works,† as also have one-pipe drop systems, so that it is proposed to deal only with special applications of one- and two-pipe systems and their calculations by methods not previously published in England. Some of the methods suggested can be applied to other systems, but references are afterwards made to these possibilities.

Temperature Drop in Pipes as a Deciding Factor in the Calculation of Irregular Circulations.

The value of the temperature-drop tables or charts is not generally realized in this country. For example, we will now consider a one-pipe drop system where the greater part of the heating surface is at or below the boiler-level, as shown in Fig. 5.

Other English text-books, without exception, advocate calculating a system such as this, firstly by taking a rough guess at the pipe sizes likely to be required, the degree of success obtained in this process depending largely upon experience with similar installations. Assuming this guess to have been correct, the designer would be in a position to find the heat loss from each length of pipe, as a first step towards calculating temperature drops. The next step would be to find the weights of the rising and falling columns and then the available circulating pressure and pipe sizes.

In the heating of private houses in particular, the conditions are generally such that a sunk boiler chamber is unobtainable, with the result that the radiators have all to stand either on the same level or below the level of the boiler. Consequently, there is no h (the distance of the centre of the radiator above the centre of the boiler)

† See *Ideal Manual*, and previous references.

to be measured from plan, whilst it may even have a negative value. The only force available for circulating the water in the system is that derived from the cooling of the pipes.

Contrary to other systems, the radiator nearest the boiler on the same plane is usually in the most unfavourable position, for as it has the shortest length of piping, this offers the worst conditions for cooling the water.

On the other hand, the large diameter of the piping with its higher velocity is also unfavourable for obtaining cooling effect. The cir-

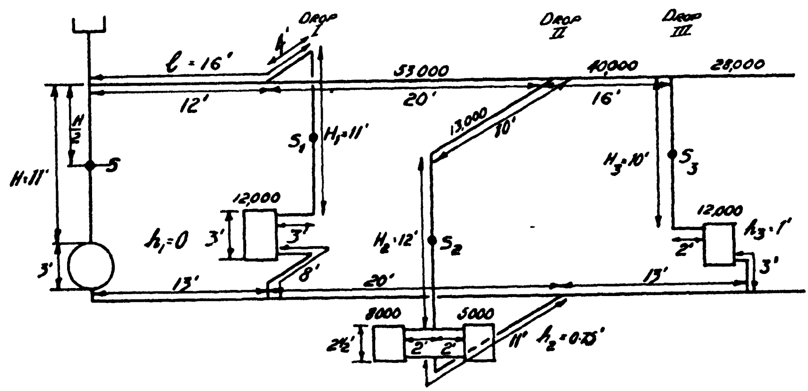


Fig. 5. One-pipe drop system.

culations farthest from the boiler, whilst they are in the most favourable position for cooling, have an increased travel. As a rule, the increased available pressure derived from the extra cooling is balanced by the increased pressure loss due to the longer circulation.

In Fig. 5, if H_1 is the vertical distance of the nearly horizontal main distributing pipe above the entrance to the radiator, then there is an available pressure head due to the difference in density between this column of water H_1 ft. high and that of an equal height in the rising column. In addition to this, there is the available head obtained due to the position of the radiator relative to the boiler. This is obtained from the measured distance of the centre of the radiator above or below the centre of the boiler. If the radiator is above the boiler, this available head is positive and must be added to that available through cooling of the pipes. If the radiator is below the boiler, this available head is negative and must be deducted from the head due to cooling.

Recknagel suggests† the procedure of finding the difference in temperature at the midpoints of the rising and falling flow pipes,

† See footnotes to page 3.

and then from the formula for the available head, the increased pressure head due to a difference of temperature between the two columns, in addition to that due to the position of the radiator relative to the boiler.

The difference of temperature between the water in the rising pipe and the drop pipe I at half the height, above the radiator and the boiler, is $(0{\cdot}5H+l_1+0{\cdot}5H_1)\times T$, where l_1 is the length in feet of the horizontal connexion from the rising pipe to drop pipe I, and T the nearest temperature drop per lineal foot of pipe in degrees Fahr.

The distance between the centre of the vertical portion of the drop pipe S_1 and the midpoint S of the rising pipe, multiplied by the average of T, gives the approximate difference of temperature between S and S_1. If this temperature difference is known, then the pressure due to the difference in density of the two columns may be found by multiplying the difference in specific weights of the water at the two temperatures by the height H_1. This gives the pressure in feet of water column, and by multiplying by 12 we have the circulating pressure in inches of water column, which must be done to bring figures in line with the formula on page 3.

Now for a temperature range from 190° F. to 170° F., which is the temperature range to be expected in flow pipes, the circulating pressure for a difference in temperature between flow and return columns of 1° F. is approximately 0·004 in. of water for each foot of height H_1.

Hence the temperature drop between S and S_1 is multiplied by H_1 and 0·004 to find the pressure available, due to cooling in pipes. We may express this as a formula, as follows:

Where P_1 = average circulating pressure in inches of water column,

$$P_1 = 0{\cdot}004H_1\,T(0{\cdot}5H+l_1+0{\cdot}5H_1).$$

This represents the additional pressure due to cooling in pipes, and if a represents the circulating pressure available for 1 ft. of height of the radiator centre-line above that of the boiler, for the boiler flow and return temperatures for which the system is calculated, and h_1 the actual difference of level between boiler and radiator centre-lines, whilst P_1 is the total available circulating pressure, then

$$P_1 = 0{\cdot}004H_1\,T(0{\cdot}5H+l_1+0{\cdot}5H_1)\pm ah_1.$$

The value of ah_1 may be obtained from Fig. 2 on page 5.

Now let L_1 represent the total length of pipe connecting the boiler with the radiator drop I, then $P_1 = L_1p_1$, where P_1 is the total available head, and p_1 is the average loss of pressure per foot run in the complete circuit from the boiler. Then we may say that the relation between the available and consumed pressures must be

$$0{\cdot}004H_1\,T(0{\cdot}5H+l_1+0{\cdot}5H_1)\pm ah_1 = L_1p_1,$$

whence $$p_1 = \frac{0{\cdot}004H_1\,T(0{\cdot}5H+l_1+0{\cdot}5H_1)\pm ah_1}{L_1}.$$

If we now consider radiators on drops II and III, it may be seen that the same expression may be used, except that the symbols will need to be given different values.

It is evident from the above that, in order to calculate a particular system, the only unknown to be determined is T. Experience in the use of this method of calculation and in the use of the temperature-drop tables will enable this to be done. For pipes near to the boiler, T is far less than for those at the end of the system.

In practice it will be found that generally T varies between 0·08 near the boiler to 0·22 at the end of the system.

We may now apply the previous remarks to circuit I in the system shown in Fig. 5, where maximum boiler-flow temperature = 180° F. and the difference between flow and return = 40° F. To designers who are in the habit of calculating pipe sizes every day, a rough guess at the pipe diameters is probably the simplest way of determining an arbitrary value for T.

Suppose we assume the diameter as 2 in. near the boiler, then from Fig. 3 for uninsulated pipes on the face of the wall, and for 65,000 B.T.U., the total for the system (see Fig. 6), we find approximately 0·09° F. as the value of T. The remaining values to substitute in the formula are as follows:

H_1 = 11 ft. = vertical distance from horizontal main to the entrance to radiator drop I.

l_1 = 16 ft. = length of the horizontal connexion from flow riser to drop circuit I.

a = 0·16 in. water = the pressure head available for a height of 1 ft., flow temp. = 180° F. and return temp. = 140° F., taken from Fig. 2.

h_1 = 0 = vertical distance from centre of boiler to centre of radiator.

L_1 = 62 ft. = total travel of water from boiler to radiator on circuit I and back to boiler.

Substituting in the formula above, we have

$$p_1 = \frac{0{\cdot}004\times 11\times 0{\cdot}09(5{\cdot}5+16+5{\cdot}5)\pm 0}{62} = 0{\cdot}00173.$$

Before we actually consider any sizes it should be mentioned that in making allowance for the resistance due to elbows, tees, etc., a round elbow has been taken to have a resistance value E of 0·5, a square elbow or tee 1·0, entrance to radiator or boiler 2·0, regulating valve 2·0, these figures being those given by Rietschel. The equivalent lengths of piping to be added to the actual straight lengths, to take account of resistance of $E = 1·0$, for various diameters of pipes, are given in the table below.

Table of single resistance (E) expressed in feet length of pipe for diameters varying from ½ in. to 7 in., and E = 1·0 up to 5·0

D	$E = 1$	1·5	2	2·5	3	3·5	4	4·5	5
inches									
½	1·0	1·5	2·0	2·5	3·0	3·5	4·0	4·5	5
¾	1·5	2·3	3·0	3·8	4·5	5·3	6·0	6·8	7·5
1	1·9	2·9	3·8	4·8	5·7	6·7	7·6	8·6	9·5
1¼	2·6	3·9	5·2	6·5	7·8	9·1	10·4	11·7	13·0
1½	3·2	4·8	6·4	8·0	9·6	11·2	12·8	14·4	16·0
2	4·5	6·8	9·0	11·3	13·5	15·8	18·0	20·3	22·5
2½	5·9	8·9	11·8	14·8	17·7	20·7	23·6	26·6	29·5
3	7·2	10·8	14·4	18·0	21·6	25·2	28·8	32·4	36·0
4	10·0	15·0	20·0	25·0	30·0	35·0	40·0	45·0	50·0
5	13·0	19·5	26·0	32·5	35·0	45·5	52·0	58·5	65·0
6	16·0	24·0	32·0	40·0	48·0	56·0	64·0	72·0	80·0
7	19·0	28·5	38·0	47·5	57·0	66·5	76·0	85·5	95·0

The calculations of pipe sizes for the average available circulating pressure, together with the actual pressure losses taken from Fig. 1, are set forth below.

		Average available circulating pressure	*Actual pressure loss per foot of pipe*	*Diameter of pipe*	*Coefficient of single resistances*	*Equivalent length of single resistances*	*Actual length of pipe*	*Total length of pipe for calculation*	*Total loss of pressure in length of pipe*
Pipe	*B.T.U.*	P_1	P_a	d	E	l_E	l	$l+l_E$	$P_a(l+l_E)$
Flow . .	65,000	0·00173	0·00165	2½″	3	18	23	41	0·063
Flow drop .	12,000	0·00173	0·0022	1¼″	1·5	4	18	22	0·049
Return .	65,000	0·00173	0·00165	2½″	0·5	3	13	16	0·027
Return drop	12,000	0·00173	0·0022	1¼″	3·5	9	8	17	0·038
	Actual pressure head consumed		.	.	.	.	.	.	0·182

The column under P_a gives the actual pressure losses per foot run for the diameters shown in column d. The last column gives the total loss in the particular length of pipe considered.

The actual pressure head consumed, of 0·182 in. of water, must be equal to the head found to be available when using the correct values of T from Fig. 3.

For 65,000 B.T.U., $d = 2\frac{1}{2}$ in., $T = 0{\cdot}113$.

For 12,000 B.T.U., $d = 1\frac{1}{4}$ in., $T = 0{\cdot}350$.

We must now calculate the actual available head from the formula derived on page 17, namely:

$$P_1 = 0{\cdot}004H_1T(0{\cdot}5H+l_1+0{\cdot}5H_1)\pm ah_1.$$

We have now actual values for T which are employed for the respective lengths of pipe as follows:

$$P_1 = 0{\cdot}004H_1\begin{bmatrix} (0{\cdot}113\times 5{\cdot}5) \text{ cooling of } H_2 \\ +(0{\cdot}113\times 12) \text{ overhead } 2\tfrac{1}{2}\text{-in. main} \\ +(0{\cdot}350\times 4) \quad\text{,,}\quad 1\tfrac{1}{4}\text{-in. branch} \\ +(0{\cdot}350\times 5{\cdot}5) \text{ cooling of } \tfrac{1}{2}H_1 \end{bmatrix}$$

$$= 0{\cdot}004\times 11\times 5{\cdot}3 = 0{\cdot}233.$$

As there was only 0·182 consumed in the pipes, there would apparently be a surplus of $0{\cdot}233-0{\cdot}182 = 0{\cdot}051$ in.

This may be consumed by reducing a section of the piping in diameter, preferably the return piping, as the cooling calculation is not then affected. The equivalent length of the return, that is, the actual straight length of pipe. augmented by an allowance for resistance due to elbows, etc., is 16 ft.

From Fig. 1 the loss in 1-in. pipe passing 12,000 B.T.U. is 0·0065 in. per ft., and the difference between this and the $1\frac{1}{4}$ in. for the same duty is $0{\cdot}0065-0{\cdot}0022 = 0{\cdot}0043$ in. per ft., so that in 16 ft. the additional pressure lost would be $16\times 0{\cdot}0043 = 0{\cdot}069$. This is slightly more than the surplus of 0·051, but is sufficiently close to be satisfactory, as the excess of $0{\cdot}069-0{\cdot}051 = 0{\cdot}018$ is only $\frac{0{\cdot}018}{0{\cdot}233}\times 100$ = under 8 per cent. in excess of the available circulating pressure, and would scarcely affect the circulation. In practice the surplus would be ignored and overcome by the use of a regulating valve.

The Calculation of Circuit II.

The pressure to be produced in this circuit must be sufficient to overcome the pressure loss of the common mains and the new pipes, and also the negative value due to the radiators being below the boiler, $h_2 = 0{\cdot}75$ ft., so that $ah_2 = 0{\cdot}16\times 0{\cdot}75 = 0{\cdot}12$ in. of water.

The pressure loss in the common main is $0{\cdot}068+0{\cdot}027 = 0{\cdot}095$ in.

For the new piping we may first assume a temperature drop T_2 of

0·14° F. per ft. The problem is now to find the probable difference of temperature between points S and S_2 on Fig. 5. It is advisable to start from the boiler figuring on the temperatures known for the calculations for circuit I.

The temperature drop from boiler to $S = 5·5 \times 0·1128° = 0·62°$, so that the temperature at $S = 180 - 0·62 = 179·38°$ F. The temperatures at the various other points may be determined in a similar manner.

The additional cooling due to the pipes from the point where drop I is taken off to S_2, that is, 36 ft., is $T = 0·14 \times 36$ or 5·02° F. The approximate temperature difference $S - S_2$ is thus

$$179·38 - (177·41 - 5·02) = 7° \text{ F.}$$

The pressure due to cooling is $0·004 \times 7 \times (H_2 = 12 \text{ ft.}) = 0·336$ in. of water.

From this amount must be deducted the pressure already consumed and the negative ah_2, leaving $0·336 - (0·095 + 0·12) = 0·121$ in., which is available for circuit II, a length of 57 ft. The allowable pressure drop will be:

$$\frac{0·021}{57} = 0·0021 \text{ in. per lineal ft.}$$

As before, we select by the use of Fig. 1 suitable sizes for pipes, calculating the actual pressure loss in the circuit as follows:

Pipe	*B.T.U.*	P_1	P_a	d	E	l_E	l	$l+l_E$	$P_a(l+l_E)$
				in.					
Flow	53,000	0·0021	0·0012	2½	..	..	20	20	0·024
Flow drop	13,000	0·0021	0·001	1½	1·5	5	22	27	0·027
Rad. conn.	8,000	0·0021	0·0011	1¼	3	8	2	10	0·011
Return conn.	53,000	0·0021	0·0012	2½	..	..	11	11	0·013
	13,000	0·0021	0·001	1½	1·5	5	11	16	0·016
	8,000	0·0021	0·0011	1¼	2	5	2	7	0·008
		Actual pressure consumed							0·099

The actual pressure head should be determined before sizing the connexions to the other radiator of 5,000 B.T.U., for which the actual temperature drops are obtained from Fig. 3, as

$$53{,}000 \text{ B.T.U.}, d = 2\tfrac{1}{2} \text{ in.}, T = 0·13°$$

$$13{,}000 \text{ B.T.U.}, d = 1\tfrac{1}{2} \text{ in.}, T = 0·38°$$

$$8{,}000 \text{ B.T.U.}, d = 1\tfrac{1}{4} \text{ in.}, T = 0·53°.$$

From these figures it is possible to determine the temperature at the entrance to the radiators.

The further temperature drop to point S_2 is:

$$\begin{aligned} &20 \text{ ft.} \times 0{\cdot}13 = 2{\cdot}6^\circ \\ &10 \text{ ft.} \times 0{\cdot}38 = 3{\cdot}8^\circ \\ \tfrac{1}{2}H_2 = \ &6 \text{ ft.} \times 0{\cdot}388 = \underline{2{\cdot}3^\circ} \\ &\qquad\qquad\qquad 8{\cdot}7^\circ \end{aligned}$$

The actual pressure drop between D and S_2 is thus

$$179{\cdot}38-(177{\cdot}41-8{\cdot}7) = 10{\cdot}7^\circ\text{F}.$$

The actual pressure due to cooling in pipes is

$$0{\cdot}004 \times 10{\cdot}7 \times 12 = 0{\cdot}512.$$

Deducting negative ah_2 leaves 0·392.
Deducting loss in common pipes leaves 0·297.

The excess of the available pressure over that consumed points to the value of T having been assumed too low in the approximation and illustrates very well the effect of this error. By decreasing the $1\frac{1}{2}$-in. pipe to $1\frac{1}{4}$ in., the pressure loss is increased by

$$(0{\cdot}0026-0{\cdot}001) \times 33 = 0{\cdot}053 \text{ in.},$$

making the actual pressure loss 0·152 in. At the same time the temperature drop in the drop pipe is decreased to the following:

$$\text{B.T.U. } 13{,}000,\ d = 1\tfrac{1}{4} \text{ in.},\ T = 0{\cdot}32^\circ.$$

The difference in temperature between S and S_2 is decreased by $16(0{\cdot}38-0{\cdot}32) = 0{\cdot}96$. The available head pressure is thus decreased by $0{\cdot}004 \times 0{\cdot}96 \times 12 = 0{\cdot}043$ in., making the actual available pressure $0{\cdot}297-0{\cdot}043 = 0{\cdot}254$ in. compared with 0·152 in. actually consumed, so that some reduction in sizes is still possible. This may best be done with the short radiator flow and return connexions. The length of these is 4 ft. and $E = 5$, the equivalent length if 1 in. being $4+10 = 14$ ft.

Pressure loss for 1-in. pipe passing 8,000 B.T.U. from Fig. 1 is $0{\cdot}0036 \times 14 = 0{\cdot}051$.

The loss for $1\frac{1}{4}$-in. pipe, where the equivalent length would be 17 ft., is $0{\cdot}0011 \times 17 = 0{\cdot}019$, so that the increased loss for the smaller pipe is $0{\cdot}051-0{\cdot}019 = 0{\cdot}032$ in. and the total loss for the circuit becomes $0{\cdot}099+0{\cdot}053+0{\cdot}032 = 0{\cdot}184$ in., which is sufficiently near to the available pressure, particularly as $\frac{3}{4}$-in. pipe would be found to give a loss exceeding that available. Further control would be obtained by the use of the radiator valve.

Circuit II, Radiator of 5,000 B.T.U.

The losses in pipes other than the connexions to this radiator are as follows:

$$0{\cdot}068+0{\cdot}024+0{\cdot}027+0{\cdot}033+0{\cdot}016+0{\cdot}013+0{\cdot}027 = 0{\cdot}228 \text{ in.}$$

The corrected total available head is 0·392—0·043 = 0·349 in., so that 0·349—0·228 = 0·121 in. may be consumed in the connexions.

For the flow, if we take $d = \frac{3}{4}$ in., $l = 4$ ft., $E = 5$, total equivalent length = 4+8 = 12 ft.

The actual loss per foot is, from Fig. 1, 0·005 in., and the total loss = 12×0·005 = 0·060 in.

To reduce flow or return to $\frac{1}{2}$ in. would result in its pressure loss being more than that available for the connexion, so that d must be $\frac{3}{4}$ in., any surplus pressure being consumed by valve regulation.

Circuit III.

This circuit and any further ones are calculated in exactly the same manner as detailed for circuit II.

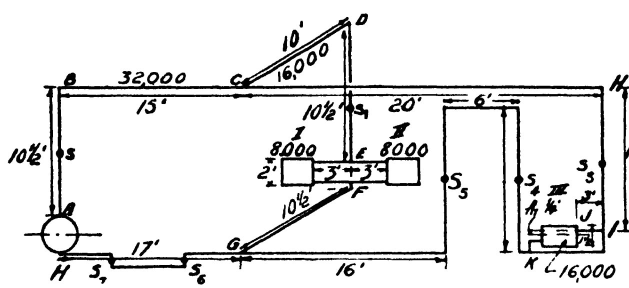


FIG. 6. Irregular circulation.

The Influence on Circulation of Pipes Rising over and Dipping under Doorways or other Obstructions.

Although many designers will not consider dipping pipes under or rising them over doorways, this may be quite safely done, and the calculations for such systems are very simple. In Fig. 6 a system is shown with the return pipe over one doorway and under another.

When a pipe, either flow or return, is raised over a doorway or similar obstruction, there is an increase in available head due to a difference in temperature between the rising and falling pipes. This temperature difference must exist because of the cooling off in the horizontal pipes, and the result is that the mean density of the rising column will thus be lower than that of the falling column.

On the other hand, with a dip under a doorway there is a decrease in, or negative value of, the available pressure head ah to be taken into account, due to the rising column having a higher mean density than the falling column.

In accordance with previous examples it is first necessary to calculate the sizes of pipes for radiators I and II. Flow temperature

$= 180°$F., return temperature $= 140°$F. $a = 0{\cdot}16$ in.; $h_1 = 0$; $h_3 = 0{\cdot}25$ ft. From the formula on page 17

$$p_1 = \frac{0{\cdot}004 \times 10{\cdot}5 \times 0{\cdot}20(5{\cdot}25+25+5{\cdot}75) \pm 0}{81}$$
$$= 0{\cdot}0036.$$

The value of $T_1 = 0{\cdot}20$ assumed above is of course purely arbitrary and may not afterwards be found correct, but at this stage of the calculation, as we have previously seen, a small error will have but little effect on the final result.

From Fig. 1 sizes are selected and the actual pressure loss in the pipe runs are calculated and tabulated as follows:

Pipe	*B.T.U.*	*P*	P_a	*d*	*E*	l_E	*l*	$l+l_E$	$P_a(l+l_E)$
				in.					
ABC . .	32,000	0·0037	0·0013	2	3	10	25·5	36	0·047
CDE . .	16,000	0·0037	0·0036	$1\frac{1}{4}$	1·5	4	21·5	26	0·094
EF . .	8,000	0·0037	0·0029	1	6	11	6	17	0·050
FG . .	16,000	0·0037	0·0036	$1\frac{1}{4}$	0·5	1	10·5	12	0·043
GH . .	32,000	0·0037	0·0013	2	3·5	12	17	29	0·038
		Actual pressure consumed							0·272 in.

From Fig. 3 we now determine the correct values of T for the sizes obtained, which are as follows:

32,000 B.T.U., $d = 2$ in., $T = 0{\cdot}19$
16,000 B.T.U., $d = 1\frac{1}{4}$ in., $T = 0{\cdot}26$
8,000 B.T.U., $d = 1$ in., $T = 0{\cdot}44$.

The corrected available pressure may then be found as:

$$P_1 = 0{\cdot}004 \times 10{\cdot}5 \left\{ \begin{array}{c} (0{\cdot}5 \times 10{\cdot}5 \times 0{\cdot}19) + (15 \times 0{\cdot}19) + (10 \times 0{\cdot}26) + \\ +(0{\cdot}5 \times 11{\cdot}5 \times 0{\cdot}26) \end{array} \right\} \pm 0$$
$$= 0{\cdot}335 \text{ in. of water gauge.}$$

We should, however, take into account the negative pressure due to the dip under the doorway.

The difference in temperature between S_6 and S_7 for the pipe diameter of 2 in., and a length of 7 ft., uninsulated and in a trench, is found from Fig. 3.

For 32,000 B.T.U., $d = 2$ in., $T = 0{\cdot}14°$, and the negative pressure is thus $0{\cdot}003 \times 1 \times 7 \times 0{\cdot}14 = 0{\cdot}003$ in., so that actually the available pressure would appear to be $0{\cdot}335 - 0{\cdot}003 = 0{\cdot}332$ in.

The value of 0·003 is here used as being a more correct value for the pressure head available for 1° F. difference in temperature at the lower mean return temperature, instead of 0·004, which is correct for flow temperatures only.

It appears, therefore, that the pressure consumed is less than that available, so that it may be possible to reduce some pipe sizes. The main flow and return pipes, having the lowest frictional loss, are those most likely to be altered. In order not to affect the circulating pressure calculation we will assume the main return to be altered to $1\frac{1}{2}$ in., for which size the loss of pressure is 0·0052 in. per ft., representing an increase of $0{\cdot}0052-0{\cdot}0013 = 0{\cdot}0039$ in. per ft. The total travel for $1\frac{1}{2}$-in. pipe would be 28 ft., increasing the pressure loss by $0{\cdot}0039\times 28 = 0{\cdot}11$ in., giving a total of $0{\cdot}272+0{\cdot}11 = 0{\cdot}382$, which now exceeds that available, so that the pipes must remain as first calculated.

This circuit serving radiator 3 is assisted by the main pipe rising to the high level above the doorway.

If we take an assumed value of T for this circuit as 0·30, there will be a drop in temperature in the horizontal main and drop CHI of $0{\cdot}30\times 41 = 12{\cdot}3°$.

The cooling effect to be taken into account for calculating the available pressure is the drop in SBC of $(0{\cdot}19\times 20{\cdot}25)+12{\cdot}3°$, that is, 16° F. $P = 0{\cdot}004\times 16{\cdot}0\times 12-(0{\cdot}25\times 0{\cdot}16) = 0{\cdot}73$ in. Of this 0·09 in. is already consumed in common mains, so that

$$p = \frac{0{\cdot}73-0{\cdot}09}{93} = 0{\cdot}0065.$$

From Fig. 1, $1\frac{1}{4}$-in. pipe is required, whilst the actual pressure loss for this pipe passing 16,000 B.T.U. is 0·0038 in. per ft.

The value of E for this circuit is 9, which is equivalent to a length of 24 ft., making with the straight pipe runs a total length of 117 ft., so that the pressure loss in this run is $0{\cdot}0038\times 117 = 0{\cdot}445$ in. The actual drop in temperature between S and S_3 is as follows:

$$\overbrace{(20{\cdot}25\times 0{\cdot}19)}^{SBC}+\overbrace{(35\times 0{\cdot}26)}^{CS_3} = 13°\text{ F.}$$

The available pressure $= 13\times 0{\cdot}004\times 12 = 0{\cdot}625$ in.

The increase of pressure due to the pipe rising over the doorway may be found as the temperature difference between S_4 and S_5, namely $18\times 0{\cdot}26$, multiplied by the constant factor 0·003, and by 0·66 for the reduced return temperature, giving an increase of 0·0093 in. Total available head is then $0{\cdot}625+0{\cdot}0093 = 0{\cdot}634$ in.

This again is far greater than that consumed, so that part of the pipe may be reduced. In this case the pressure which remained unconsumed in the first circuit should also remain on this circuit, so that a pressure of $0{\cdot}614-0{\cdot}445-(0{\cdot}332-0{\cdot}272) = 0{\cdot}109$ in. is still to be consumed. The difference in pressure loss of 1¼-in. and 1-in. pipe carrying 16,000 B.T.U. is 0·0074 in., so that $\frac{0\cdot109}{0\cdot0074}$ = say 15 ft. of the return could be made 1-in. diameter.

Self-Balancing Effect in Hot Water Circulation.

We have seen that in each circuit there is a considerable excess of pressure, resulting in practice in a decreased temperature drop in the system.

For 40° F. temperature drop we have calculated the available head of circuits I and II to be 0·332 in. The pressure loss was found to be 0·272 in. Very approximately, then, we may say that the temperature drop varies directly as the square root of the pressure loss, and in this case would be $40°\text{ F.} \times \sqrt{\left(\frac{0\cdot272}{0\cdot332}\right)} = 36°\text{ F.}$

It should be noted, however, that in this case the values for both the available head and the pressure loss would be incorrect as they were calculated for 40° F. temperature drop. This is, in effect, the natural compensating action of gravity hot-water circulation.

Firstly, the available head, depending as it does upon the difference in weight of the rising and falling water columns, would be expected to be lower. The weight of the water varies inversely as its absolute temperature. Assuming the flow temperature to have decreased 2° F. and the return to have increased 2° F., then the relative weights at the original and revised flow and return temperatures would be in the ratio of 1 to

$$\text{Flow} \quad \frac{460+180}{460+(180-2)} = 1\cdot003 \text{ approx.}$$

$$\text{Return} \quad \frac{460+140}{460+(140+2)} = 0\cdot996 \text{ approx.}$$

The available head, on the other hand, has fundamentally been shown to vary directly as the difference of the weight of flow and return columns, so that the ratio of the available heads at the original and revised temperatures will be as 1 is to $\frac{1\cdot003}{0\cdot996} = 1\cdot007$.

So that actually the available head varies but a very little, and would become

$$\frac{1}{1\cdot008} \times 0\cdot332 = 0\cdot329 \text{ in.}$$

Secondly, the pressure loss will increase as the square of the velocity or quantity flowing through the pipe system. As the velocity varies inversely as the temperature drop the ratio of pressure losses would be $\left(\frac{40}{36}\right)^2 : 1$, that is, $1{\cdot}23 : 1$, so that for 36° F. drop the revised pressure loss would be $0{\cdot}272 \times 1{\cdot}23 = 0{\cdot}335$ in.

We may now make a further trial calculation to find a revised temperature drop, as follows:

$$40^\circ \text{F.} \times \sqrt{\left(\frac{0{\cdot}335}{0{\cdot}327}\right)} = 40{\cdot}5^\circ \text{F.}$$

By trial, we may now repeat this calculation until such a condition is reached that the resulting temperature is that assumed in the calculation for finding the available and consumed heads. This would be found to give the temperature drop of the system as somewhere between 36° F. and 40° F., certainly not less than 38° F., or about 2° discrepancy in temperature drop due to a surplus or unconsumed pressure of about 22 per cent. This shows that it is in no case necessary to balance pressure losses closer than 10–15 per cent. above or below the available head and is actually an explanation of why a hot-water heating system will give good results in spite of 'rule-of-thumb' design.

This must not be taken to mean, however, that one circuit may have 5 per cent. surplus pressure and another 50 per cent., in which case short circuiting would occur in some part of the system. What it is intended to convey is, that provided the various branch circulations are balanced one with another, a general surplus pressure or excess pressure loss is of little account.

Two-pipe Systems with Flow and Return Mains at High Level.

Whilst a hot-water heating system with both flow and return mains at high level is perhaps well known in the case of circulations accelerated either by mechanical or other means, it is a very rare occurrence to meet with such a system where circulation is by gravity alone. In many cases, particularly in existing buildings, it is neither desirable nor convenient to excavate the ground floor for the purpose of forming trenches, neither is it desirable to run large-diameter main pipes above floor-level owing to their unsightliness; in these circumstances the system referred to is particularly useful.

In suggesting this system in 1928† the author believed that something novel was being put forward, but subsequently it has been found that the system is quite common in Germany and France. Indeed, *Chauffage et Ventilation*‡ reprinted in 1930 an article which

† 'Temperature drop in Pipes', *Heat. and Vent. Engineer*, Sept. 1928.

‡ 'Le Chauffage d'appartements avec retour au plafond', *Chauffage et Ventilation*, June 1930.

was originally produced by M. Durupt in 1914 dealing with the problems of this type of circulation. In referring to this matter he mentioned that the greater the height of the descending flow pipes to the radiator the greater will be the circulating pressure derived, owing to the increased cooling in pipes. It was further stated that the lower the radiator was placed, even to the extent of carrying it below boiler-level, the greater again the circulating pressure would become; but this advantage is to a large extent balanced by the fact that the greater the temperature drop becomes in the pipes the less

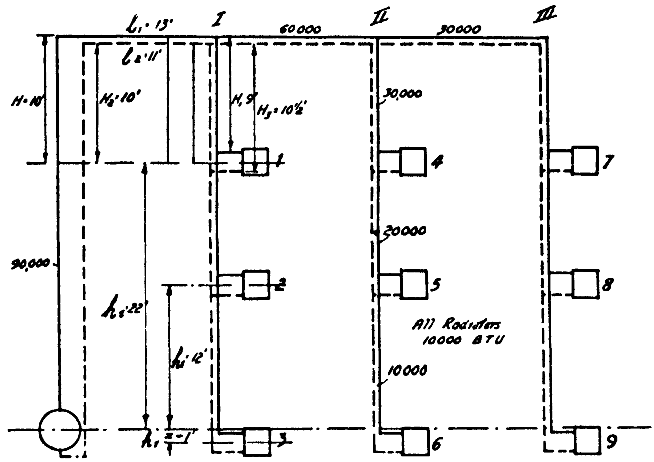


FIG. 7. Two-pipe overhead main system.

temperature drop there is available for the radiator itself for a given maximum temperature difference between flow and return at the boiler, with the result that the quantity of water required at the radiator is increased.

Mention of this type of system has also been made by R. Dupuy† in a graphical development of pressure distribution in hot-water heating systems.

The system has also been referred to by M. Grellert‡ of Wiesbaden, Ludwig Kopp,§ Helmut Scheel,|| and others.

Fig. 7 illustrates a system of this type in which some of the radiators are below the level of the centre of the boiler, and it is proposed to go into the accurate calculations for this type of system.

Let us first, however, consider the general principles involved if the only radiator on the system were number 1. Knowing that the

† Du Chauffage à eau chaude par thermo-siphon', by R. Dupuy, *Chaleur et Industrie*, 1929.

‡ *Haustechnische Rundschau*, Feb. 10 and 20, 1934.

§ 'Die örtliche Regelbarkeit der Stockwerkswarmwasserheizungen', by Ludwig Kopp, *Ges. Ing.*, Aug. 4, 1934.

|| *Die Heizungsbauschule*, by Helmut Scheel.

system as a whole is a two-pipe distribution, then the system still remains a two-pipe distribution, for the remaining radiators might only be shut off. On the other hand, the system could then be also considered as a one-pipe system with both flow and return mains taken up to high level, substantially similar to examples previously referred to in which it has been shown that by lifting flow or return pipes to high level the available pressure is increased by a definite and calculable amount. In similar terms to previous examples we may say that the available pressure for any radiator will be as follows:

$$\{0{\cdot}004H_1\,T(0{\cdot}5H+l_1+0{\cdot}5H_1)\}+\{0{\cdot}003H_3\,T_r(0{\cdot}5H_2+l_2+0{\cdot}5H_3)\}\pm ah_1,$$

where H = height of horizontal flow main above centre of radiator;

H_1 = length of flow drop to radiator;

H_2 = height of horizontal return main above centre of radiator;

H_3 = height of return riser from radiator;

l_1 = length of overhead horizontal flow;

l_2 = length of overhead horizontal return;

ah_1 = circulating pressure due to height of radiator above or below the boiler, determined as in previous examples;

T = average cooling in degrees F. per ft. run in flow piping;

T_r = average cooling in degrees F. per ft. of the return piping.

The first part of this expression gives the increase of circulating pressure due to lifting the flow main: the second part that due to lifting the return; and the third part gives the circulating pressure by virtue of the relative position of radiator and boiler.

The expression appears complicated, but is actually very simple in use, as will be seen from the calculations for Fig. 7.

It is a little difficult to decide upon an 'index' circulation, but it is suggested that the lowest radiator in the nearest drop should be the one adopted for this purpose, in the example, radiator 3.

Substituting assumed values of $T = 0{\cdot}09$ and $T_r = 0{\cdot}06$, and the various other values taken from the drawing, gives:

$$\begin{aligned}&\{0{\cdot}004\times 32\times 0{\cdot}09(16{\cdot}5+13+16{\cdot}1)\}+\\&\qquad+\{0{\cdot}003\times 34\times 0{\cdot}06(16{\cdot}5+11+16{\cdot}9)\}-(1\times 0{\cdot}16)\\&\qquad= 0{\cdot}585+0{\cdot}272-0{\cdot}16 = 0{\cdot}697 \text{ in.}\end{aligned}$$

It is interesting to notice that, in spite of the radiator being below the level of the boiler, a considerable circulating pressure is obtained.

The average pressure loss for a total measured travel of 161 ft.

$$= \frac{0{\cdot}697}{161} = 0{\cdot}0043.$$

We now proceed to obtain suitable sizes of pipes and to determine pressure losses in a similar manner to previous examples, tabulated as follows:

Pipe	*B.T.U.*	P_1	P_a	*d*	*E*	l_E	*l*	$l+l_E$	$P_a(l+l_E)$
				in.					
Flow rise .	90,000	0·0043	0·0029	$2\frac{1}{2}$	3	18	44	62	0·18
Flow drop .	30,000	0·0043	0·0046	$1\frac{1}{2}$	0·5	2	9	11	0·051
Flow drop .	20,000	0·0043	0·0022	$1\frac{1}{2}$	..	..	10	10	0·022
Rad. connexion	10,000	0·0043	0·0045	1	2·5	5	15	20	0·09
Rad. return .	10,000	0·0043	0·0045	1	2·5	5	15	20	0·09
Return rise .	20,000	0·0043	0·0022	$1\frac{1}{2}$	..	..	10	10	0·022
Return rise .	30,000	0·0043	0·0046	$1\frac{1}{2}$	0·5	2	11	13	0·06
Return drop .	90,000	0·0043	0·0029	$2\frac{1}{2}$	1·5	9	47	56	0·162
		Actual pressure consumed				.	.	.	0·677

It is assumed that the system is to work with a flow temperature of 180° F. and a 40° F. drop.

From the tabulated figures it will be noticed that the available pressure is consumed within 3 per cent., and no further reduction of sizes is necessary. It will also be observed in following the tabulated data as taken from the pipe-sizing nomograph, Fig. 1, that in some cases pipes were chosen with pressure losses slightly greater than the calculated value of P_1, which was done as an effort to balance the far less actual values found for preceding pipes, without recourse to subsequent check calculations.

It is now necessary to calculate the actual increment of pressure, due to lifting flow and return pipes, for the correct value of T and T_r. Dealing firstly with the flow pipe, by reference to Fig. 3 we find its value of $T = 0{\cdot}078$. All pipes will be assumed as bare on the face of walls. T_r will, as previously described, be $0{\cdot}66 \times 0{\cdot}078 = 0{\cdot}051$.

Similarly, the values for all flow and return drops are:

$$30{,}000 \text{ B.T.U.}, d = 1\tfrac{1}{2} \text{ in.}, T = 0{\cdot}163, T_r = 0{\cdot}107$$
$$20{,}000 \text{ B.T.U.}, d = 1\tfrac{1}{2} \text{ in.}, T = 0{\cdot}245, T_r = 0{\cdot}162$$
$$10{,}000 \text{ B.T.U.}, d = 1 \text{ in.}, T = 0{\cdot}345, T_r = 0{\cdot}228.$$

The temperature difference between $\frac{1}{2}H$ and $\frac{1}{2}H_1$ becomes:

$$(16{\cdot}5 \times 0{\cdot}078)+(13 \times 0{\cdot}078)+\{(9 \times 0{\cdot}163)+(7{\cdot}1 \times 0{\cdot}245)\} = 5{\cdot}5^\circ \text{ F.},$$

so that the pressure increment for lifting the flow pipe is actually $0{\cdot}004 \times 32 \times 5{\cdot}5 = 0{\cdot}705$ in., which exceeds that assumed in our preliminary calculation.

Similarly, the increment due to lifting the return is:

$$(16{\cdot}5 \times 0{\cdot}051)+(11 \times 0{\cdot}051)+$$
$$+\{(10\tfrac{1}{2} \times 0{\cdot}107)+(6{\cdot}4 \times 0{\cdot}162)\} \times 0{\cdot}003 \times 34 = 0{\cdot}372 \text{ in.},$$

so that the available pressure is actually

$$0{\cdot}705+0{\cdot}372-0{\cdot}16 = 0{\cdot}917 \text{ in.},$$

compared with 0·697 in the preliminary calculation. This excess of pressure may be consumed by reducing the radiator return connexion to $\frac{3}{4}$ in., when its pressure loss is 0·0176 in. per ft.

As the length $l+l_E$ for $\frac{3}{4}$ in. = 19 ft. for the return, the extra pressure consumed would be $(0{\cdot}0176-0{\cdot}0045)\times 19 = 0{\cdot}249$ in., increasing the total loss to $0{\cdot}677+0{\cdot}249 = 0{\cdot}926$, which is sufficiently close to the available pressure.

We have now to calculate the connexions to radiators 1 and 2, on the same circuit; and dealing firstly with radiator 1, we may approximate the values of the pressure increments in proportion to the values of H_1 and H_3 for radiators 1 and 3,

H_1 being 9 ft., the increment is $\frac{9}{32}\times 0{\cdot}705 = 0{\cdot}198$;

H_3 being 0·105 ft., the increment is $\frac{10{\cdot}5}{34}\times 0{\cdot}372 = 0{\cdot}115$;

$ah_1 = 0{\cdot}16\times 22 = 3{\cdot}52$;

$P_1 = 0{\cdot}198+0{\cdot}115+3{\cdot}52 = 3{\cdot}833$ in.

We can take from the previous tabulation on page 29 the values of pressure losses up to the point where the radiator branches join the vertical pipes, namely, $0{\cdot}18+0{\cdot}051+0{\cdot}06+0{\cdot}162 = 0{\cdot}453$. It is therefore necessary to absorb $3{\cdot}833-0{\cdot}453 = 3{\cdot}380$ in. in the radiator flow and return.

$l = 4$ ft.; and for $d = \frac{1}{2}$ in., $E = 6$, then $l_E = 6$ and $l+l_E = 10$ ft. The average loss of pressure required in the connexions is therefore $\frac{3{\cdot}380}{10} = 0{\cdot}338$ in. per ft.

We find on reference to Fig. 1 that this value is not reached. We must remember that the B.T.U. varies approximately as the square root of the pressure, so that at $\frac{0{\cdot}338}{4} = 0{\cdot}085$ in. per ft., the corresponding B.T.U. would be $\frac{10{,}000}{2} = 5{,}000$.

Actually from the nomographic calculator the B.T.U. would be 8,000, so that the pressure loss would only be

$$\left(\frac{10{,}000}{8{,}000}\right)^2\times 0{\cdot}085 = 0{\cdot}133 \text{ in. per ft.}$$

The surplus pressure is thus $(0{\cdot}338-0{\cdot}133)\times 10 = 2{\cdot}05$ in., which must be consumed by a regulating valve or disk in the control valve union. The calculation of regulating disks is referred to elsewhere.

For deciding the size of connexions to radiator 2, in a similar way, the pressure increments become as follows:

H_1 being 19·3 ft., the increment is $\frac{19 \cdot 3}{32} \times 0 \cdot 705 = 0 \cdot 425$;

H_3 being 21 ft., the increment is $\frac{21}{34} \times 0 \cdot 372 = 0 \cdot 23$;

$ah_1 = 0 \cdot 16 \times 12 = 1 \cdot 92$;

$P_1 = 0 \cdot 425 + 0 \cdot 23 + 1 \cdot 92 = 2 \cdot 575$ in.

The pressure already lost to the junction of radiator connexions is that lost to radiator 1, namely, 0·453 in. plus the loss between radiators 1 and 2, namely, 0·044 = 0·497 in.

We must therefore consume 2·575—0·497 = 2·078 in. in the radiator branches, and as we have seen above for radiator 1, only $0 \cdot 133 \times 10 = 1 \cdot 33$ in. is consumed in ½-in. connexions, leaving 2·078—1·33 = 0·748 in., to be consumed by regulation.

It so happens that in this example all radiators and drop pipes are similar in B.T.U. requirements, so that having sized one drop, a little ingenuity enables all other drops and radiator connexions to be similar. In comparing drop II with drop I, it must be apparent that, as all heights are similar, the only difference in available circulating pressure is that due to further cooling of the overhead flow and return pipes between the two drops. We may easily arrange for this additional pressure to be absorbed in the pipes between the two drops, in which case, at the junction of the drops, we have somewhat similar available or unconsumed pressure, enabling all drops to be similar.

The additional pressure may be calculated by assuming a value of T and T_r for the overhead pipes of 0·1 and 0·07 respectively.

The cooling in an extra 13 ft. of flow pipe is then $13 \times 0 \cdot 1 = 1 \cdot 3°$ F. and the increment of pressure, taking H_1 for radiator 6 as 32 ft., is $0 \cdot 004 \times 32 \times 1 \cdot 3 = 0 \cdot 167$ in.

The cooling in 13 ft. of return is $13 \times 0 \cdot 07 = 0 \cdot 91°$ F. and the pressure increment $0 \cdot 003 \times 34 \times 0 \cdot 91 = 0 \cdot 093$ in. The total pressure increment of $0 \cdot 167 + 0 \cdot 093 = 0 \cdot 260$ in. must be used up in 26 ft. of flow and return piping, requiring a loss of $\frac{0 \cdot 260}{26} = 0 \cdot 01$ in. per ft., for which 2-in. pipe would be chosen with an actual loss of 0·004 in. per ft., the total loss in the length being $0 \cdot 004 \times 26 = 0 \cdot 104$ in.

In this instance it will be observed that the assumed value of $T = 0 \cdot 1$ was correct, so that no revision is necessary.

In a similar manner the horizontal mains to drop III having for consistency been sized as 1½ in., to agree with the drop sizes would be found to lose, in a total length of 26 ft., $0 \cdot 0045 \times 26 = 0 \cdot 117$ in.

It will have been realized that the effect of the increased lengths of flow pipe are felt in proportion on all radiators on drops II and III, so that increased surplus pressures will exist as compared with drop I.

We will determine the available and consumed pressures for radiators for which this has not already been done, as follows:

Radiator 4. Available pressure as radiator 1, plus the additional pressure owing to extended mains which approximately is proportionate to the length of flow drop for radiator 4 compared with that for radiator 6, that is, $\frac{9}{32}\times 0{\cdot}260 = 0{\cdot}073$.

$$\begin{aligned}\text{Available pressure} &= 3{\cdot}833 \text{ (that for radiator 1) } +0{\cdot}073 = 3{\cdot}906\\ \text{Consumed pressure} &= 0{\cdot}18+0{\cdot}162+0{\cdot}104+\\ &\quad +0{\cdot}051+0{\cdot}06+1{\cdot}33 = \underline{1{\cdot}887}\\ &\qquad\qquad \text{Surplus} = 2{\cdot}019\end{aligned}$$

Radiator 5. Available pressure as radiator 2, plus the additional pressure owing to longer mains, which approximately in proportion to the flow lengths is $0{\cdot}260\times\frac{19{\cdot}3}{32} = 0{\cdot}156$.

$$\begin{aligned}\text{Available pressure} &= 4{\cdot}093+0{\cdot}156 = 4{\cdot}249\\ \text{Consumed pressure} &= 1{\cdot}887+0{\cdot}022+0{\cdot}022 = \underline{1{\cdot}931}\\ &\qquad \text{Surplus} = 2{\cdot}318\end{aligned}$$

Radiator 6.

$$\begin{aligned}\text{Available pressure} &= 0{\cdot}917 \text{ (that for radiator 3) } +0{\cdot}260 = 1{\cdot}177\\ \text{Consumed pressure} &= 0{\cdot}926 \text{ (that for radiator 3) } +0{\cdot}104 = \underline{1{\cdot}130}\\ &\qquad\qquad \text{Surplus} = 0{\cdot}047\end{aligned}$$

Radiator 7. Available pressure as radiator 4, plus that due to further extension of mains. Cooling in 1½ in. carrying 30,000 B.T.U. was found as $T = 0{\cdot}163$ and $T_r = 0{\cdot}107$.

$$\begin{aligned}\text{Pressure increment for flow} &= 0{\cdot}004\times 9(13\times 0{\cdot}163) = 0{\cdot}076\text{ in.}\\ \text{Pressure increase for return} &= 0{\cdot}003\times 10\tfrac{1}{2}(13\times 0{\cdot}107) = \underline{0{\cdot}044\text{ in.}}\\ &\qquad \text{Total increment} = 0{\cdot}120\text{ in.}\end{aligned}$$

$$\begin{aligned}\text{Available pressure} &= 3{\cdot}906 \text{ (that for radiator 4) } +0{\cdot}120 = 4{\cdot}026\\ \text{Consumed pressure} &= 1{\cdot}887 \text{ (that for radiator 4) } +0{\cdot}117 = \underline{2{\cdot}004}\\ &\qquad\qquad \text{Surplus} = 2{\cdot}022\end{aligned}$$

Radiator 8. Available pressure as radiator 5, plus that due to extended main for the particular length of drop, in proportion to flow lengths, $0{\cdot}120\times\frac{19{\cdot}3}{9} = 0{\cdot}257$.

$$\begin{aligned}\text{Available pressure} &= 4{\cdot}249+0{\cdot}257 = 4{\cdot}506\\ \text{Consumed pressure} &= 1{\cdot}931 \text{ (as radiator 5) } +0{\cdot}117 = \underline{2{\cdot}048}\\ &\qquad\qquad \text{Surplus} = 2{\cdot}458\end{aligned}$$

Radiator 9. Available pressure as radiator 3, plus that due to extended mains, namely, $0 \cdot 120 \times \frac{32}{9} = 0 \cdot 427$.

$$\begin{aligned} \text{Available pressure} &= 0{\cdot}917 \text{ (as radiator 3) } + 0{\cdot}427 = 1{\cdot}344 \\ \text{Consumed pressure} &= 1{\cdot}130 \text{ (as radiator 6) } + 0{\cdot}117 = \underline{1{\cdot}247} \\ &\qquad\qquad\qquad\qquad \text{Surplus} = 0{\cdot}097 \end{aligned}$$

By partial reduction of pipe diameters in suitable places the surplus pressures could all have been consumed, but the various drop pipes would not have had similar dimensions. In the following section, referring to the use of regulating disks, this system will be taken as an example.

The Use of Regulating Disks.

In many instances in practice disks or orifice fittings have been used to check flow through radiators, in cases where it cannot be effected even by cracking the regulating valve. The necessity for such a procedure has often arisen only through systems being installed without care being taken to balance pressures, even in a rough manner.

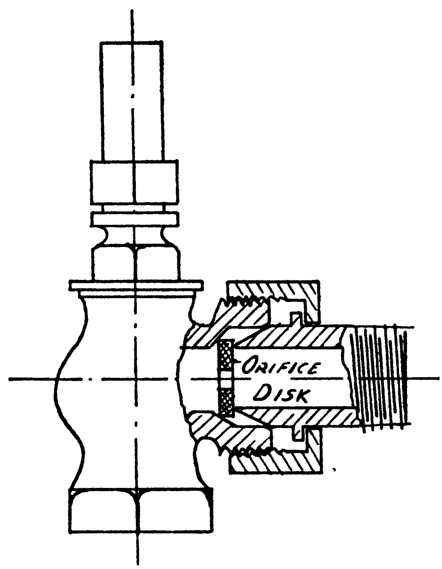


FIG. 8. Regulating disk in union of valve.

Where comparatively accurate calculation is undertaken, as we have seen, large surplus pressures could still exist, and the application of regulating disks which are fixed in the union of the radiator valve, as shown in Fig. 8, can be carried out with advantage.

It has been shown by Rietschel that the value of the single resistance coefficient E for a sudden change of velocity, such as would be obtained in a through-regulating disk, is as follows:

$$(a) \qquad E = 2\left(\frac{V_2}{V_1}\right)^2 - 1,$$

where V_1 represents velocity in pipe leading to the orifice, V_2 represents velocity in orifice.

It may also be stated that, approximately, the velocity of the water in the pipes of a gravity circulation hot-water system is given by the following expression:

(*b*)
$$V = \frac{\text{B.T.U.}}{(T_f - T_r)1200D^2},$$

where V = velocity in ft. per sec.;
B.T.U. = B.T.U. per hour passing through pipe;
$T_f - T_r$ = temperature drop in system;
D = diameter of pipe in inches.

Examining now the values given to the coefficient E, see page 18, we find that approximately we could say that the length equivalent to E for various diameters conformed to the following expression:

(*c*)
$$L = 2ED,$$

where L = equivalent length in feet;
E = value of resistance coefficient;
D = diameter of pipe in inches.

In finding that there is a surplus pressure to be consumed by the use of a disk, we know also the size of the pipe and can easily calculate the additional length of this size of pipe required to consume the surplus pressure.

Substituting expression (*b*) for V_1 in expression (*a*) gives

(*d*)
$$E = 2\left(\frac{V_2}{\dfrac{\text{B.T.U.}}{(T_f - T_r)1200D^2}}\right)^2 - 1.$$

By rearranging expression (*c*) we have $E = L/2D$, and by substituting $L/2D$ for E in expression (*d*) we have

(*e*)
$$\frac{L}{2D} = 2\left(\frac{V_2}{\dfrac{\text{B.T.U.}}{(T_f - T_r)1200D^2}}\right)^2 - 1.$$

What we are to do now is to rearrange and simplify this expression to give the velocity V_2 required through the disk in terms of the remaining quantities, thus:

(*f*)
$$V_2 = \sqrt{\left(\frac{L}{4D} + \frac{1}{2}\right)} \frac{\text{B.T.U.}}{(T_f - T_r)1200D^2}.$$

Leaving this for the moment, we know also that the relation between the areas of pipe and disk must be inversely proportional to the velocities V_1 and V_2, whilst the area A_1 of the pipe is related to the diameter as $A_1 \propto D^2$.

Hence, the diameter in inches of the disk, d, must be

(*g*)
$$d = \sqrt{\left(\frac{V_1}{V_2} D^2\right)}.$$

Substituting the value of V_2 in (f) in this expression, and also the value $\dfrac{\text{B.T.U.}}{(T_f - T_r)1200D^2}$ for V_1, gives

$$(h) \qquad d = \sqrt{\left[\frac{\dfrac{\text{B.T.U.}}{(T_f - T_r)1200D^2} \times D^2}{\sqrt{\left(\dfrac{L}{4D} + \dfrac{1}{2}\right)}\ \dfrac{\text{B.T.U.}}{(T_f - T_r)1200D^2}}\right]}$$

which simplifies to:

$$d = \frac{D}{\sqrt[4]{\left(\dfrac{L}{4D} + \dfrac{1}{2}\right)}}.$$

As in actual tests it has been found that the contraction coefficient for such orifice is 0·85, the diameter must be multiplied by 1/0·85, resulting in the final formulae becoming

$$d = \frac{D}{0{\cdot}85\sqrt[4]{\left(\dfrac{L}{4D} + \dfrac{1}{2}\right)}}.$$

This expression in its final form is simple in application.

Let us refer to the system calculated on pages 27–33, where the surplus pressures of the various radiators were as follows:

Radiator No.	*Surplus pressure*	*Radiator No.*	*Surplus pressure*
1	2·05	6	0·047
2	0·748	7	2·022
3	−0·009	8	2·458
4	2·01	9	0·097
5	2·318		

We must now determine in each case the length of pipe of diameter equal to the radiator connexions which would consume the surplus pressure, and by substituting this and diameter D in the formula, the orifice diameters are obtained. These data are all tabulated below:

Radiator No.	*B.T.U.*	*Diameter of connexions*	*Surplus pressure*	*Equivalent length of pipe*	*Diameter of disk orifice*	*Nearest drill size*
		in.			in.	in.
1	10,000	½	2·05	15·4	0·35	11/32
2	,,	½	0·748	5·6	0·38	7/16
3	,,	¾	−0·009	..	..	..
4	,,	½	2·01	15·4	0·35	11/32
5	,,	½	2·318	17·4	0·34	11/32
6	,,	¾	0·047	..	..	..
7	,,	½	2·022	15·2	0·35	11/32
8	,,	½	2·458	18·5	0·33	11/32
9	,,	¾	0·097	5·5	0·72	23/32

It will be interesting now to consider how best we may decide upon the application of the orifice disks to faulty systems, particularly those where the regulation of valves has not had satisfactory results. If it is possible to survey the whole system, it is simple then to calculate what is likely to be happening and the pressure distribution at different points, the disks being calculated from the data so obtained.

On the other hand, in some cases it is not possible to see more than a small part of the piping, so that precise calculation is impossible.

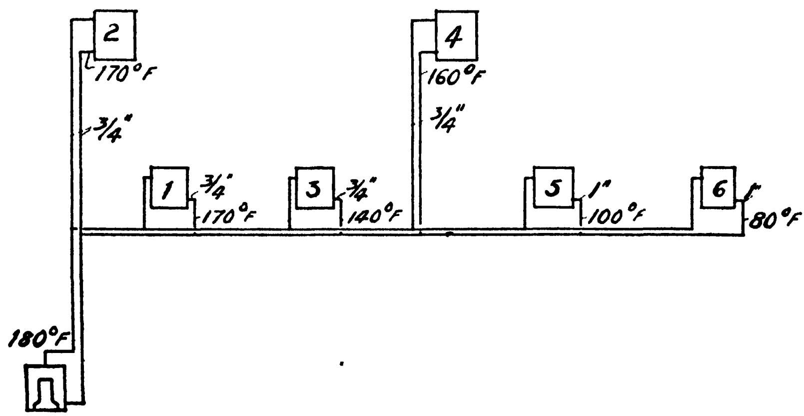


Fig. 9.

It should be remembered that the first indication of faulty circulation and surplus pressures at any particular radiator is the distinctly high return temperature which is found on test. This may be obtained by a surface temperature thermometer.

Similarly, if any radiator is not obtaining the correct quantity of water, this is indicated by a low temperature to the return.

It seems feasible, therefore, that the temperature of the return may be used as an indication of the pressure distribution, which is actually so.

We must of necessity make some assumptions, namely, that the flow temperature at the radiator and the return temperature should be the same for every radiator in the system, which in the case of the return in particular is nearly correct.

Suppose now we had a system as shown in Fig. 9, the temperatures at radiator returns and the main flow and return near the boiler having been measured and found as indicated on the diagram.

With 180° F. flow temperature, the temperature drop through each of the radiators is:

Radiator 1 = 10° F.	Radiator 4 = 20° F.
,, 2 = 10° F.	,, 5 = 80° F.
,, 3 = 40° F.	,, 6 = 100° F.

For the correct amount of water to be passing through each radiator all temperature drops must be $180-140 = 40°$ F.

The number of B.T.U. actually passing through each radiator will be proportional to the measured temperature drop. Again, the total B.T.U. required for each radiator is known from its dimensions and type, so that the total B.T.U. for the system may easily be obtained, in our case 60,000 B.T.U. per hour.

Taking radiator 1, the fact that it has a 10° drop, instead of 40°, indicates that $\frac{40}{10} = 4$ times the correct amount of water is following, this figure for all radiators being:

Radiator 1 = 4	Radiator 4 = 2
,, 2 = 4	,, 5 = 0·5
,, 3 = 1	,, 6 = 0·4

At this stage we should remember that the difference in temperature drops is not all caused by unbalanced pressures alone, for with increased temperature drop the mean temperature of the radiator has fallen, and therefore the radiator emission has decreased. If we assume the required room temperature to be 60° F., the mean temperature for the system being $\frac{180+140}{2} = 160°$ F., there would be 100° F. difference between the radiator mean and the surrounding air. The figures for excess or decrease of water flowing must therefore be multiplied by $100/T_d$, where T_d is the difference for the particular radiator, as follows:

Radiator 1. $4 \times \frac{100}{115} = 3{\cdot}5$	Radiator 4. $2 \times \frac{100}{110} = 1{\cdot}8$
,, 2. $4 \times \frac{100}{115} = 3{\cdot}5$	,, 5. $0{\cdot}5 \times \frac{100}{90} = 0{\cdot}56$
,, 3. $1 \times \frac{100}{100} = 1{\cdot}0$	,, 6. $0{\cdot}4 \times \frac{100}{70} = 0{\cdot}57$

This calculation has assumed radiator emission to vary directly as the temperature difference between radiator and surrounding air, which is sufficiently correct for our purpose.

The figures just derived are now a measure of the quantities of water passing due to unbalanced pressures.

As the relation $P \propto V^2$ is the approximate relation between P the pressure and V the velocity of flow, and the quantity of water flowing is directly proportional to V, it is likely that P, which we will now consider as a measure of the unconsumed pressure, is proportional to the square of the quantity factors, so that P for each radiator is:

Radiator 1 = 12·2	Radiator 4 = 3.3
,, 2 = 12·2	,, 5 = 0·31
,, 3 = 1·0	,, 6 = 0·33

It must be realized, however, that by virtue of the increased or decreased temperature drops in some radiators the available circulating pressure will have been increased, whilst for others it will become less. The available pressure can be stated to be directly proportional to $t_f - t_r$ over a wide range. The figures obtained must, therefore, be adjusted in proportion to the values of $t_f - t_r$ for each radiator, compared with $T_f - T_r$ for the whole system, that is, by $\dfrac{40}{\text{radiator } t_f - t_r}$, the figures becoming

Radiator 1. $12{\cdot}2 \times \frac{40}{10} = 48{\cdot}8$	Radiator 4. $3{\cdot}3 \times \frac{40}{20} = 6{\cdot}6$
,, 2. $12{\cdot}2 \times \frac{40}{10} = 48{\cdot}8$	,, 5. $0{\cdot}31 \times \frac{40}{80} = 0{\cdot}16$
,, 3. $1{\cdot}0 \times \frac{40}{40} = 1{\cdot}0$	,, 6. $0{\cdot}33 \times \frac{40}{100} = 0{\cdot}13$

Radiator 6 is the one with the least favourable conditions, owing to its high temperature drop, and therefore whatever regulation of valves might be required or whatever disks or other resistances are to be inserted for balancing pressures, nothing is required for this radiator. We may therefore consider it is a basis radiator.

It has been stated that the orifice disk coefficient is

$$E = 2\left(\frac{V_2}{V_1}\right)^2 - 1.$$

Where no resistance is inserted in the circuit there is no velocity change and the coefficient becomes $E = 1$, which indicates that one velocity pressure equivalent to the velocity in the particular pipe is required.

Assuming radiator 6 to represent $E = 1$, the adjusted values of E for all radiators would be:

Radiator 1. $E = 375$	Radiator 4. $E = 51$
,, 2. $E = 375$	,, 5. $E = 1{\cdot}23$
,, 3. $E = 7{\cdot}7$	,, 6. $E = 1{\cdot}0$

Rearranging our expression we have, where $V_1 = 1$,

$$V_2 = \sqrt{\left(\frac{E+1}{2}\right)}.$$

As the diameter of the orifice is proportional to the square root of the area, and considering $V_1 = 1$, the proportion of the diameter of the pipe D which the diameter of the orifice d must be, is

$$\frac{d}{D} = \frac{1}{\sqrt[4]{\left(\frac{E+1}{2}\right)}}.$$

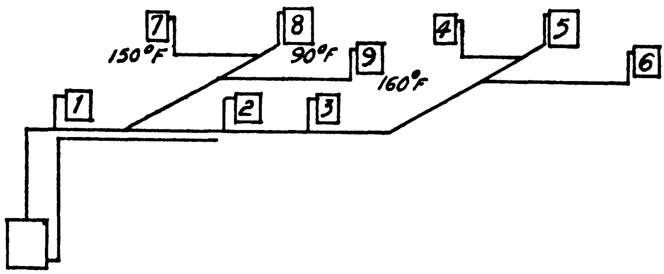


FIG. 10.

From this expression the diameters of disk orifices for the radiators are found as follows:

Radiator No.	*E*	*Diameter of connexions (D)*	$\frac{d}{D}$	*Diameter of size orifice*	*Nearest drill size*
		in.		in.	in.
1	375	$\frac{3}{4}$	0·27	0·202	$\frac{7}{32}$
2	375	$\frac{3}{4}$	0·27	0·202	$\frac{7}{32}$
3	7·7	$\frac{3}{4}$	0·69	0·517	$\frac{17}{32}$
4	51	$\frac{3}{4}$	0·45	0·337	$\frac{11}{32}$
5	1·23	1	0·97	0·97	$\frac{31}{32}$
6	1·0	1	1·0	1·0	..

Disks inserted in the radiator valves with orifices of the calculated diameters would satisfactorily consume all surplus pressures and result in the return temperatures of all radiators becoming substantially similar.

Disk Control for Partial Failures.

Where a system is found to have satisfactory circulation to all radiators but a few, it is usually found that the unbalanced section of the system forms a small sub-circuit. It is not found in practice that one radiator on a system will not circulate unless, without another near to it having a high return temperature, there is a stoppage or an air lock. If on test it is found in a system such as Fig. 10

that all radiators are circulating properly except that radiator 9 has a return temperature of 160° F., radiator 7, 150° F., and radiator 8, 90° F., it is apparent that short circuiting is taking place through radiators 7 and 9 to the detriment of radiator 8.

In such cases calculation of orifice disks can be confined to the sub-circuit for radiators 7, 8, and 9.

If, on the other hand, these radiators had all been found to have return temperatures below the boiler return temperature, it would indicate that other circuits and radiators were receiving too much water and needed checking accordingly, but in such instances it would be found that regulation could be obtained by the use of valves.

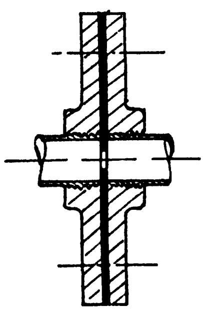


FIG. 11. Regulating disk in pipe run.

Disk Control on One-pipe Systems.

With one-pipe systems of distribution the radiator return temperatures give no indication of the degree of circulation through sub-circuits, but if the temperatures of returns are measured at their junctions, the diameters of disks to be inserted at the junctions, usually between flanges as in Fig. 11, can be readily calculated, each branch circuit becoming analogous to a radiator in the detailed example which we have calculated.

Expressing Disk Proportions as a Function of Return Temperatures.

It will simplify application of the suggested method if we can express the calculations in one formula. To do this we must employ an algebraic value for E in the formula on page 38.

As, however, it is necessary to reduce at one stage of the calculation all quantities in proportion to the minimum one, this is not easy to do, but we can say that E may be obtained by calculating for each radiator the value of

$$\left(\frac{T_f-T_r}{T_f-t_r}\right)^3 \times \left(\frac{0\cdot5(T_f+T_r)-t_a}{0\cdot5(T_f-T_r)-t_a}\right)^2,$$

where T_f = boiler flow temperature,

T_r = boiler return temperature,

t_r = radiator return temperature,

t_a = temperature of air in room,

and then adjusting this value to suit the minimum value, this being 1·0. The value of E so obtained may then be inserted in the disk-ratio formula on page 39.

If now we assume $T_f = 180°$ F., $T_r = 140°$ F., and $t_a = 60°$ F., as in the example calculated, the expression may be simplified to

$$\frac{400{,}000}{(T_f - t_r)^3}.$$

In this form it is easy to calculate values for all radiators, converting to 1 as a basis and substituting in the orifice-ratio formula. It will be found quicker when once T_f, T_r, and t_a are known for any particular system to simplify the formula in the manner suggested, and then to insert the values of t_r for each radiator.

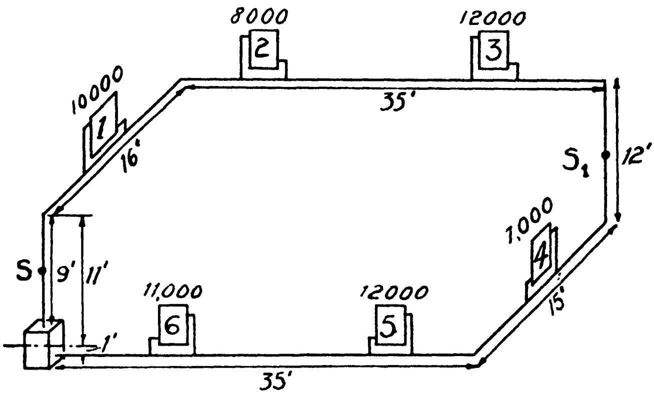


FIG. 12. Ring-main circulation.

Calculation of Ring-main Systems.

The ring-main distributing system is employed in many small installations, and may be calculated in a manner similar to that suggested for other systems of irregular circulation.

Considering Fig. 12, it may be seen that the temperature of the falling column is not only affected by the cooling in pipes but by the cooling of the radiators in the upper part of the circuit.

On similar lines to other systems, we must firstly assume a value for T for flow and return which will be taken as 0·075 and 0·05 respectively, the system being designed for a 30° value of $t_f - t_r$ in view of the high proportion of losses from the pipes, which will be taken as being bare and exposed on the wall.

It is difficult to decide which is flow and which return, but in such cases assume half of the total length of the circuit to be flow.

The total measured circuit length is 122 ft., so that the preliminary calculation for cooling in pipes will be:

$$\begin{aligned} \text{Flow pipe 61 ft.} \times 0{\cdot}075 &= 4{\cdot}6° \text{ F.} \\ \text{Return pipe 61 ft.} \times 0{\cdot}05 &= 3{\cdot}1° \text{ F.} \\ \text{Total pipe cooling} &= 7{\cdot}7° \text{ F.} \end{aligned}$$

As the maximum temperature drop for which the system is being designed is 30° F., the drop of $30-7{\cdot}7 = 22°$ F. approximately must be accounted for by losses from the radiators, the temperature drop in each being proportional to the ratio of its B.T.U. compared with the total radiator B.T.U. of the system.

The total B.T.U. required in the system is 60,000, and the cooling of the water in the main circuit, due to each radiator, will be:

$$\text{Radiator 1.}\quad \frac{10{,}000}{60{,}000}\times 22 = 3{\cdot}7°\ \text{F.}$$

$$\text{,,}\quad 2.\quad \frac{8{,}000}{60{,}000}\times 22 = 2{\cdot}9°\ \text{F.}$$

$$\text{,,}\quad 3.\quad \frac{12{,}000}{60{,}000}\times 22 = 4{\cdot}4°\ \text{F.}$$

$$\text{,,}\quad 4.\quad \frac{7{,}000}{60{,}000}\times 22 = 2{\cdot}6°\ \text{F.}$$

$$\text{,,}\quad 5.\quad \frac{12{,}000}{60{,}000}\times 22 = 4{\cdot}4°\ \text{F.}$$

$$\text{,,}\quad 6.\quad \frac{11{,}000}{60{,}000}\times 22 = 4{\cdot}0°\ \text{F.}$$

We have now to determine the difference of temperature between S and S_1, the midpoints of rising and falling mains.

The temperature at $S = 180-(4{\cdot}5\times 0{\cdot}075) = 179{\cdot}7°$ F.

,, ,, $S_1 = 180-(61{\cdot}5\times 0{\cdot}075)-(3{\cdot}7+2{\cdot}9+4{\cdot}4)$
$= 164{\cdot}4°$ F.

From the formula on page 16, but in this type of installation considering H_1 as a height measured from the centre of the boiler upwards, we have $P_1 = 0{\cdot}004\times 11(179{\cdot}7-164{\cdot}4) = 0{\cdot}67$ in.

There is no value of ah_1 in this type of installation, but we must take into account the fact that the return pipe has to rise to the centre of the boiler and is cooler and heavier than a corresponding length of flow pipe, hence causing a back pressure which must be deducted from the calculated circulating pressure.

The temperature drop in the return pipe from the boiler to the drop at a point level with the centre of the boiler is

$$(51\times 0{\cdot}05)+2{\cdot}6+4{\cdot}4+4{\cdot}0 = 13{\cdot}6°\ \text{F.}$$

The back pressure in the return will be $0{\cdot}003\times 1\times 13{\cdot}6 = 0{\cdot}04$ in., so that the actual available pressure is $0{\cdot}67-0{\cdot}04 = 0{\cdot}63$ in., which must be absorbed in a total measured length of 122 ft., requiring an average loss of $\frac{0{\cdot}63}{122} = 0{\cdot}0052$ in. per ft.

By reference to Fig. 1, the requisite size for a total of 60,000 B.T.U. on 30° drop is $2\frac{1}{2}$ in. with an actual loss of 0·0023 in. per ft., whilst by reference to Fig. 3 the true temperature drop $T = 0{\cdot}09$, and by previous explanation $T_r = 0{\cdot}66 \times 0{\cdot}09 = 0{\cdot}06$.

The cooling in pipes is thus more than that assumed, so that the cooling from S to S_1 will be increased, and also the available pressure, resulting in a lower value of $t_f - t_r$ than required without in any way leading to faulty operation. The size of pipe could well have been chosen as 2 in. in view of the actual pressure loss in this size being close to that available, when the value of $t_f - t_r$ would have been very slightly increased again without detriment to the efficiency of the apparatus. The calculation of exact working values of $t_f - t_r$ is not necessary.

Arrangements for Increasing Circulating Pressure.

With some systems the arrangement of piping must of necessity be such as does not result in more than a fractional value for the available circulating pressure. Apart from accelerated systems there are other simple methods of producing higher available pressures.

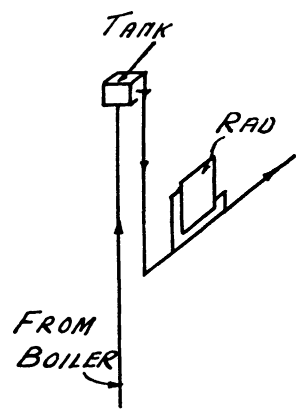


FIG. 13. Accelerating device.

The system in Fig. 12 could be modified in such a manner that the flow pipe from the boiler is taken direct to the expansion tank, resulting in increased lengths of rising flow and dropping pipes and subsequent increase in the total difference of weight between the two columns, and also in a further loss of heat from the surface of the tank, which gives a higher value to the temperature difference between S and S_1.

The suggested arrangement is that in Fig. 13. As the size of the tank is known for any particular installation, it is possible to calculate the loss of heat from the surface and find the effect on temperature drop. For a 20-gallon capacity tank which would be used on the system in question, the surface area of the tank would be 8 sq. ft., and taking a value of K, the emission coefficient, as 2·0, and a temperature of water 180° F. and surrounding air 40° F., the heat loss from the tank would be

$$8 \times 2 \times (180 - 40) = 2{,}240 \text{ B.T.U. per hour.}$$

The cooling-conditions are now entirely different, for the travel is also increased by 20 ft. and the approximate cooling in the flow pipe is $81 \times 0{\cdot}075 = 6{\cdot}1°$ F., bringing the total approximate pipe cooling to $6 \cdot 1+3 \cdot 1 = 9 \cdot 2$, and leaving $30-9 \cdot 2 = 20 \cdot 8°$ F. to be lost by emission from the tank and the radiators. The total B.T.U. is now $60{,}000+2{,}240 = 62{,}240$, and the tank accounts for

$$\frac{2{,}240}{62{,}240} \times 20 \cdot 8 = 0 \cdot 7° \text{ F.}$$

cooling, and similarly the radiators for

Radiator 1.	3·3° F.	Radiator 4.	2·3° F.
,, 2.	2·7° F.	,, 5.	4·0° F.
,, 3.	4·0° F.	,, 6.	3·8° F.

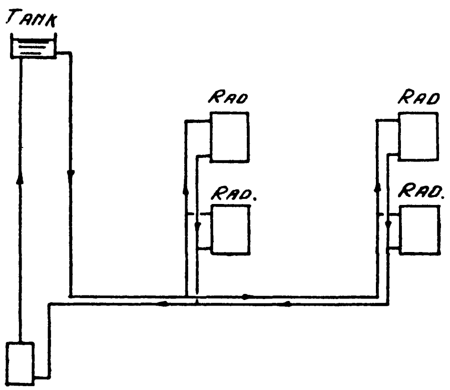


FIG. 14. Accelerating device.

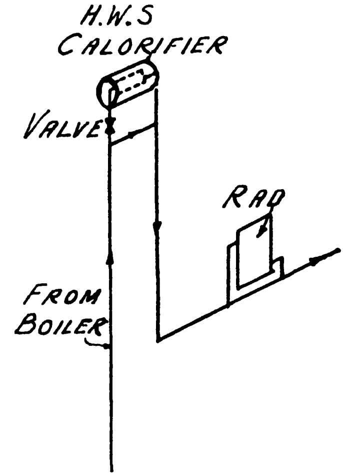


FIG. 15. Accelerated system.

The temperature at S_1 now becomes

$$180-(81 \cdot 5 \times 0 \cdot 075)-(0 \cdot 7+3 \cdot 3+2 \cdot 7+4 \cdot 0) = 163 \cdot 2° \text{ F.},$$

so that $P_1 = 0 \cdot 004 \times 11(179 \cdot 7-163 \cdot 2) = 0 \cdot 73$ in. We can accept the back pressure due to lifting the return to the centre of the boiler as the same, and the circulating pressure becomes $0 \cdot 73-0 \cdot 04 = 0 \cdot 69$.

But, in addition to this, we have a further pressure due to the difference in weight of the flow pipe to and from the tank, which is ascertained as follows.

The difference of temperature between the midpoint of these pipes will be $(10 \times 0 \cdot 075)+0 \cdot 6 = 1 \cdot 35°$ F.

The increment of circulating pressure is thus

$$0 \cdot 004 \times 10 \times 1 \cdot 35 = 0 \cdot 054 \text{ in.}$$

The actual available pressure is thus $0 \cdot 69+0 \cdot 054 = 0 \cdot 744$ in., compared with 0·63 in. for the arrangement without the flow pipe running through the expansion tank, an increase of over 17 per cent. in circulating pressure.

Suppose, now, that hot-water supply services are required for the same building, and that instead of the expansion tank we have an indirect cylinder which would be requiring a minimum of 10,000 B.T.U. per hour. In this case the circulating pressure would be increased even more, resulting in a smaller diameter of the ring main. This arrangement is illustrated in Fig. 14. A thermostat may be arranged to shut the valve in the branch flow to the indirect cylinder, when the temperature of the storage water is sufficiently high; circulation to the heating system then takes place through the by-pass.

The application of the heated expansion tank is often of use for improving badly designed systems, particularly where the tank is situated several floors above the boiler level. Indeed, it may be applied to all types of circulation, even of the two-pipe underfeed type, as illustrated in Fig. 15.

Simplification of Pipe-sizing Practice.

There are several methods which may be employed for eliminating tedious calculations, which at the same time have the merit of being accurate, and the following summarizes the procedure to follow with the various types of hot-water heating systems.

(1) *Irregular Circulations as Figs. 5 and 6.*

Determine B.T.U. required in all pipes, these being the actual values for balancing heat losses, ignoring losses from pipes. Size in accordance with the examples given with values of 30° or 40° F. for temperature drop.

(2) *Ring-main Systems as Fig. 12.*

Proceed as above, but with a maximum of 30° F. temperature drop, ignoring heat loss from pipes.

(3) *One-pipe Drop Systems as Fig. 16.*

The available circulating pressure should be calculated by assuming values of temperature drop in pipes, proportioning the further drop due to radiators, and calculating the mean temperature of rising and falling columns from the sum of the products of the particular length multiplied by its arithmetical mean temperature, the whole being divided by the sum of the lengths. The difference between these two values is used to find the correct value of circulating pressure from Fig. 2, which is then multiplied by the total length of the column. This may be expressed as follows:

$$T_d = \frac{\{[h_1 \times 0{\cdot}5(t-t_1)]+[h_7 \times 0{\cdot}5(t_m-t_8)]\}- -\{[h_2 \times 0{\cdot}5(t_2-t_3)]+\ldots+[h_6 \times 0{\cdot}5(t_6-t_7)]\}}{H}.$$

The references to temperature t and heights h and H may be seen from Fig. 16.

The circulating pressure is determined by multiplying the value for T_d as found on Fig. 2 by H, and pipes are sized in accordance with the general principles outlined elsewhere.

In all cases the index circulation is considered to be that having the least average value of circulating pressure per foot of travel.

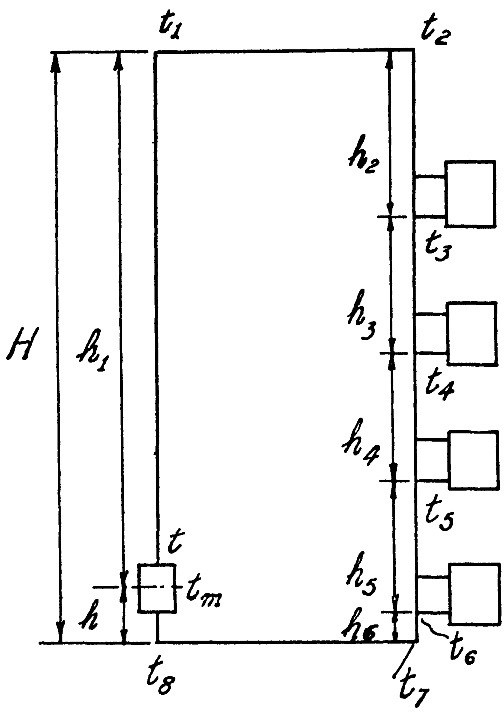


FIG. 16. One-pipe drop system.

The maximum value for $t_f - t_r$ should be taken as 40° F. It should be noted that this method takes into the calculation the increased circulating pressure due to cooling of the overhead flow pipe, which is not accomplished by other methods. The heat losses from pipes are to be ignored.

(4) *Two-pipe Underfeed Systems.*

Circulating pressure P to be calculated from $P = ah_1$, where a = available pressure for the temperature drop $t_f - t_r$ decided on, and h_1 = distance between centre of boiler and radiator. Sizes to be determined either from average friction losses per foot, or try the diminishing pressure loss system where thought desirable. This latter will be described elsewhere. Maximum value of $t_f - t_r$ to be 40° F. and pipe-heat losses to be ignored.

(5) *Two-pipe Systems with Flow and Return at High Level.*

Proceed in accordance with the calculations for the typical example in Fig. 7 with maximum value of $t_f - t_r = 40°$ F., and pipe-heat losses to be ignored.

(6) *Two-pipe Drop Systems.*

Proceed substantially as for underfeed systems with maximum value of $t_f - t_r = 50°$ F., ignoring pipe-heat losses. The higher temperature drop suggested is balanced by the increased pressure due to cooling in the overhead flow pipe, which is not allowed for in calculations.

Branch Connexions to Radiators on One-pipe Systems.

The author does not agree with the generally accepted statement that the circulation through radiator connexions on a one-pipe system is entirely independent of the main circulation from the boiler. *Every radiator on a one-pipe system is influenced by its height above the boiler just as in any other system.*

The accurate expression for the circulating pressure available for the connexions is as follows:

$$p = \frac{H \times 0{\cdot}004(t_f - t_r) + \left(\frac{W - W_1}{W}\right)^2 P_1 l_1}{L},$$

where H is distance in feet from centre of radiator to centre of horizontal pipe, or, if the radiator is on a drop system, the distance from the centre of the radiator to the point at which the radiator return joins the drop pipe.

If both flow and return connexions are at the bottom of the radiator, H is to be measured from a point a quarter of the way up, between top and bottom nipples.

p = permissible loss of pressure per foot run of equivalent radiator connexions in inches of water column.

t_f = flow temperature of radiator.

t_r = return temperature of radiator.

W = B.T.U. per hour passing through main pipe.

W_1 = „ „ „ radiator.

P_1 = pressure loss of main pipe in inches of water column per foot run.

l_1 = distance in feet between radiator tees.

L = total equivalent length of the connexions to the radiator in feet.

Apart from the circulating pressure due to the height H in Fig. 17, which is a simple one-pipe system, there is that due to H_1, measured from the centre of the boiler to the horizontal main. For a balanced circulation, the unconsumed pressure must be the same round each circuit. In this case the whole of the pressure available for a height H_1 must be consumed in $ABCDEFGH$, whilst the pressure available for a height of H_1+H must be consumed in $ABCIJDEFGH$. But, of the pressure available for H_1+H, the majority of that available for a height H_1 is consumed in ABC and $DEFGH$. That is to say, there is a pressure difference between CD equal to the loss of pressure in CD, in addition to the pressure difference due to the height H.

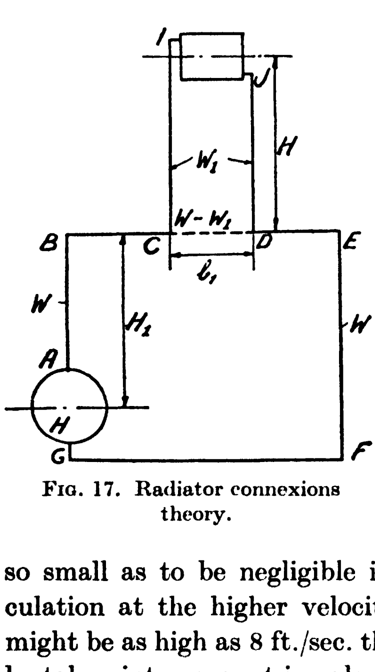


FIG. 17. Radiator connexions theory.

The pipe CD does not, however, carry the whole of the B.T.U. for the system, but $W-W_1$. The calculated pressure loss in the main circuit invariably neglects this factor, so that the actual increase of pressure available for the radiator connexions is less than $p_1 l_1$, and is, in effect, $\left(\frac{W-W_1}{W}\right)^2 p_1 l_1$.

For the velocities obtained in a gravity circulation this pressure is so small as to be negligible in practice, although for pumped circulation at the higher velocities where the velocities in the pipes might be as high as 8 ft./sec. this pressure is considerable, and should be taken into account in calculating the pipe sizes.

Although a system may be calculated for a temperature drop of 40° F., the radiator connexions may be sized upon any other drop of temperature. If it is thought desirable, the whole of the radiators may be calculated for the same mean temperature, thus facilitating the calculation of heating surface in the preliminary stages. This has the effect of varying the drops of temperature through the various radiators, there being possibly 60° drop through the first radiator from the boiler and 10° drop through the last.

Another method is to size the whole of the radiator connexions on the same drop of temperature as the main circulation, the mean temperature of the radiators decreasing with the distance from the boiler. The author prefers this method, as the first sometimes results in radiator connexions being larger than the main.

Method of Determining Sizes of Connexions to Radiators on One-Pipe Systems from Nomograph. (Fig. 18.)

It will be observed that this nomograph has the following scales:

A, diameter of pipes.
B, an ungraduated scale.
C, the value of H, calculated as Fig. 17.
D, B.T.U. per hour required at radiator.
E, temperature drop through radiator, degrees Fahr.

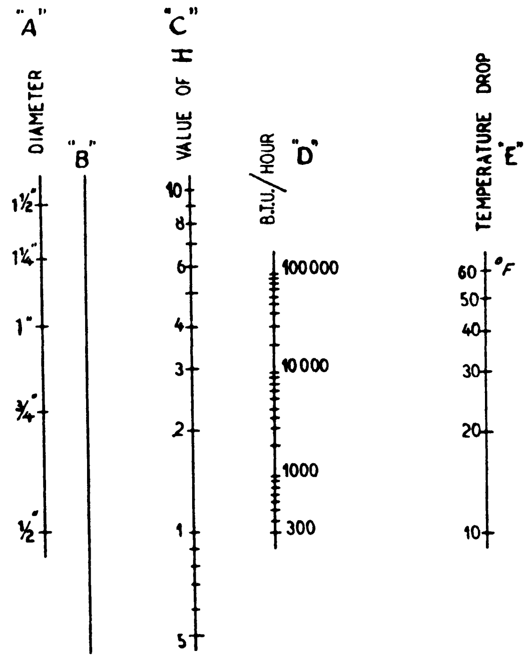


FIG. 18.

EXAMPLE 1. What size connexions would be required for a radiator on a one-pipe system, if the value of *H* is 4 ft., the B.T.U. required 10,000, and the approximate temperature drop 20° F.?

Place a straight edge through 10,000 B.T.U. on scale D and 20° F. on scale E to intersect the ungraduated scale B. Another straight line through this intersection and the value of $H = 4$ ft. on scale C gives on scale A a diameter of 1 in. for the connexions.

EXAMPLE 2. A radiator on a one-pipe system has a value of H = 6 ft. and requires 8,000 B.T.U., whilst it has ¾-in. connexions. What temperature drop would be obtained?

A line through ¾ in. on scale A and 6 ft. on C. intersects the ungraduated scale B, and a further straight line through this intersection and 8,000 B.T.U. on scale D gives the temperature drop on scale E as 28° F. approximately.

Whichever method is used, the installation will be quite successful as long as the correct B.T.U. are taken off at each point.

The nomograph in Fig. 18 enables the connexions to be determined for any desired value of temperature drop and H_1, whilst, similarly, knowing the size of the pipe, the B.T.U., and H_1, the temperature drop may be determined to facilitate calculation of the mean temperature of the radiator for deciding upon the necessary heating surface.

Tongue Tees and Ejector Fittings.

In a well-designed gravity hot-water heating system no tongue tees or similar fittings need be employed, but as no other work has dealt with the extra pressure derived from such fittings it is thought worth considering here.

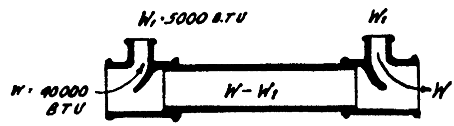


FIG. 19. Tongue tee branches.

The use of these fittings is invariably confined to one-pipe systems. It has been explained that there is an extra circulating pressure available for the branch connexions other than that due to the position of the radiator relative to the main pipe, equal to the pressure loss between the tees in the main. If tongue tees are used as in Fig. 19, the orifice through which the water must pass is reduced due to the obstruction offered by the tongues or lips projecting into the pipe. The increase of velocity over the tongues causes a pressure loss equal to the difference in velocity pressures through the fittings and pipes respectively.

Taking, for example, a radiator giving 5,000 B.T.U. per hour, the main pipe, passing 40,000 B.T.U. per hour, with a value of $t_f - t_r = 40°$ F., being 2-in. diameter, the velocity through the pipe will be, as explained,

$$V = \frac{\text{B.T.U.}}{(t_f - t_r)1200D^2} = \frac{40{,}000 - 5{,}000}{40 \times 1200 \times 2^2} = 0{\cdot}182 \text{ ft./sec.}$$

If we take the tongues to constrict half the area of the pipe, the velocity through the fitting would be $2 \times 0{\cdot}182 = 0{\cdot}364$ ft./sec.

The pressure loss is actually the difference in velocity pressure at the two velocities, the velocity pressure being V.P. $= \frac{V^2}{2g}$, so that the difference in V.P. for the two velocities, one being double the other, will be $3V^2/2g$, this giving the result in feet of water column; for inches of water column the result is $36V^2/2g$. By inserting the previously given expression for velocity we may say that

$$P = 36\frac{\left\{\dfrac{\text{B.T.U.}}{(t_f-t_r)1200D^2}\right\}^2}{2g}.$$

For 40° value of t_f-t_r and inserting $2g = 64{\cdot}4$, whilst the expression is multiplied by 2 to account for the two tongue tees used for the radiator, this simplifies to

$$P = 1{\cdot}12\left(\frac{\text{B.T.U.}}{48{,}000D^2}\right)^2.$$

The expression can be similarly simplified for other values of t_f-t_r if required.

For the figures taken as an example the additional pressure due to the tongue tees would be $1{\cdot}12\left(\dfrac{35{,}000}{48{,}000\times 4}\right)^2 = 0{\cdot}037$ in. of water.

It will be appreciated from this figure that to insert tongue tees for all radiators on a ring-main system, for instance, would give a greatly increased pressure loss in the main, resulting in a greater value of t_f-t_r than that for which the system is calculated.

The only value of tongue tees or ejector fittings is in accelerated ring-main or one-pipe systems, where their use will result in radiator-branch circulations being small—at the expense of an increased head on the pumps.

Calculations for tongue tees having other ratios of construction may be carried out in a similar manner.

Pipe Sizing for Pump Accelerated Systems.

Although most of the remarks referring to gravity hot-water heating systems apply equally well to pump-accelerated circulation, there are several fundamental differences which need considering.

The accelerating pump provides a definite pressure differential on the system expressed in either feet or inches of water column, but at the same time the circulating pressures due to temperature differences still exist, and in balancing pressures for the various circuits must be taken into account. It is not necessary, however, to determine these pressures with strict accuracy, for compared with the head provided by the pump they are only small. If the gravity circulating pressure is ignored, particularly in the case of a building with several floors, the result is to cause partial short-circuiting round the sub-circuits serving radiators on the upper floors, even if the system has been calculated accurately upon the pump differential.

In sizing the pipes for an accelerated system, there are several methods by which the problem may be approached. The pipes may be sized upon equal friction loss per foot run for all pipes, determined from a fixed maximum pump head and the measured travel, or the velocities in the pipes may be decided upon, and the sizes and friction losses calculated from this point. Apart from any consideration as to power consumption, there is no reason why velocities as high as 8 to 10 ft./sec. should not be employed in main distributing pipes outside buildings, dropping to a maximum of 4 ft./sec. inside the building, to avoid noise.

Similarly, considering the desirable maximum pump head, there is no actual limit, each installation being decided purely on its merits. Generally, however, for systems in a compact building, in which the heating load does not exceed 2,000,000 B.T.U. per hour, the pump head should not be greater than 15 ft., that is, 180 in. of water column; but for systems where the heating load is required by blocks of considerable size situated some distance apart, 40 ft. or even more is not unusual.

With regard to the temperature drop in the system for pumped circulation, where it is intended that circulation should never take place unless the pump were running, up to 40° F. may be allowed. There is no justification for having the temperature drop less than 20° F. with pumped circulation.

Whilst a substantially similar formula is applicable for the flow of hot water through pipes at the higher velocities employed for accelerated systems, some small alterations are required in the method of application for pipe sizing. Although it is feasible to ignore pipe-heat losses in gravity circulation, with the assurance that doing so will give satisfactory circulation, with a pumped system it is necessary to make allowances for pipe-heat losses. This is essential owing to the pump head or pressure being a fixed amount irrespective of small variations in the temperature drop in the system. But if pipe-heat losses are ignored, it will result only in a slightly increased total temperature drop in the system.

To avoid, therefore, the unnecessary addition of percentages to all B.T.U. required to compensate pipe-heat losses, a slightly lower temperature drop may be employed for calculations than that allowable in practice, dependent upon the estimated proportion of pipe-heat losses. Experience alone gives a true understanding of this proportion, but generally with pumped systems the proportion of pipe-heat losses to the heat requirements is 10–25 per cent. If, therefore, for a particular system, pipe losses were assumed as 28 per cent. of the required B.T.U., and the maximum temperature drop required was 30° F., the system should be designed for $\frac{100}{125} \times 30 = 24°$ drop, the B.T.U. loads for all pipes being obtained from the actual heat loss requirements at the various points, without addition for pipe-heat losses.

To facilitate problems requiring pipe sizing of pump-accelerated systems, the nomographic calculator in Fig. 20, substantially similar to that for gravity circulation, has been designed, extended to higher values in all scales, and with a graduated temperature-drop scale. The use of this calculator is explained on page 54.

The Application of Pumps to Gravity Circulations.

There are many instances of pumps being applied to gravity circulations which are not satisfactory due to bad design or applied to the system with a view to providing a means of rapidly heating all radiators when required.

Supposing a system to have been designed to work on 40° F. drop of temperature, and that it is desired to apply a pump for boosting, it is not easy to decide what the pump capacity and head shall be. For gravity circulation, however, the loss of pressure in the index circulation, *P*, is known as a basis of design, so that, according to the desired temperature drop with the pumped circulation, the frictional head or resistance will increase.

It is known that the frictional loss will increase approximately as the square of the velocity through the pipes, so that if, for instance, the temperature drop required is 20° F., the head required must be $\left(\frac{40}{20}\right)^2 = 4$ times as great as that for the gravity conditions of the greatest pressure-losing circuit. If the circuits have been so calculated as to have balanced pressures for gravity conditions, the application of a similar artificial head to all circuits will not result in the balanced conditions remaining similar.

On the other hand, where a system is designed for pumped circulation, even on a low temperature drop, and balanced for these conditions, if it is run on gravity circulation, distribution is also not satisfactory.

In cases where a pump is being applied to an unknown system, thought to be giving bad circulation, owing to a large observed temperature drop between flow and return, the only satisfactory way of applying the pump is to decide upon the total required B.T.U. from radiators of known sizes, the calculated available pressure for the

Method of Using Nomographic Calculator for Finding Sizes of Pipes for Accelerated Hot-Water Circulation. (*Fig.* 20.)

The method of employing this calculator is substantially similar to that for the gravity circulating calculator, Fig. 1.

It should be noted that the figures on the B.T.U. scale A represent *thousands of B.T.U.*

The approximate velocity of water in the pipes may be found from the following expression:

$$V = \frac{B.T.U.}{(T_f - T_r)\,k},$$

where V = velocity in feet per second,

$(T_f - T_r)$ = temperature drop in system,

k = coefficient varying with the pipe diameter, as follows:

Diameter	k
½ in.	300
¾ ,,	675
1 ,,	1,200
1¼ ,,	1,875
1½ ,,	2,700
2 ,,	4,800
2½ ,,	7,500
3 ,,	10,800
4 ,,	19,200
5 ,,	30,000
6 ,,	42,200
7 ,,	59,000
8 ,,	77,000
9 ,,	97,500
10 ,,	120,000
12 ,,	173,000
14 ,,	235,000
15 ,,	270,000
16 ,,	310,000
18 ,,	400,000
24 ,,	700,000
27 ,,	900,000
30 ,,	1,100,000
36 ,,	1,600,000
42 ,,	2,100,000

For example, a 6-in. pipe carrying 4,000,000 B.T.U. per hour on a 40° drop would have a velocity of

$$\frac{4{,}000{,}000}{40 \times 42{,}200} = 2{\cdot}4 \text{ ft. per second.}$$

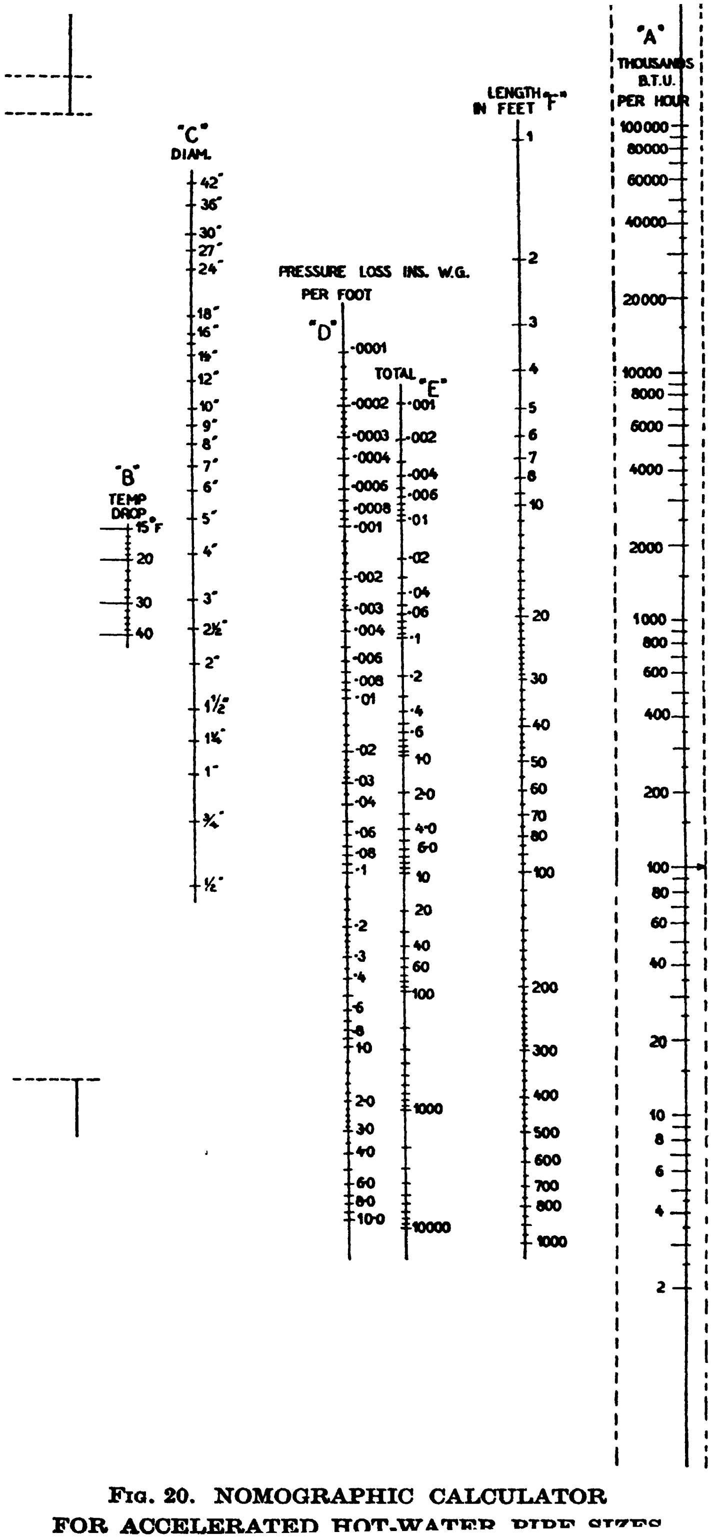


Fig. 20. NOMOGRAPHIC CALCULATOR FOR ACCELERATED HOT-WATER PIPE SIZES

highest radiator for the observed temperature conditions, and then proceed as follows.

Suppose that the observed temperatures are 180° F. for the flow and 118° F. for the return, whilst the calculated radiator B.T.U. is 800,000.

The centre of the highest radiator is 60 ft. above the boiler centre, so that the circulating pressure is approximately

$$0{\cdot}004 \times 60 \times (180-118) = 14{\cdot}9 \text{ in. of water.}$$

If the pump is required to give 20° F. difference of temperature it must provide a head of

$$\left(\frac{62}{20}\right)^2 \times 14{\cdot}9 = 144 \text{ in. approx.}$$

The pump must therefore deliver

$$\frac{800{,}000}{20 \times 10 \times 60} = 67 \text{ gallons per minute against 144 in. head, or 12 ft.}$$

Having provided the maximum pressure required for the circuit already having the highest pressure loss for the desired increased quantity of water, we are now faced with large surplus pressures on lower radiators.

The surplus pressures can then be balanced by valve regulation and the application of orifice disks, as previously described.

Diminishing Pressure-loss System.

The use of the *reversed return system*, illustrated in Fig. 21, is prevalent in connexion with hospital systems and similar installations to scattered blocks of buildings, but *there is no justification whatever for applying this system* as it is costly to install and similar results may be obtained by an intelligent application of pipe-sizing principles, in the form of the diminishing pressure-loss system of sizing, mentioned in no other work.

It will be appreciated that the object of the reversed return system, particularly for serving identical blocks, is to secure identical pressure losses to the branch serving each particular block, to facilitate all blocks being similarly sized as far as internal equipment is concerned. This system does not even accomplish what it claims, as calculation of pressure losses always proves. It certainly results in equal travel to all blocks, but the pressure losses in all pipe lengths are not necessarily similar owing to the need for working to standard pipe diameters.

It may be said that the better-known two-pipe distribution in Fig. 23 would give similar results to the reversed return systems, provided surplus pressures were consumed by reduction in branch-pipe sizes to each block. This procedure is not to be advocated, for, as will be seen later, for large institutions, employing water softeners, hot-water supply services may be taken from common main distributing pipes serving also the heating system, only branching to separate systems within the blocks. This being so, it is a disadvantage to have branch mains to any block reduced to small diameter, in view of the effect on hot-water draw-off.

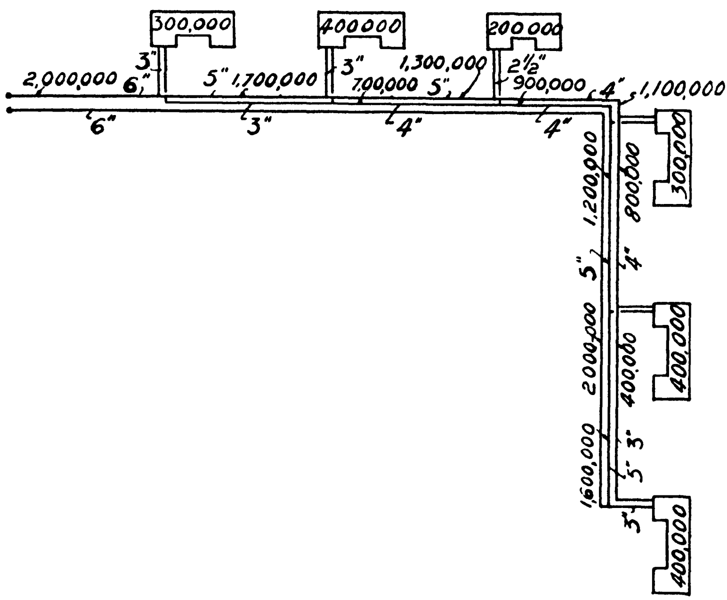


FIG. 21. Reversed return pipe-sizing.

Referring to Fig. 22 it will be seen that the distribution is by the well-known two-pipe system.

Let T = maximum travel in ft.,

P = average pressure loss per ft.

If the total travel for the first branch circuit is x ft., the average pressure loss for the whole circuit, including the common main, must be $\frac{T}{x} \times P$.

But part of this circuit is common to the main, perhaps y ft., and the total loss in $y = \left(\frac{T}{x} \times P\right) \times y$.

For the next complete circuit, of total length x_2, the necessary average loss must be $\frac{T}{x_2} \times P$. But the total loss in y_1 is common, that

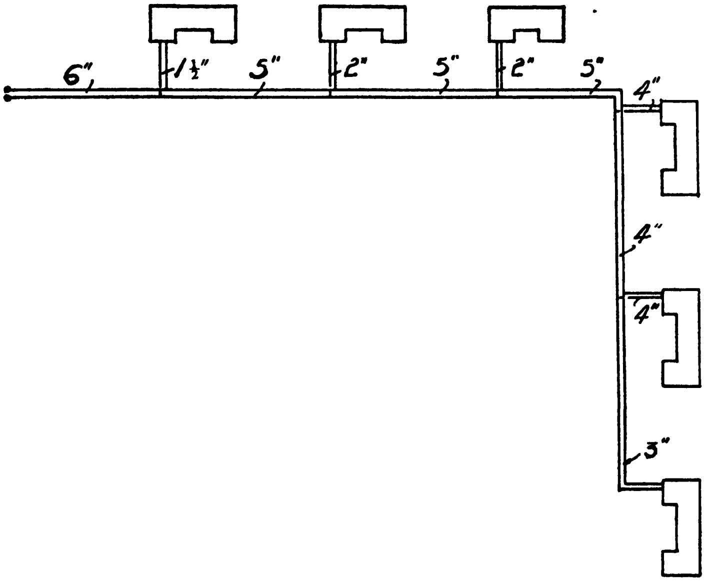


FIG. 22. Diminishing pressure loss pipe-sizing.

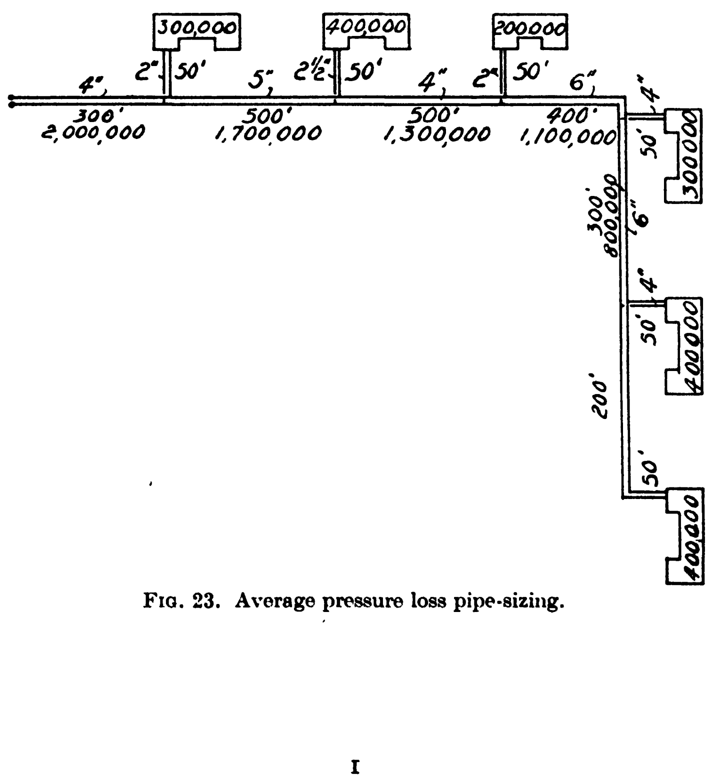


FIG. 23. Average pressure loss pipe-sizing.

is, $\left(\frac{T}{x}\times P\right)\times y$, so that there only remains for the length x_2-y_1, $(T\times P)-\left\{\left(\frac{T}{x}\times P\right)y\right\}$.

The average loss which the length x_2-y_1 can have is therefore

$$\frac{(T\times P)-\left\{\left(\frac{T}{x}\times P\right)y\right\}}{x_2-y_1}.$$

And for the final complete circuit of length, $T-(y_1+y_2)$, the average available pressure is

$$\frac{(T\times P)-\left\{\left(\frac{T}{x}\times P\right)y\right\}-\left[\frac{(T\times P)-\left\{\left(\frac{T}{x}\times P\right)y\right\}}{x_2-y_1}\times y_2\right]}{T-(y_1+y_2)}.$$

Although these expressions demonstrate the pressure losses for various pipes, with the diminishing pressure-loss system of pipe sizing, they cannot conveniently be employed in this form, but if

P_x = average pressure loss on which any branch and its preceding section of main should be sized,

P = total pressure to be lost,

p_x = pressure loss per ft. in all preceding mains, having length l_x,

L_x = length of any branch and its preceding length of main,

then $$P_x = \frac{P-p_x l_x}{L_x}.$$

The diminishing pressure-loss system is perfectly easy of practical application, as the example given will show.

We will assume that the total head to be consumed in the mains of the system in Fig. 22 is 25 ft., that is, 300 in. of water column.

Considering firstly the circulation to block 1, the total of 300 in. must be consumed in the pipes leading from the calorifiers to the entrance to this block—just as it must for any other block—if balanced conditions are to be obtained.

The total travel to block 1 is 700 ft., so that the available pressure loss is $\frac{300}{700} = 0{\cdot}43$ in. per ft.

Referring to Fig. 20, with a 20° drop of temperature a 6-in. main pipe, with an actual loss of 0·23 in. per ft., is required, the total consumed in the 600 ft. of main being $600\times 0{\cdot}23 = 138$ in., having

$300-138 = 162$ in. available for the next length of main and its next branch.

The permissible average loss for this will therefore be $\frac{162}{1,100} = 0{\cdot}147$ in. per ft., requiring a 5-in. main with an actual loss of 0·052 in. per ft., and a total loss in the main of $1,000 \times 0{\cdot}052 = 52$ in., leaving $162-52 = 110$ in. available for the next length of main and the branch to block 3.

Similarly, the mains leading to other blocks are sized and the pressure losses calculated, the necessary sizes being those indicated on Fig. 22.

It is now necessary to size the branches to the various blocks. Dealing with block 1, we found the actual loss in the main up to the junction of the branch to block 1 to be 138 in., and 162 in. must be consumed in the branch, having a length of 100 ft., that is, $\frac{162}{100} = 1{\cdot}62$ in. per ft. must be consumed, requiring a pipe $1\frac{1}{2}$ in. diameter.

Similarly, for block 2, the available pressure at the junction is 110 in. to be consumed in 100 ft., requiring $\frac{110}{100} = 1{\cdot}10$ in. per ft. pressure loss, calling for 2-in. pipe. The sizes of other branches calculated in the same manner are shown in Fig. 22.

Considering the sizes obtained, it is observable that the mains and branches tend to increase in size towards the extremity of the system, the connexions to the last block actually being 5 in. compared with the section of main carrying the whole load which is 4 in.

For comparison, the same system has been sized on average pressure loss per foot of travel, balancing being obtained by consuming surplus pressures in branches to the blocks, the sizes obtained being as given on Fig. 23, and similarly the pipes are sized for a reversed return system, on Fig. 21.

It will be noticed on Fig. 23 how small some branches would become, and on Fig. 21 the fact that a 6-in. diameter return pipe must be taken from the extreme block back to the central station, both systems showing disadvantages.

We will endeavour to assess the costs of main pipes for the various systems.

For this purpose we will ignore branch pipes leading to the blocks. The estimated costs of tube, supports, insulation, and labour costs are as follows:

Average Pressure-loss System.

			£	s.	d.
600 ft. — 6-in. steam tube	@ 2s. 6d.	=	75	0	0
2,800 „ — 5 „ „ „	@ 1s. 11d.	=	269	6	8
600 „ — 4 „ „ „	@ 1s. 5d.	=	42	10	0
500 „ — 3 „ „ „	@ 1s.	=	25	0	0
Supports.		=	18	0	0
600 ft. — 6-in. insulation	@ 2s. 2d.	=	65	0	0
2,800 „ — 5 „ „	@ 2s.	=	280	0	0
600 „ — 4 „ „	@ 1s. 4d.	=	40	0	0
500 „ — 3 „ „	@ 1s. 1d.	=	27	1	8
Labour cost.		=	160	0	0
	Total	=	1,001	18	4

Reversed Return System.

			£	s.	d.
2,500 ft. — 6-in. steam tube	@ 2s. 6d.	=	312	10	0
1,500 „ — 5 „ „ „	@ 1s. 11d.	=	143	15	0
1,600 „ — 4 „ „ „	@ 1s. 5d.	=	113	6	8
700 „ — 3 „ „ „	@ 1s.	=	35	0	0
Supports.		=	25	0	0
2,500 ft. — 6-in. insulation	@ 2s. 2d.	=	270	16	8
1,500 „ — 5 „ „	@ 2s.	=	150	0	0
1,600 „ — 4 „ „	@ 1s. 4d.	=	106	13	4
700 „ — 3 „ „	@ 1s. 1d.	=	37	18	4
Labour cost.		=	220	0	0
	Total	=	1,415	0	0

Diminishing Pressure-loss System.

			£	s.	d.
1,400 ft. — 6-in. steam tube	@ 2s. 6d.	=	175	0	0
1,500 „ — 5 „ „ „	@ 1s. 11d.	=	143	15	0
1,600 „ — 4 „ „ „	@ 1s. 5d.	=	113	6	8
Supports.		=	18	0	0
1,400 „ — 6-in. insulation	@ 2s. 2d.	=	151	13	4
1,500 „ — 5 „ „	@ 2s.	=	150	0	0
1,600 „ — 4 „ „	@ 1s. 4d.	=	106	13	4
Labour cost.		=	145	0	0
	Total	=	1,003	8	4

These figures amply prove that the diminishing pressure-loss system can be carried out at a cost substantially similar to the average pressure-loss system. Indeed, the whole of the advantages of the reversed return system are obtained and several others at considerably less cost. *To employ the reversed return system not only gives no advantage in working, but is more costly in maintenance and 40 per cent. more expensive in the initial cost of distributing mains.*

Simplification of Pipe Sizing in Large Buildings by Use of Diminishing Pressure-loss System.

In the case of large blocks of flats, offices, and similar buildings arranged for hot-water heating with a two-pipe underfeed system it often happens that many if not all of the risers have similar B.T.U. loads. With pipe-sizing systems hitherto employed, the sizes of pipes for such risers has varied according to their position and the unconsumed pressure available at their junction with the main pipes.

By using the diminishing pressure-loss system, for either gravity or pumped circulation, it is possible to have a similar available pressure for each riser irrespective of its position, and all risers therefore become similar in sizes. The additional time required in sizing mains is recovered many times by the saving in intricate pressure balancing on the risers.

Graphical Balancing of Pressures.

We have seen in examples of pipe sizing how involved the calculations for pressure balancing can become. The 'total length' scale on

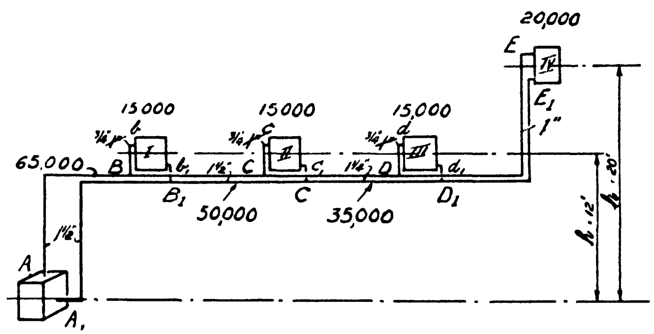


FIG. 24. Two-pipe underfeed system.

the nomographic calculators is, however, a means of eliminating one multiplication. There is a comparatively simple means of dispensing with nearly all written calculations by the application of graphical addition and multiplication, a method which can be applied to any pipe-sizing calculation, particularly for large systems. Indeed, for the largest systems it is possible to produce the whole of the calculations graphically on a comparatively small sheet of paper.

As an example, consider the two-pipe underfeed system illustrated in Fig. 24, which is to be sized for $t_f - t_r = 40°$ F. We are already familiar with the general method of procedure, and find the circulating pressure to be 0·16 in. per ft. of h, that is, 1·92 and 3·2 in. total pressure according to the height of the radiators. The pipes are provisionally sized in accordance with established principles, these sizes being indicated in circles on the isometric of the system.

We have now to balance all pressures to the various radiators. For this purpose it is recommended that calculations should be done graphically as in Fig. 25. On the paper, draw a base line AB and a vertical scale AC. The scale AC should be graduated in units and tenths of any convenient dimension, so that the maximum circulating pressure for the highest radiator can be represented within the length of the scale. On the vertical line ab the pressure losses at any points along the system may be projected across and also the available pressure for the radiators at the lower level.

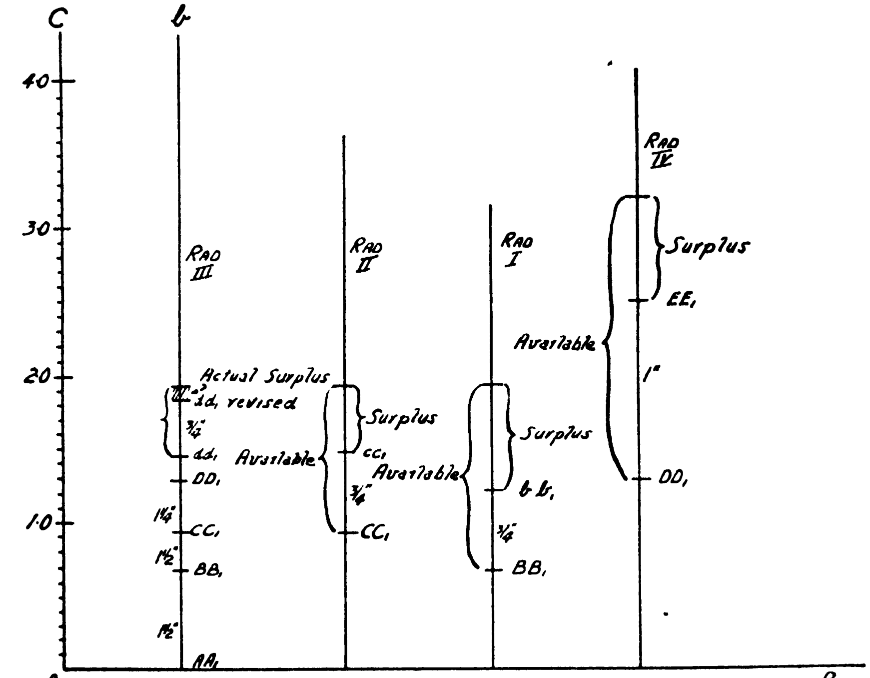


FIG. 25. Graphical pressure balancing.

Knowing the lengths of pipe and single resistance coefficients, the actual pressure loss in lengths AA_1 to BB_1 on Fig. 24 may be read from the nomographic calculator, Fig. 1, and projected from the vertical scale on Fig. 26 on to the line ab, and similarly other pressure losses up to DD_1 may be indicated on line ab. The losses in the various radiator branches may for a small system be indicated on this same line, or for a larger and more involved system, which is usual, by separate lines, the consumed pressure up to the radiator branches being projected across to these lines. For clearness we will adopt this procedure, but radiator 3 may be conveniently shown on the first line. It may be seen that the surplus pressure for 1-in. connexions, shown bracketed on line ab, is very large, and $\frac{3}{4}$ in. is therefore used. The revised position of dd_1 is shown, the surplus pressure being the small length above this point.

Proceeding to radiator 2, the available pressure can be easily taken off with dividers and a suitable size pipe determined, $\frac{3}{4}$ in. being taken, the surplus being as shown. Similarly radiators 1 and 4 are decided.

Without reducing flow or return or parts of them, the graphical calculation shows surplus pressures approximately equal for all radiators except radiator 3, but it must be apparent that to accept the first diameter found for this of 1 in. would give the lower position of dd_1 as the final one, approximately balancing all radiator pressures.

It is felt that this example sufficiently explains the graphical method of balancing pressures, and it will be noticed that it provides a complete record of the sizes of all pipes and the unconsumed pressures for every radiator, these being readily visible.

In practice, the author advocates the use of standard-size squared paper, 13 in. $\times$ 8 in. being a convenient size. If the whole of the calculations cannot be shown on one sheet, it is a simple matter to transfer with dividers the pressure loss at any point to a further sheet. This method definitely eliminates errors due to arithmetical addition.

The references to surplus and available pressure on Fig. 25 are only given to explain the method and in practice are not used, as it becomes obvious which lengths refer to these, if the method is always employed.

Graphical pressure balancing enables the largest of systems to be accurately sized without any calculation on paper, and yet leaves an accurate record of the pressure losses in the various lengths of pipe.

In applying the method to systems having small available pressures care should be taken to choose a sufficiently large scale of pressures, whilst for pumped circulation the scale must of necessity be smaller, according to the total pressure employed. The various junction points are lettered or numbered for easy reference to the balancing graph or plans.

Graphical Calculation of Heat Losses.

The calculation of heat losses from rooms in an accurate and scientific manner is the first step in the design of any heating system. It is a calculation which, however, cannot be done rapidly by existing lengthy methods, and, moreover, being the key calculation it is not desirable to trust too inexperienced designers on this work, for an error at this stage is repeated throughout the design, and if discovered later cannot be properly rectified without altering many subsequent calculations.

Whilst extensive graphs have been published† for heat-loss calculations, none has hitherto been based on the nomographic principle, such as that in Fig. 26, which contains all the data and calculation steps for accurate heat-loss calculation in one simple graph.

It is customary to take off all surfaces losing heat and tabulate these in a standard form, multiplying to obtain area, and afterwards multiplying by the coefficients and temperature differences to obtain the B.T.U. requirements of the room.

The author's method eliminates the use of standard forms and all but mental calculation. The coefficients of heat transmission for various types of surface may be obtained from many reference books,‡ together with explanations of their derivation.

The nomograph, Fig. 26, has scales for *length, width, height of room*, the coefficient K in B.T.U. per sq. ft. *per degree F. per hour*, and the *total B.T.U. for the surface considered.*

It is based on a room temperature of 60° F. internally when 30° F. externally, and is capable of giving the heat required to balance infiltration losses equal to one air change per hour. There is no justification for giving further scales to take account of other temperature differences or air changes, as these are easily accounted for, in most cases by mental calculation.

The usual method of tabulating heat-loss calculations is also inconvenient owing to the time lost in referring from heat-loss calculations to layout drawings, so that, in view of having a calculator to give the heat loss of surfaces directly in B.T.U., the figures so obtained may be easily written on the plan inside the room in question.

The following example explains the manner of use of the nomograph, the room considered being similar to that calculated in detail in *Treatise on Central Heating* by F. Broadhurst Craig, and shown in Fig. 27.

Firstly we will deal with the air change B.T.U., requirements for two air changes per hour. As the nomograph deals with one air change any dimension may be doubled to take account of this, so that we use the dimensions $20 \times 20 \times 30$ in referring to the nomograph. Place a straight edge from 20 on scale A to 20 on scale D, the line joining these points intersecting the ungraduated scale B. From this intersection, with the straight edge, strike a line to 30 on scale F, intersecting the scale C at 7,000 B.T.U.

Now, dealing with the windows, as subsequently the walls will be

† *Gesundheits Ingenieur*, Oct. 5, 1929; 'Calcul rapide des déperditions par procédé graphique', by M. L. Mossé, *Chauffage et Ventilation*, Aug. 1931.

‡ See *Heating and Ventilation* by L. J. Overton, *The Heating and Vent. Engr.*, *Barker on Heating*, *Inst. H. & V. Engrs. Proc.*, etc., etc.

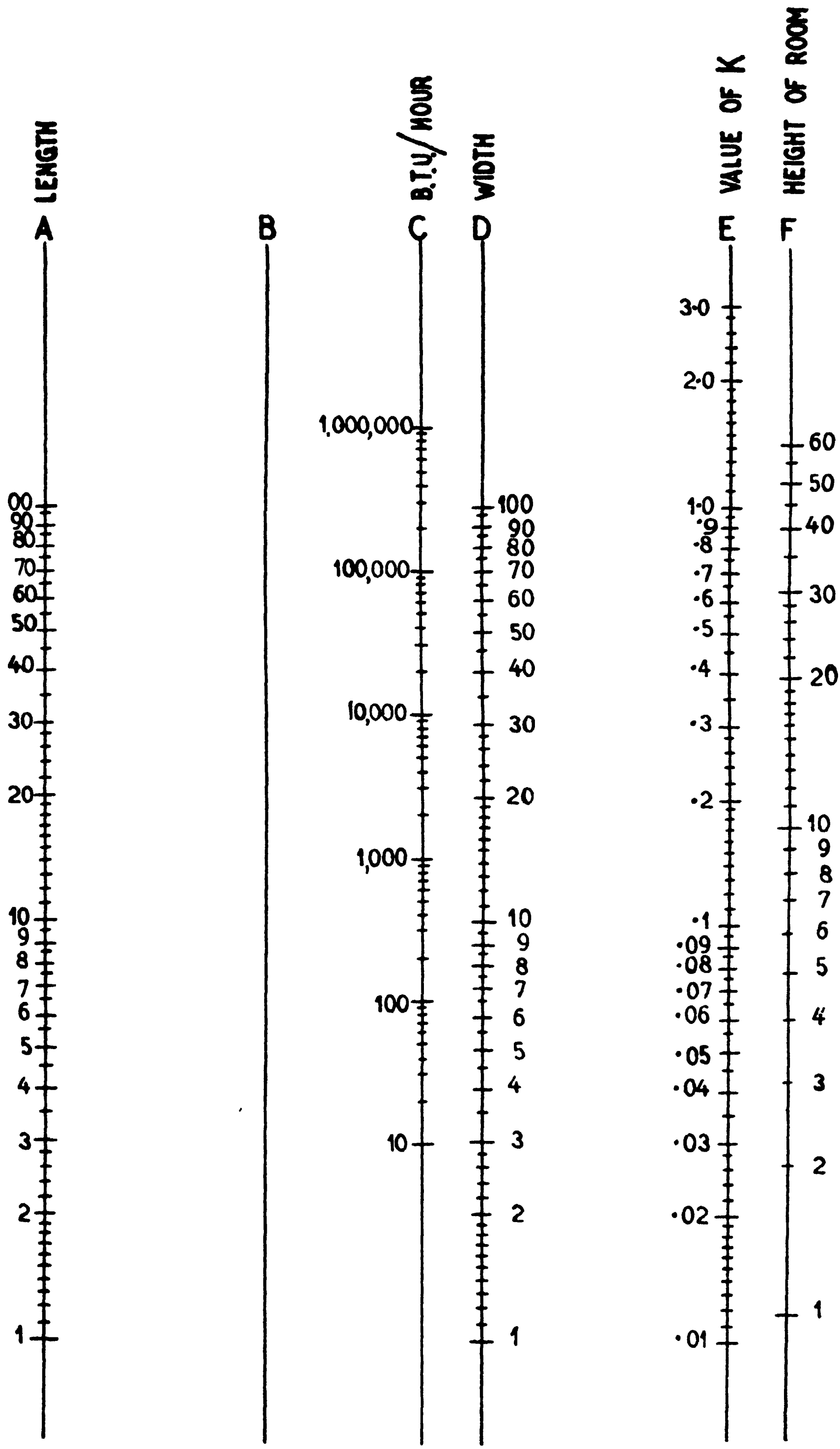


FIG. 26. Heat loss nomograph.

measured to include these, the coefficient K used should be the actual value for glass, namely 1·09, minus that chosen for the outside wall. namely, 0·24, leaving $1{\cdot}09 - 0{\cdot}24 = 0{\cdot}85$ as the value to be used.

Using the straight edge, a line from 12 (the total length of window) on scale A to 8 (the height of the window) on scale D intersects the ungraduated scale B, and the line from this intersection to 0·85 on scale E intersects scale C at 2,500 B.T.U.

As it is usual to allow additional losses for east-exposed walls, this is now taken separately. A line from 20 on scale A to 15 on scale D

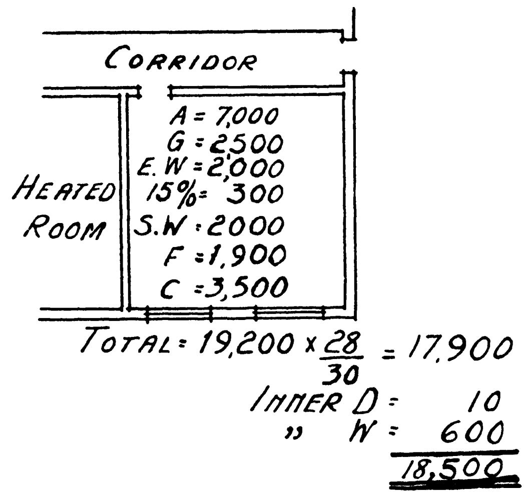


FIG. 27. Example room for heat losses.

intersects scale B, and a line from this intersection to 0·24 on scale E intersects scale C at 2,000 B.T.U.

Similarly, the other outer wall is found to require 2,000 B.T.U.; the floor for $K = 0{\cdot}15$, 1,900 B.T.U.; the ceiling for $K = 0{\cdot}28$, 3,500 B.T.U.

In this example there are also inner-wall losses due to adjacent spaces being at lower temperatures, and in these cases it is found that the easiest way of dealing with them is to adjust K mentally for the temperature difference in question. That is, for the inner wall, instead of $K = 0{\cdot}36$ take $K = 0{\cdot}06$, and for the door, instead of $K = 0{\cdot}4$ take $K = 0{\cdot}07$.

The value for the door should, however, be treated similarly to windows, that is, $0{\cdot}07 - 0{\cdot}06 = 0{\cdot}01$ should be taken, when the B.T.U. becomes 10 and the inner wall 600.

The various losses are written on the drawing in the manner indicated, and as the outside temperature required is 32° F., giving $60 - 32 = 28°$ F. difference instead of the 30° F. for which the nomograph is designed, the outside losses are multiplied by 28/30 as indicated.

In taking off heat losses for any room, it is an easy matter, having scaled a dimension, to remember it for sufficient length of time to refer to the nomograph, but having done this the dimensions are not required and hence are not recorded. The maximum error in using the nomographic calculator carefully does not exceed half of 1 per cent., whilst the time required for calculating heat losses is reduced to one-quarter of that for normal calculation methods. The edge of the scale employed for measuring area from the drawing is convenient for use as a straight edge on the nomograph.

Moreover, having the heat requirements marked in each room is of advantage in the subsequent calculations of B.T.U. to be carried by the various pipes. The heat-loss coefficients are usually remembered by the designer without reference to tables, unless the construction is of an unusual nature.

Instances may arise in the application of the nomograph where surfaces having one dimension greater than the range of scales have to be dealt with, such as 200 ft. × 30 ft., but in such cases reference can be made to 100 × 60 without inconvenience.

Panel- and Floor-heating Design.

Whilst the application of radiant-heating systems differs only slightly from any other, there has been some difficulty in defining a rational method of calculation for determining the requisite surface of the radiant element or panel.

Barker has shown† that the usual heat-loss calculation has little connexion with the heat requirements where heating is principally by radiation, but rather depends upon the power of the human body to lose heat according to the surroundings, the amount of heat required being such as will tend to balance this loss.

As a basis of calculation, he suggests the calculation of the 'mean radiant temperature', this being found by multiplying the area of various surfaces of the room, such as outer and inner walls, ceiling, floor, windows, etc., by their respective assumed surface temperatures, adding these products together and dividing by the total area considered, that is,

$$\text{Mean radiant temperature} = \frac{(\text{glass area} \times \text{temperature}) + (\text{wall area} \times \text{temperature}) + (\text{ceiling and floor areas} \times \text{temperature})}{\text{total area}}.$$

† 'The principle of calculation of low temperature radiant heating', by A. H Barker, *Proc. Inst. H. & V. Engrs.*, vol. xxx.

The surface temperatures suggested are as follows:

Window glass	45° F.
Outside walls	50° F.
Inside walls	55° F.
Ceiling and floor	55° F.

We will consider the calculation of this temperature for the room taken as an example in heat-loss calculations, Fig. 27.

The mean radiant temperature will be found as follows:

Windows	96 × 45	=	4,320
Outer walls . . .	504 × 50	=	25,200
Inner walls . . .	300 × 55	=	16,500
Ceiling and floor . .	800 × 55	=	44,000
	1,700		90,020

$$\frac{90{,}020}{1{,}700} = 53^\circ \text{ F. mean radiant temperature.}$$

According to the temperature of the heat-radiating surface, its emissivity compared with a black surface, and the desired mean radiant temperature to be provided for the body of the occupant, so the surface of the radiant panel will vary. It will be appreciated that the calculated mean radiant temperature of the surfaces in the room is an assumed condition before heating takes place, just as we assume 32° F. or 30° F. as a suitable basis external temperature for calculating heat losses in connexion with other systems of heating.

From the figures given by Barker the nomograph in Fig. 28 has been designed, enabling, without extensive calculation, the areas of radiant surface to be decided.

The surface temperature at which the panel will operate depends entirely upon its construction, and for the known types of panel is as follows:

Morganite high temperature (electric) . .	500° F.
Electrorad	250° F.
Rayrads, variable at will	140° F. upwards
Plaster surface (hot water)	110° F.
Floor heating	70°–80° F.
Plaster surface (electric)	90°–110° F.

It is not possible to calculate with any degree of accuracy the surface temperature for any desired type of construction, and experimental observations have therefore to be made for each particular type.

In applying the nomograph it should be noted that it applies only to ceiling- or wall-panels, *suitably arranged.*

Suppose, now, that the typical room we are considering is to be heated by panels at a temperature of 130° F. From Fig. 28, the

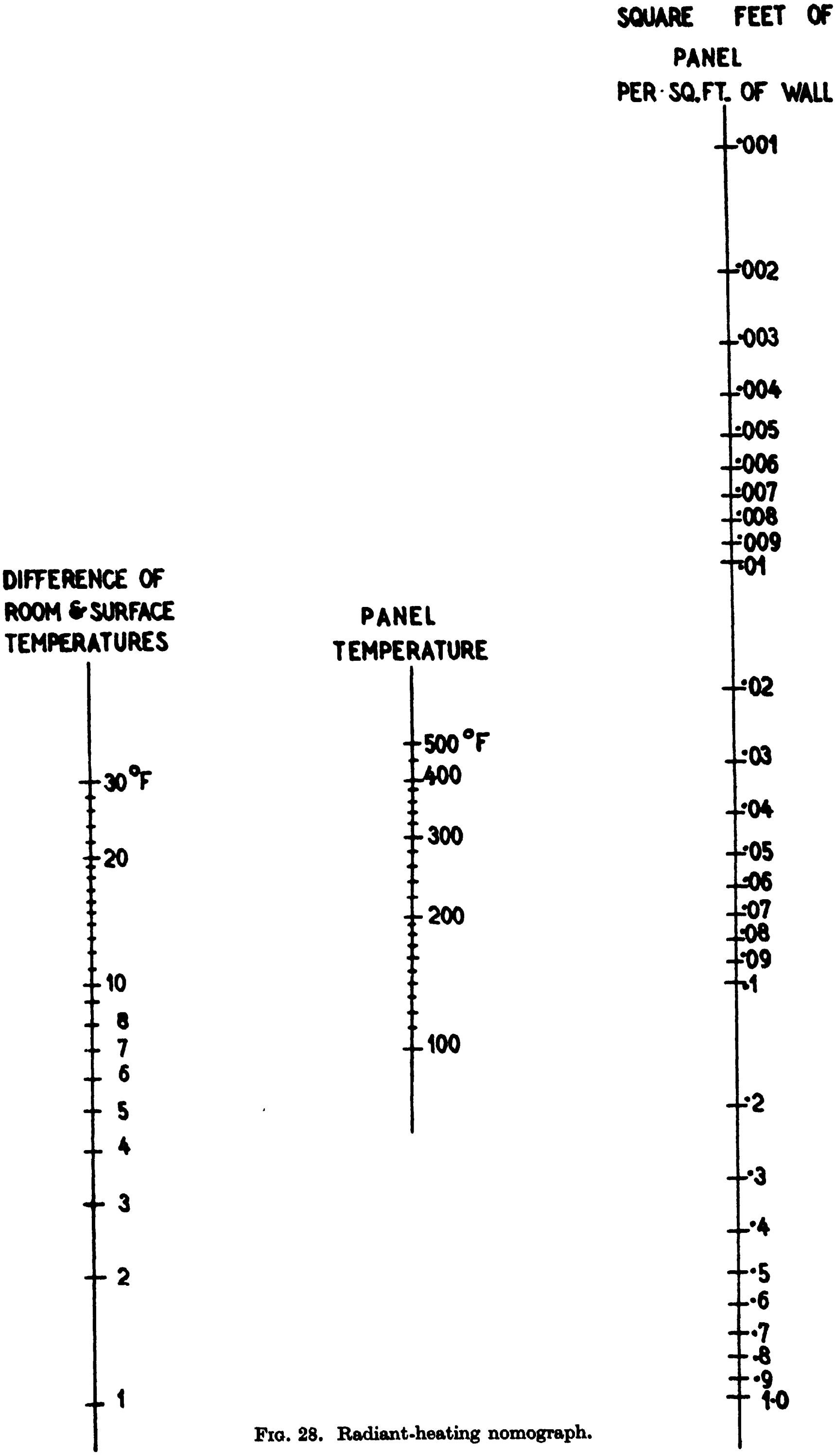


Fig. 28. Radiant-heating nomograph.

surface of panel required is 0·07 per sq. ft. of surface. The total area required is therefore

$$1{,}700 \times 0{\cdot}07 = 119 \text{ sq. ft.}$$

Although no other practicable method of calculating the necessary radiant surfaces has been devised, it must be appreciated that the disposition of the panels is a matter greatly affecting results, and for this reason the ceiling position is generally to be preferred to any other. The proposed arrangement of panel surface is shown in Fig. 29.

The 'Rayrad', a cast-iron construction radiant-heating panel, is one extensively used, and is suitable for higher temperatures. If

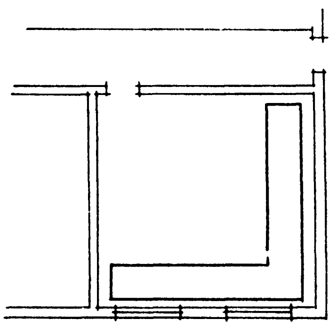


FIG. 29.
Arrangement of panels.

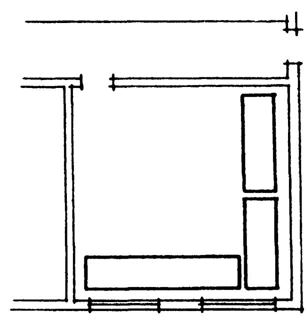


FIG. 30.
Arrangement of rayrads.

the typical room is to be heated by this type of panel, at a temperature of 160° F., from Fig. 28, 0·058 sq. ft. of radiant panel for each sq. ft. of room surface is required. The necessary total area is thus $1{,}700 \times 0{\cdot}058 = 98{\cdot}5$ sq. ft., which could be provided by 30 sections each 30 in. high, the assembled panels being arranged as in Fig. 30.

Apart from the radiant panel areas, it is also necessary to know the precise amount of heat dissipated from the surface, to determine that required to be carried by the pipes. For this purpose the following figures may be employed:

Plaster panel	@ 110° F.	=	75	B.T.U.	per sq.	ft.
Rayrad panel	@ 160° F.	=	170	,,	,,	,,
Rayrad on wall	@ 160° F.	=	190	,,	,,	,,

With all radiant-heating systems, that being most efficient in heating effect is the one having the largest area of radiant surface and the lowest temperature, assuming the position of the surface to be similar when comparing.

Floor-heating systems are of several types, some of which are illustrated in section in Figs. 31, 32, and 33. The emission from each type is stated. The floor-heating system is of particular value for churches, open-air schools, and similar buildings, but for sanatoriums it is necessary to employ either wall or ceiling panels. For private houses either floor, wall, or ceiling positions are suitable, whilst for offices ceiling position is advisable.

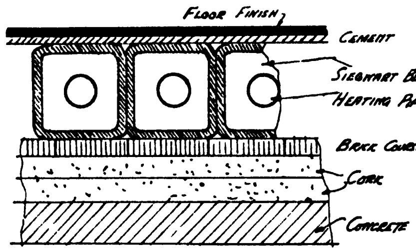


FIG. 31. Floor-heating system. Low-pressure steam at 212° F., surface temperature = 83° F., emission = 46 B.T.U./sq. ft.

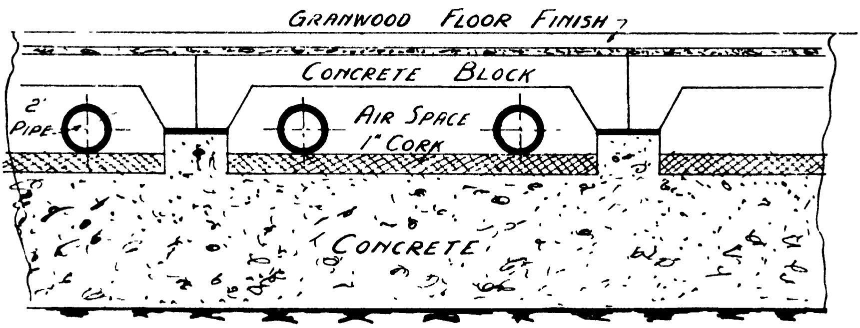


FIG. 32. Floor-heating system.

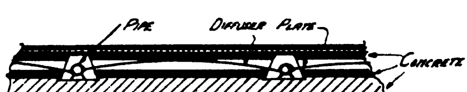
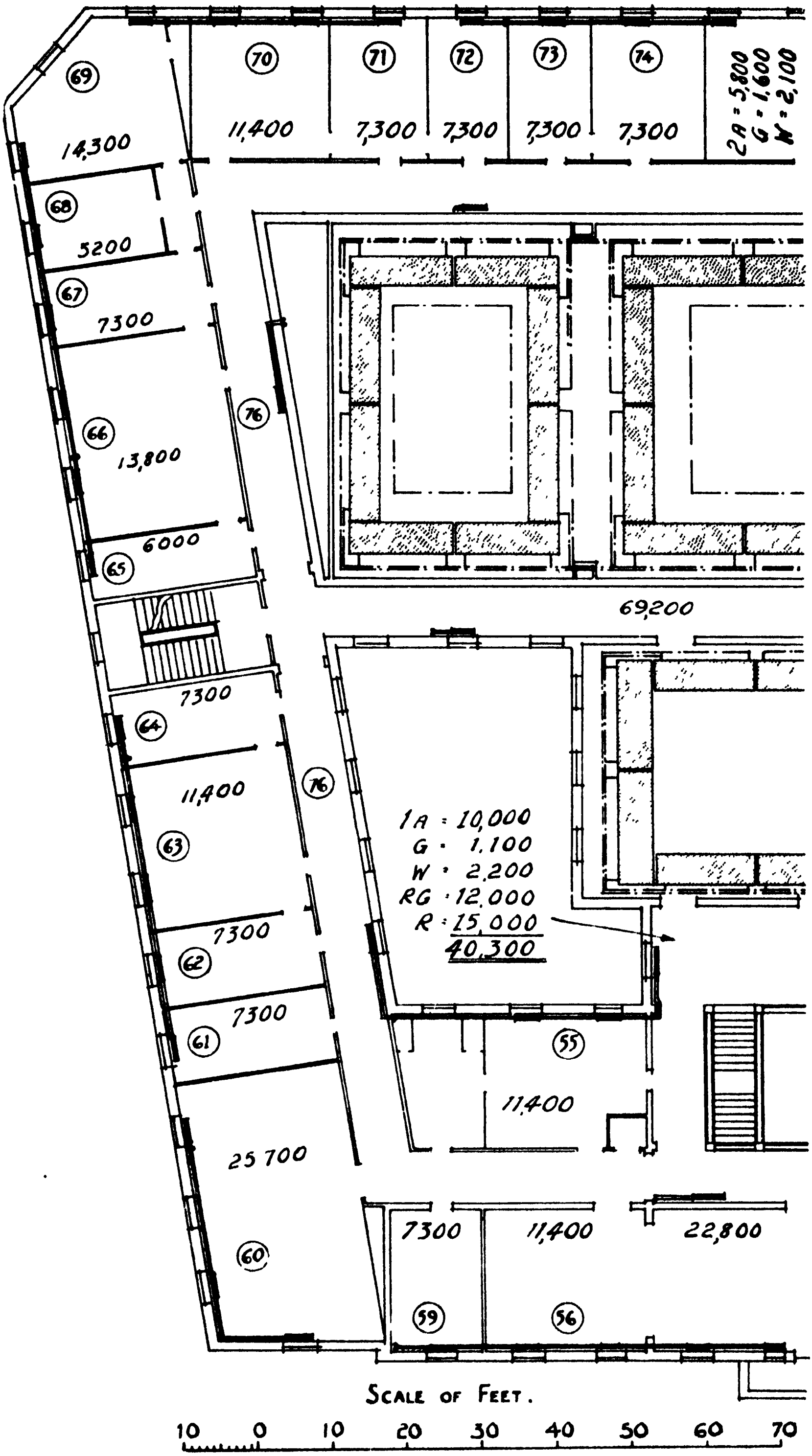


FIG. 33. Floor-heating system. Low-pressure steam at 212° F., emission = 77 B.T.U./sq. ft.

Floor-heating systems comprising 1-in. diameter pipes fixed at 1-in. centres in a bed of sand are also employed. Plaster panels usually consist of ½-in. or ¾-in. iron or copper pipes arranged at 4-in. to 6-in. centres, the thickness of plaster on the face of the pipes being ½ in.

Heating System for Town Hall Building.

To demonstrate in greater detail the methods of design previously outlined, it is proposed to deal with the equipment of a typical town hall building such as that illustrated in the loose plate, Fig. 34, the building design of which is reproduced by kind permission of Mr. C. Cowles-Voysey, F.R.I.B.A.



69
70
71
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14,300
11,400
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7,300
7,300
7,300
2A = 5,800
G = 1,600
W = 2,100
68
5200
67
7300
66
13,800
76
6000
65
69,200
7300
64
11,400
76
63
1A = 10,000
G = 1,100
W = 2,200
RG = 12,000
R = 15,000
40,300
7300
62
7300
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11,400
25 700
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11,400
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60
59
56
Scale of Feet.
10 0 10 20 30 40 50 60 70

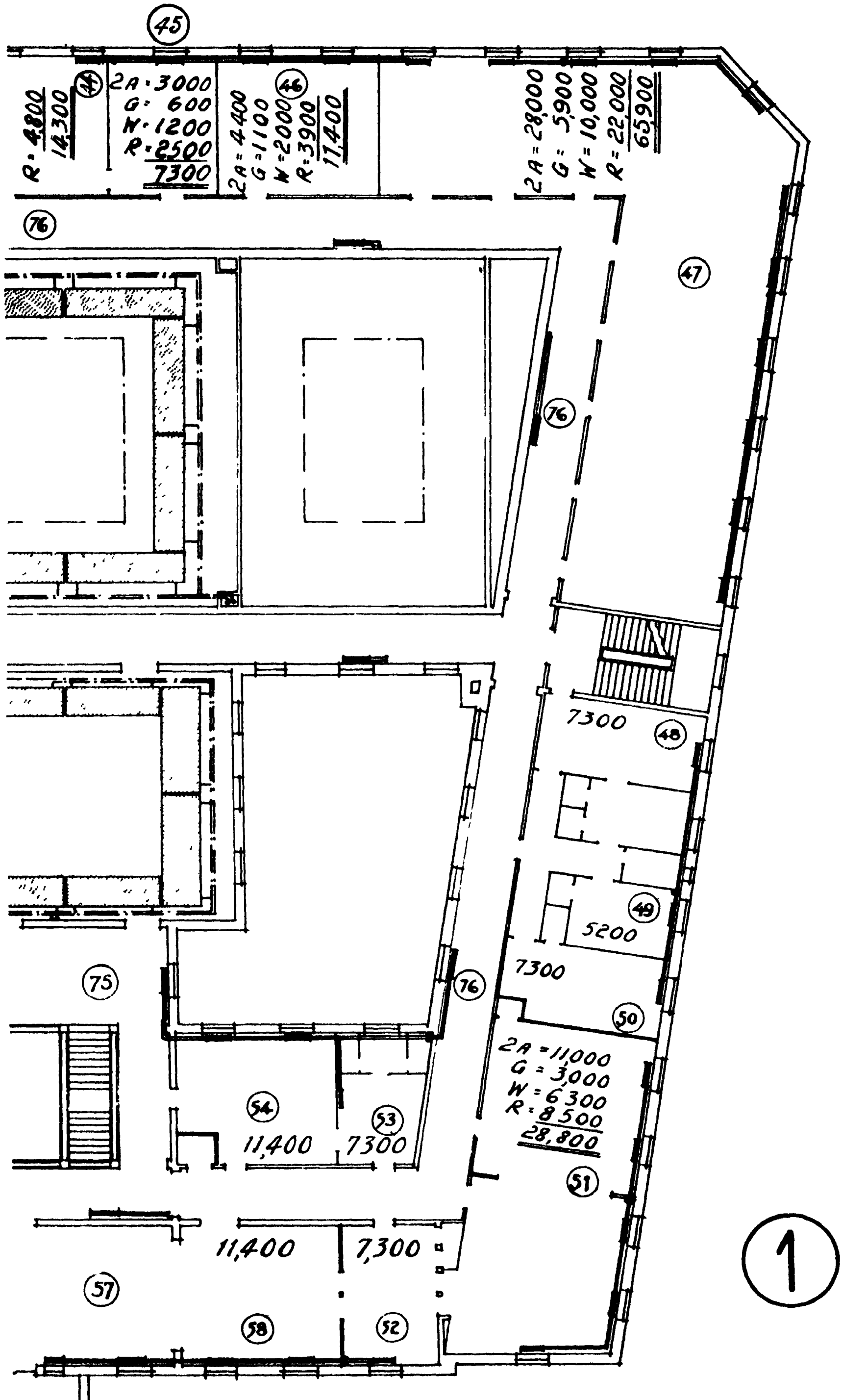

FIG. 34 A. First-floor plan showing heating system in Town Hall building.

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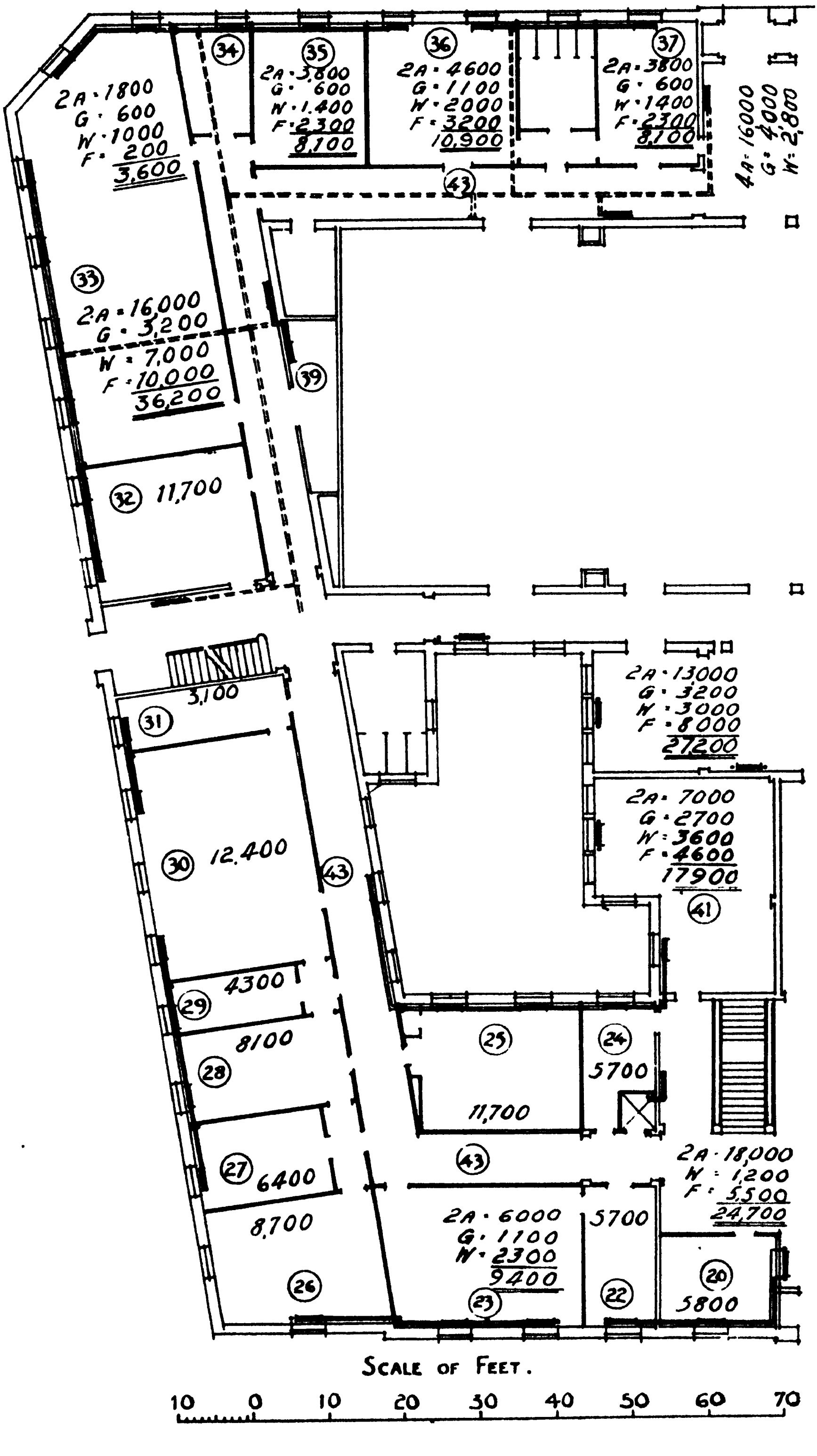
2A · 1800
G · 600
W · 1000
F · 200
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35
2A · 3,600
G · 600
W · 1.400
F · 2,300
8,100
36
2A · 4600
G · 1100
W · 2000
F · 3200
10,900
37
2A · 3800
G · 600
W · 1400
F · 2300
8,100
4A · 16000
G · 4000
W · 2,800
43
33
2A · 16,000
G · 3,200
W · 7,000
F · 10,000
36,200
39
32
11,700
3,100
31
30
12.400
43
4300
29
8100
28
27
6400
8,700
26
2A · 13000
G · 3200
W · 3000
F · 8000
27200
2A · 7000
G · 2700
W · 3600
F · 4600
17900
41
25
11,700
24
5700
43
2A · 18,000
W · 1,200
F · 5,500
24,700
2A · 6000
G · 1100
W · 2300
9400
23
5700
22
20
5800
Scale of Feet.
10 0 10 20 30 40 50 60 70

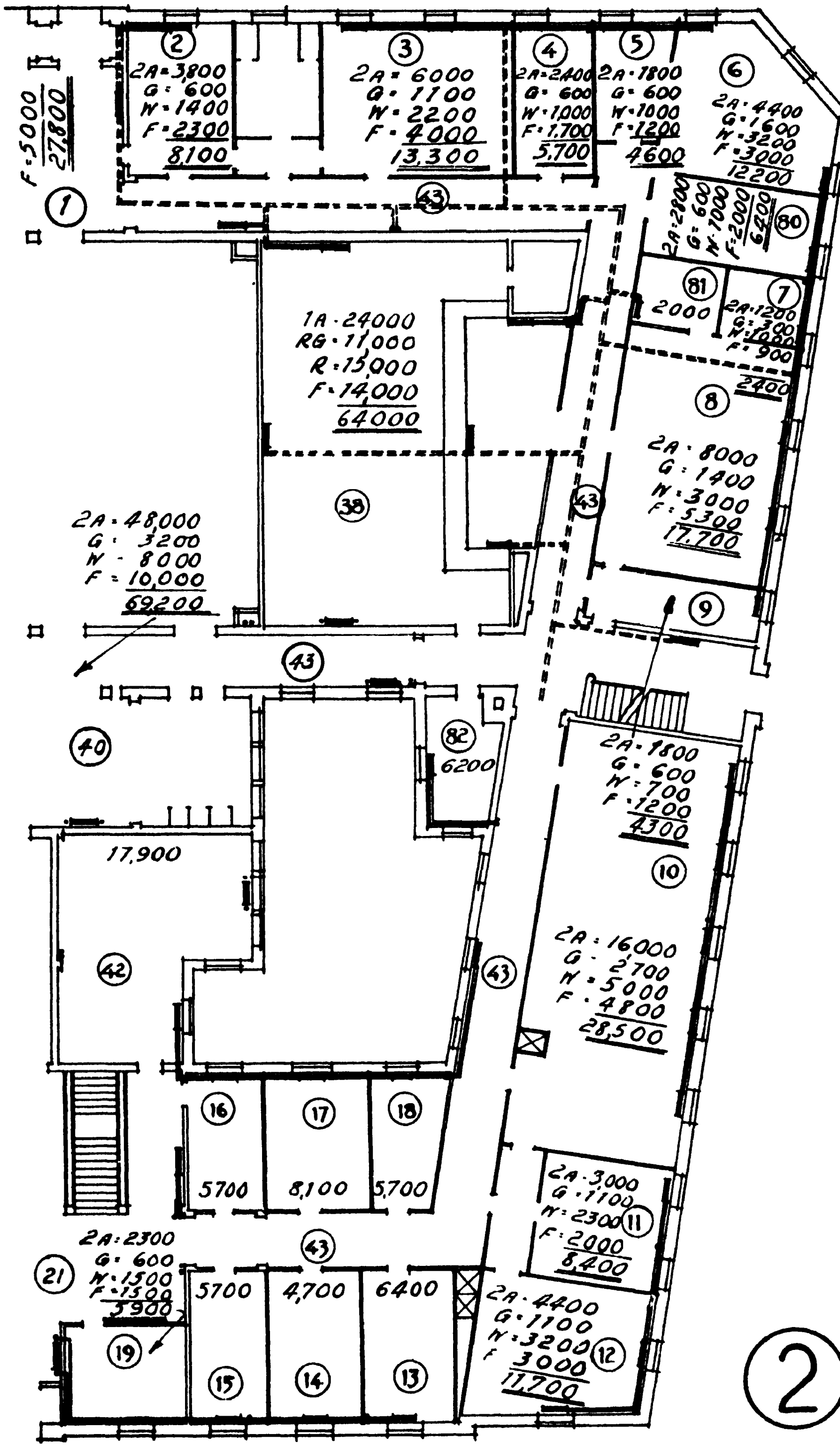
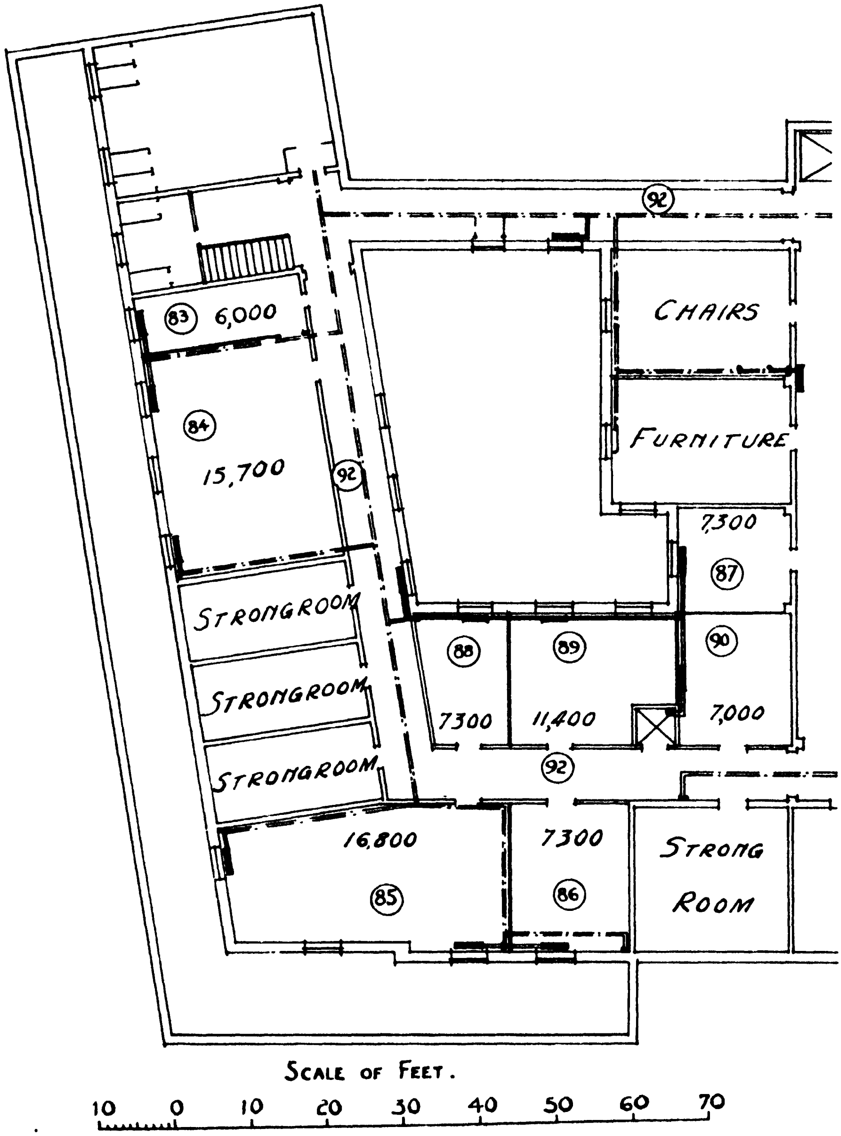


Fig. 34 b. Ground-floor plan showing heating system in Town Hall building.



92
CHAIRS
83
6,000
FURNITURE
84
15,700
92
7,300
87
STRONGROOM
88
89
90
STRONGROOM
7300
11,400
7,000
STRONGROOM
92
16,800
7300
STRONG
ROOM
85
86
SCALE OF FEET.
10
0
10
20
30
40
50
60
70

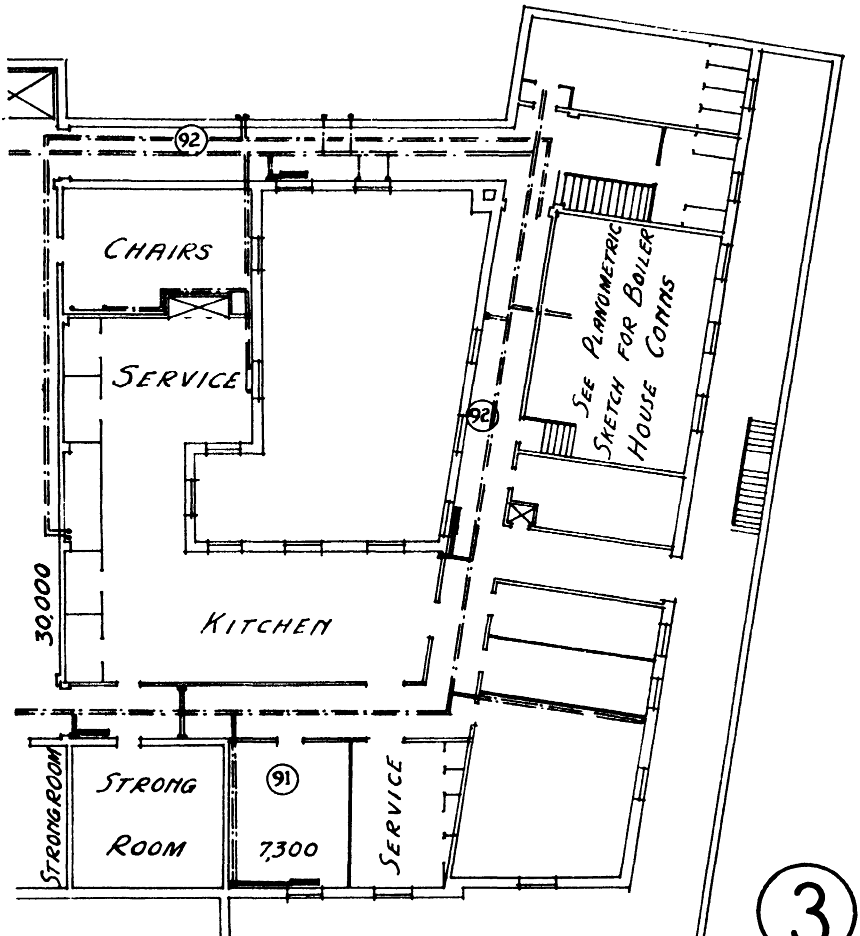


FIG. 34 c. Basement plan showing heating system in Town Hall building.

Although the nature of the building is such as to call for ventilation and air-conditioning equipment, we will deal at this stage only with the hot-water heating system. We must firstly decide whether to adopt radiator or concealed panel heating, and in this case would perhaps propose that the majority of the rooms would be served by radiators, only the Council Chamber, Division Lobby, and Assembly Halls having radiant heating.

By the use of the heat-loss nomograph, Fig. 26, calculation of the heat losses from all rooms, except those served by radiant panels, should now be made, the figures for these being indicated in their respective rooms in accordance with the method outlined on page 66.

For this purpose, the nature of the exposed surfaces and the heat-loss coefficients have been taken as follows:

Outer walls, 14 in. brick stone faced	$K = 0{\cdot}27$
Windows	$K = 1{\cdot}09$
Floor, wood or concrete	$K = 0{\cdot}25$
Roof lights, glass	$K = 1{\cdot}2$
Roof, 6 in. concrete flat	$K = 0{\cdot}28$

It is also assumed that two air changes per hour will take place in all rooms.

With regard to those rooms provided with radiant heating, the calculation of the necessary surfaces of radiant panels will be carried out by the method given on pages 67–71, the details of calculations being as follows:

Large Assembly Hall

		Sq.ft. × *Mean temp.*		
Roof glass, 40 × 25	=	1,000 × 45	=	45,000
Roof, (60 × 50) − 1,000	=	2,000 × 50	=	100,000
Outer wall, 120 × 8	=	960 × 50	=	48,000
Inner wall, 220 × 22	=	4,840 × 55	=	266,000
Floor, 60 × 50	=	3,000 × 55	=	165,000
Totals		11,800		624,000

$$\text{Mean radiant temperature of room} = \frac{624{,}000}{11{,}800} = 53^\circ \text{ F.}$$

Referring to the radiant heating nomograph, Fig. 28, and assuming 130° F. as the surface temperature of the radiant panel, the difference between 60° F., the radiant temperature for which the nomograph is designed, and 53° F. = 7° F. A straight line from 7° F. on the first scale through 130° F. on the second intersects the panel-surface scale at 0·082 sq. ft.

The necessary panel area is thus 11,800 × 0·082 = 970 sq. ft. It will be seen that this area is arranged to surround the roof light.

Assuming an emission of 120 B.T.U. per hour per sq. ft. of panel, at the temperature employed, the total emission for the room is $970 \times 120 = 117{,}000$ B.T.U. per hour.

Small Assembly Hall

		Sq. ft.	Mean temp.		
Roof glass, 15 × 25	=	375 ×	45	=	16,900
Roof, (50 × 34) − 375	=	1,325 ×	50	=	66,250
Outer wall, 118 × 8	=	944 ×	50	=	47,200
Inner wall, 168 × 22	=	3,700 ×	55	=	203,000
Floor, 50 × 34	=	1,700 ×	55	=	93,500
Totals		8,044			426,850

$$\text{Mean radiant temperature} = \frac{426{,}850}{8{,}044} = 53^\circ \text{ F.}$$

Panel surface required per sq. ft. of room surface = 0·082 sq. ft.
Panel surface = 8,044 × 0·082 = 660 sq. ft.
Panels arranged on the ceiling as shown.
Approximate emission of panels at 120 B.T.U. per sq. ft. = 79,500 B.T.U. per hour.

Council Chamber

		Sq. ft.	Mean temp.		
Roof, 60 × 35	=	2,100 ×	50	=	105,000
Glass, 30 × 7	=	210 ×	45	=	9,450
Outer wall, (82 × 20) − 210	=	1,430 ×	50	=	71,500
Inner wall, 102 × 20	=	2,040 ×	55	=	112,000
Floor, 60 × 35	=	2,100 ×	55	=	115,000
Totals	=	7,880			412,950

$$\text{Mean radiant temperature} = \frac{412{,}950}{7{,}880} = 53^\circ \text{ F.}$$

Panel surface = 7,880 × 0·082 = 645 sq. ft. arranged on the ceiling.
Approximate emission of panels = 645 × 120 = 77,500 B.T.U. per hour.

It will have been noticed that in all cases the calculated mean radiant temperature of the room surfaces has been 53° F., this being the usual figure for average rooms, without undue glass or outer wall area.

Summary of Heat Losses.

The summary of heat-loss requirements for all rooms, taken from the totals on the plans, is as follows:

Room	B.T.U. per hour
1 Foyer	27,800
2 Cloaks 1	8,100
3 Men's Retiring	13,300
4 Office 1	5,700
5 Deputy Treasurer	4,600
6 Borough Treasurer	12,200
7 Typists	2,400

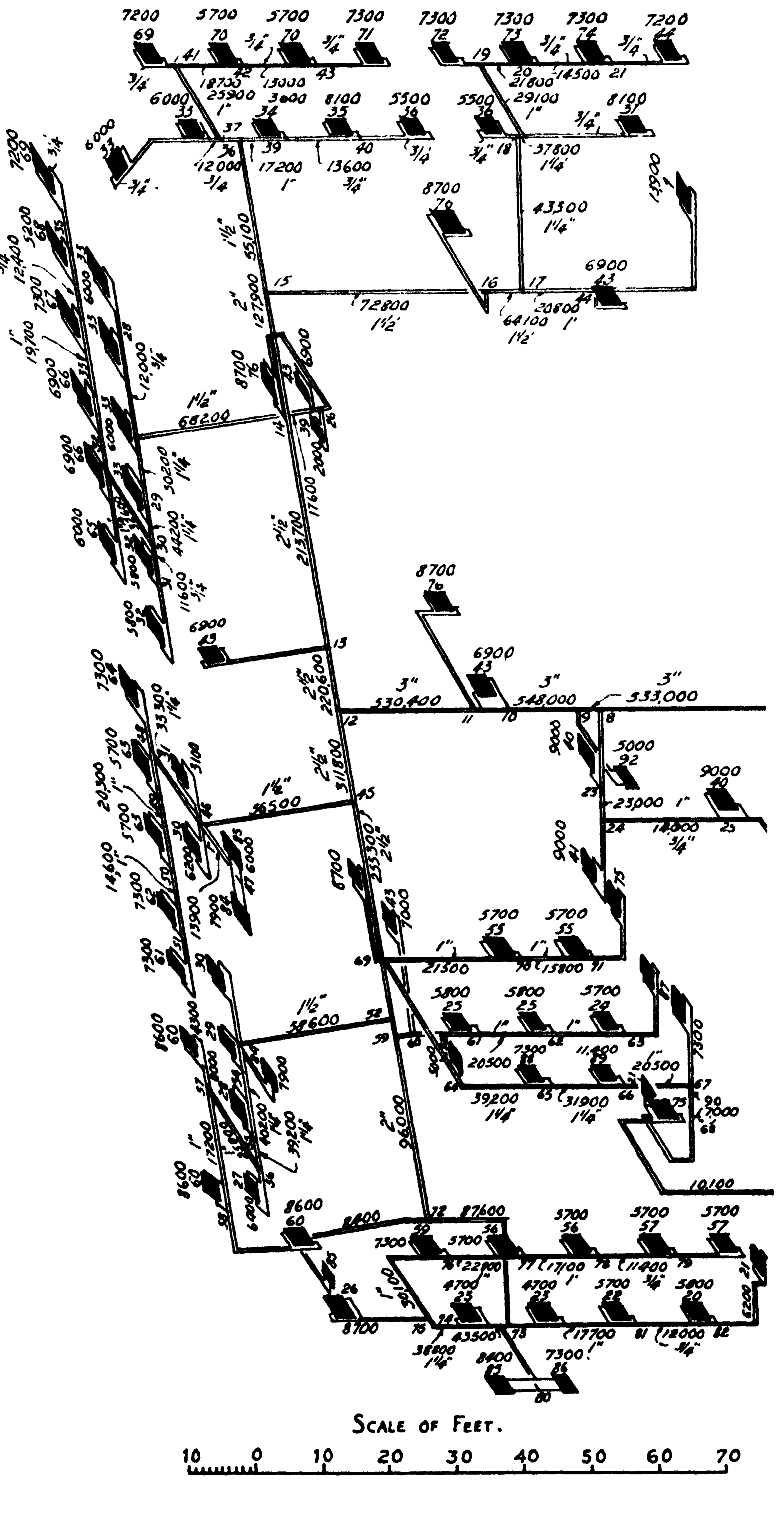
Room	B.T.U. per hour
8 Clerks' Office 1	17,700
9 Inquiries	4,300
10 Drawing Office	28,500
11 Deputy Engineer	8,400
12 Borough Engineer	11,700
13 Road Surveyor	6,400
14 Building Inspector	4,400
15 Sewers	5,700
16 Chief Clerk	5,700
17 Typists' Filing	8,100
18 Interviews 1	5,700
19 B.E. Inquiries	5,800
20 General Inquiries	5,800
21 Entrance Hall	24,700
22 Health Visitors 1	5,700
23 ,, ,, 2	9,400
24 Inspectors	5,700
25 Laboratory	11,700
26 M.O.H.	8,700
27 Consulting Room	6,400
28 Assistant	8,100
29 Interviews 2	4,300
30 Clerks' Office 2	12,400
31 Inquiries	3,100
32 Street Traders	11,700
33 General Office	36,200
34 Registrar	4,600
35 Marriages	8,100
36 Women's Retiring Room	10,900
37 Cloaks 2	8,100
38 Rates Office	64,000
39 Waiting Room	2,000
40 Refreshment	27,200
41 Inspectors, Male	17,900
42 Accountants General	17,900
43 Gd. Floor, corridors	69,200
44 Valuation Clerks	14,300
45 Rating	7,300
46 Electors	11,400
47 Spare Offices	65,900
48 Living Room	7,300
49 Bedroom 1	5,200
50 ,, 2	7,300
51 Staff Dining Room	28,800
52 Service	7,300
53 Robing	7,300
54 Lady Members	11,400
55 Members' Robing	11,400
56 Committee 1	11,400
57 ,, 2	22,800
58 ,, 3	11,400
59 Deputation	7,300

Room	B.T.U. per hour
60 Members' Room	25,700
61 Aldermen robing	7,300
62 Mayor's robing	7,300
63 Mayor's parlour	11,400
64 Mayor's Secretary	7,300
65 Inquiries	6,000
66 General Office	13,800
67 Chief Clerk	7,300
68 Interviews	5,200
69 Typists	14,300
70 Town Clerk	11,400
71 Deputy Town Clerk	7,300
72 Assistant Town Clerk	7,300
73 Assistant Solicitor	7,300
74 Legal Office	7,300
75 Division Lobby	40,300
76 1st Floor Corridors	69,200
77 Council Chambers	77,500
78 Large Assembly Hall	117,000
79 Small Assembly Hall	79,500
80 Machinery	6,400
81 Tickets	2,000
82 Chief Clerk	6,200
83 Office	6,000
84 Stationery	15,700
85 Plan Room	16,800
86 Store	7,300
87 Telephone Exchange	7,300
88 P.H. Stock	7,300
89 ,, ,,	11,400
90 ,, ,,	7,000
91 Store	7,300
92 Basement Corridors	30,000
Total B.T.U. requirements	1,469,200

This figure, with the addition of the heat loss from pipes and further heating requirements for ventilating or similar equipment, will represent the total power of the installation.

We are now faced with the problem of having to serve a radiant heating system at a low temperature and the radiator system at the usual 180° F. flow temperature, and to overcome this difficulty it will be decided to provide a calorifier for heating the radiant panel circulation, the internal heating surface of this calorifier being connected to the boiler plant, which will consist of cast-iron sectional boilers.

The suggested system of piping is indicated on the plans, and is a two-pipe system, accelerated circulation being employed, the main pipes being run in trenches in the ground floor. The scheme is shown in isometric form in Fig. 35, the B.T.U. required at each radiator


Scale of Feet.
10 0 10 20 30 40 50 60 70

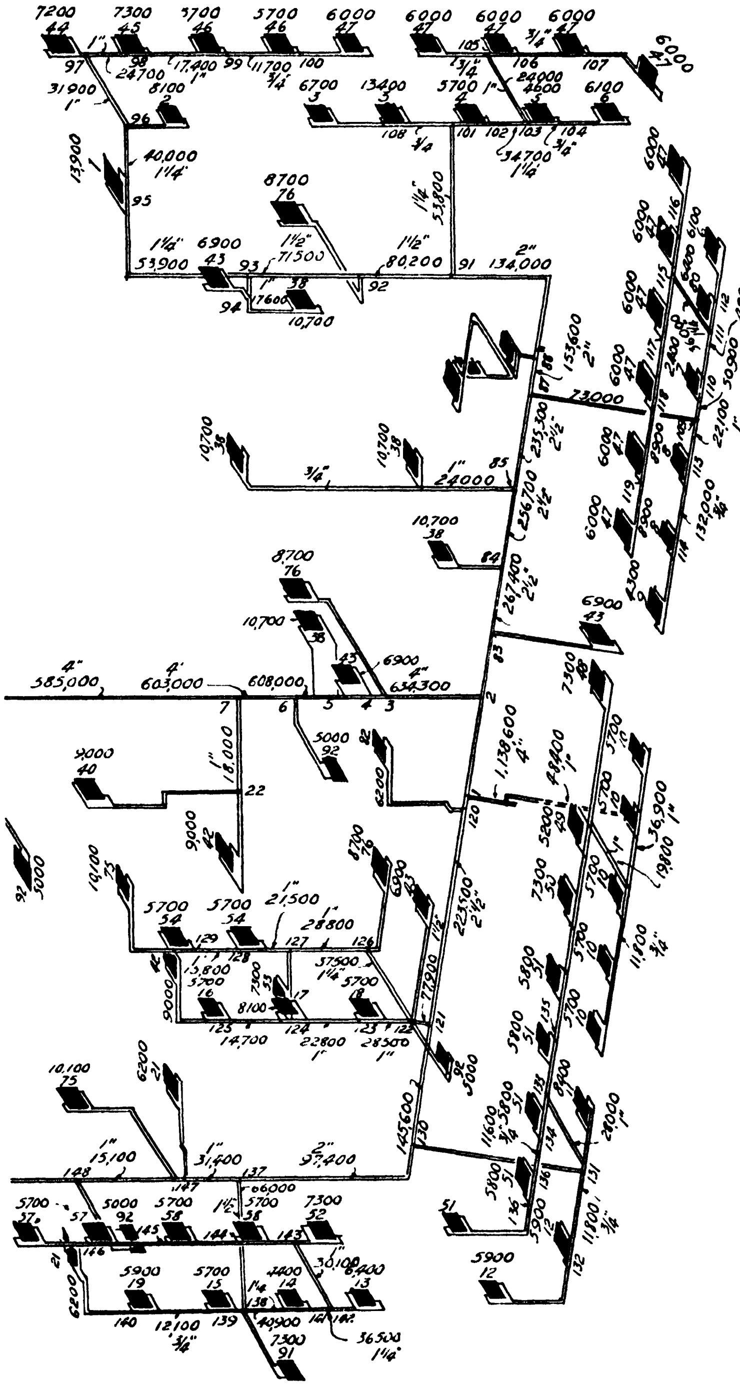


FIG. 35. Isometric layout of heating system.

and that to be carried by all pipes also being shown, whilst the various points in the piping are figured for reference.

We are now able to commence upon the sizing of the pipes.

It should be noted in summating the B.T.U. quantities to be carried by the various pipes that the total should agree substantially with the total of the heat losses. There is, of course, every possibility of slight errors in addition unless mechanical adding machines are employed, but provided the two totals agree within 1 per cent., any error will not be of sufficient magnitude to affect the calculation of pipe sizes. Should, however, on completion of the process of adding the B.T.U. for the pipes, any greater error be found, it is necessary to check every step until this is found.

We must also note at this point that, in applying the heat-loss nomograph for calculating losses from the rooms, there are often cases where common sense enables the loss for any particular room to be determined by comparison with similar rooms which have been calculated in detail, and Fig. 34 has several instances of this.

As the system being considered is to be run as an accelerated circulation, we must decide upon some basis of total pump-head upon which to work. This matter is one often causing trouble to inexperienced designers owing to the numerous possibilities for which there can apparently be no general rule. If, however, it is remembered that the maximum loss of pressure per foot run of pipe is not to exceed 0·2 in. of water, we have a basis upon which to commence sizing the pipes.

The total or maximum travel from the boilers to the farthest radiator and back is approximately 800 ft., and allowing 12 per cent. addition to cover the resistance of fittings, gives a total of 900 ft., but in choosing pipes the average loss per foot will perhaps not exceed 0·13 in.

The desired pump-head will therefore be $\frac{900 \times 0{\cdot}13}{12} = 10$ ft. We will proceed to work out pipe sizes on this basis, but it should be noted that there is no reason why 5 or 15 ft. maximum should not be employed, although this would lead to larger or smaller pipes. The head of 10 ft. is likely to give pipe sizes which are economic to instal, will certainly give satisfactory circulation, and, indeed, viewed from all angles, there is no possible reason for not employing it. On the other hand, a pump-head higher than 10 ft. would mean employing velocities in the pipes too high for silence, and of course the amount of power consumed in driving the pump will be too great in comparison with the saving on pipes.

For the preliminary sizing of the whole of the pipes in the system, therefore, employ Fig. 20, and with a fixed pressure loss per foot of pipe of 0·2 in. select pipes according to the B.T.U. required, noting the sizes on the isometric arrangement in Fig. 35.

It must be obvious that many of the pipes so chosen, particularly in the case of branch circulation nearer the boiler, are then too large to consume the whole of the pump-head, but this may be adjusted in the graphical balancing of pressures, which is to follow.

Although in high buildings the additional pressure due to differences of temperature—or better described, the 'gravity head'—should be taken into the calculation, in this case it is so small in comparison with the total head of the pump as not to warrant consideration.

The normal temperature drop for which the system is to be designed has been taken as 15° F., and in view of the amount of piping in the system it would perhaps be found that it worked in practice on anything up to 20° F.

In the final balancing of systems of this nature it is necessary to effect a compromise between theory and practice to the extent of having in some cases large surplus pressures at radiators, which in practice will be balanced by valve regulation.

The system of graphical representation of pressure losses has already been briefly described, and we will now apply it to the system under consideration. For this purpose a sheet of paper should be used as in Fig. 36, with a base line and a vertical left-hand scale graduated to represent inches of water from 0 to, say, 120, a convenient scale being 1/20th full size.

A vertical line is required for every radiator on the system, on which the pressure consumed and the surplus, together with pipe sizes, will be indicated. The first step is to determine with the aid of the nomographic calculator, Fig. 20, the losses of pressure in the various lengths of pipe from the boiler, 1, 2, 3, 6, 7, etc., to radiator in room 44, joining at 21, these losses being marked on to the scale on the first vertical line. The dotted portion of the vertical pressure drop line is a measure of the unconsumed pressure, whilst the solid lower part represents the total pressure consumed for each radiator. Having calculated and plotted losses for the radiator mentioned, it is a simple matter to project on to the vertical line for any other particular radiator the loss of pressure up to the point at which it joins the piping system, and then to decide upon the diameter of pipe for each radiator in turn, adjusting diameters to consume as far as possible the whole of the maximum head. In following through this example it will have been noticed that the first main circuit considered consumes approximately 105 in. of water, and it is therefore advisable to decide on a maximum, say 108 in. or 9 ft., which is not to be exceeded by any circuit, indicating this by the horizontal chain dotted line at the top of the sheets. It will be observed that in very few cases is it possible to consume the whole of the available head or pressure; it would have been possible to do this by reducing sections of piping, or employing pipes of diameter smaller than ½ in., but this is not desirable. It is better to do any further control by the use of valves or disks, the surplus pressures being evident for each radiator, for rapid calculation of disk orifices or valve settings.

The panel heating system to certain rooms, in view of the necessity of working at temperatures lower than that required for the radiator system, would be served by a calorifier, or heated from a separate boiler if thought desirable. The method of sizing pipes and graphical balancing of pressures would be similar to that described for the main systems. Similarly, the heating requirements of the plenum plant would be served by independent piping circuits.

The author particularly wishes to stress the fact that the multitude of multiplying and addition calculations required in all other systems of sizing and pressure balancing have been eliminated in the method given.

Moreover, as very large sections of the system are represented graphically on one sheet, it is possible to obtain a far more comprehensive understanding of the whole system than with other methods involving multitudinous calculation spread over dozens of sheets of paper.

Above all, the time required for accurate pipe sizing is less than half of any other known method. An identical procedure would be followed in calculating systems with gravity circulation, except that a more suitable and in some cases exaggerated scale of pressure losses would need to be employed.

Having accurately balanced our example, the sizes are corrected on the planometric sketch to those finally chosen.

Chapter Two

HOT- AND COLD-WATER SUPPLY SYSTEMS

Hot-Water Requirements.

THE design of hot-water supply systems for providing water for bathing or industrial uses, whilst at first glance it might appear to be far simpler than a hot-water heating system, is actually far more difficult owing to the problematical nature of the demand for hot water. Unlike a heating system, even when a basis of calculation has been decided upon there is every likelihood of irregular demands for hot water, some parts of the building calling for less and some more than the average for which the system is designed. Although a basis of design can be employed, it must always depend upon indefinite happenings when the system is installed. This difficulty is, however, no excuse for applying any guess methods in the design, for there is every opportunity for employing logical if not strictly scientific reasoning in deciding upon the proportions of the apparatus.

Before commencing upon the design of any system it is necessary to define as closely as possible the quantity of hot water required at any particular temperature, the purpose for which it will be used, and the time of the day at which it will be required. In addition, the time available for drawing off a specific quantity of water and the period elapsing before further draw-off will take place are points which must be considered. It is in defining these items that we meet the greatest difficulties, for every type of building differs in every respect from another, and often two similar buildings are entirely different in hot-water requirements, through varying requirements of the occupants, these being seldom discovered until the completion of the installation.

Regarding water temperature for different uses, it may be taken that the following figures are representative of usual requirements:

Baths and lavatory basins	100°–110° F.
Shower-baths	80°–90° F.
Kitchen sinks	140°–150° F.
Slaughter-houses	212° F.
Other industrial uses	up to 212° F.

The quantity of water required may be taken to be 50 gallons at 100° F. for baths, 3 gallons at 100° F. for lavatory basins, 8 gallons for sinks at 150° F., 20 gallons at 80° F. for a shower-bath. For industrial applications the amount of water required should be discussed with the people concerned, to decide upon the exact requirements of the process.

Variation in Relation of Boiler and Storage Capacity.

With any hot-water supply installation there is the possibility either of having a large storage of water, sufficient to supply the whole of any particular demand period almost without assistance during such period from the boiler, or having very little storage of water and an extremely rapid heating system which could supply the full demand with very little recourse to storage. Between these two extremes there lie many other possibilities with regard to the relation between hot-water storage capacity and heating capacity. Each type of building will differ in regard to the best ratio of these factors, for the period over which hot water is to be drawn off and the time which is available for heating between draw-off periods will decide to a large extent the extreme possibilities.

In order to decide this matter it is convenient to refer to the requirements of hot water in terms of B.T.U. representing that amount of heat required to raise the specified quantity of water from an assumed entering cold-water temperature of 50° F. to the specified temperature at which the water will be used. The B.T.U. equivalent therefore becomes as follows:

	B.T.U.
Baths	25,000
Lavatory basins	1,500
Sinks	8,000
Shower-baths	6,000

These figures represent the quantity of heat required for one filling or use of the particular fitting.

Considering, firstly, an average-size private house with two bathrooms and six bedrooms, the fittings would be as follows:

2 baths
2 sinks
3 basins.

It must be apparent that it is extremely unlikely that all these fittings would be in use at the same time, and in a private house it is found that if the design is based upon the needs of the baths, there is more than sufficient water available to supply the lesser requirements of other fittings, occurring as they do at different periods of the day.

Assuming the six occupants to be in the habit of bathing at one particular period, the total heat required is $6 \times 25{,}000 = 150{,}000$

B.T.U. With two bathrooms in use, therefore, the bathing period is likely to be $1\frac{1}{2}$ hours, and the hot-water supply system, by the combined use of storage water and heating capacity, must be able to supply 150,000 B.T.U. in this period.

With hot-water storage at 150° F., each gallon of storage would be holding $(150-50)\times 10 = 1{,}000$ B.T.U., so that if the storage had to provide nearly the full demand, the boiler might be of such capacity as would provide the maximum demand in 4 hours, that is

$$\frac{150{,}000}{4} = 37{,}500 \text{ B.T.U. per hour.}$$

Let us consider the heat requirements at different stages of the bathing period of $1\frac{1}{2}$ hours.

Initially two baths require $2\times 25{,}000$	=	50,000 B.T.U.
After $\frac{1}{2}$ hour the boiler will have provided		18,700
so that the effective draw-off has been		31,300
Two more baths are now taken, namely		50,000
the effective draw-off then being		81,300
During the second half-hour the boiler gives		18,700
the effective draw-off then being		62,600
The final two baths require a further		50,000
Total B.T.U. to be provided by storage		112,600

The storage required would be approximately $\frac{112{,}600}{1{,}000} = 113$ gallons.

This extreme method of dealing with the problem has the distinct disadvantage in a private house of providing a system very slow in heating up and incapable of rapid recovery after draw-offs in excess of those for which the system has been designed have taken place. It is also at a disadvantage if the usual procedure of burning domestic rubbish is carried out.

As the other extreme, we will consider providing a system capable of providing all but the initial draw-off by heating capacity rather than storage. With this system, the B.T.U. required for the first two baths, namely 50,000 B.T.U., must be available as storage of a capacity $\frac{50{,}000}{1{,}000} = 50$ gallons.

The remaining $150{,}000-50{,}000 = 100{,}000$ B.T.U. must then be provided in 1 hour, and the boiler would therefore heat up the 50 gallons storage in $\frac{50{,}000}{100{,}000} = \frac{1}{2}$ hour. With this method, there would be great difficulty in keeping the fire sufficiently low during non-draw-off periods, without it going out entirely, and similarly when the sudden hot-water demand is made it is unlikely that the stoking would be such as would enable the rated boiler output to be obtained.

It is therefore desirable with a private house hot-water supply scheme to arrange for the boiler to be capable of providing the total B.T.U. required during the maximum demand period in approximately 3 hours. In the example considered this would amount to a rated capacity of $\frac{150,000}{3} = 50,000$ B.T.U. per hour.

The storage required is, therefore, $\frac{150,000-50,000}{1,000} = 100$ gallons, and the boiler is capable of heating this in 2 hours.

Where, however, considerably larger systems are being considered, such as an hotel with 1,000 bathrooms, each bedroom having a bathroom, a different line of reasoning needs to be applied.

For such a case, the total heat required would be

$$1,000 \times 25,000 = 25,000,000 \text{ B.T.U.}$$

Although each bedroom might be equipped with its own bathroom, it is almost impossible in practice to find more than half of the baths used over any particular half-hour, and, indeed, it would be good practice to assume that two-thirds of the maximum demand would take place over 1 hour, that is, $\frac{25,000,000 \times 2}{3} = 17,000,000$ B.T.U. approximately. The heating capacity of such a system should be capable, as far as calorifier heating surface is concerned, of providing this amount of heat in 3 hours, that is 6,000,000 B.T.U. per hour approximately.

The storage capacity would then need to be 17,000,000 less the heat provided by calorifiers in the half-hour elapsing between the first and second draw-off periods, that is $17,000,000-3,000,000 = 14,000,000$ B.T.U., requiring 14,000 gallons storage at 150° F.

The storage could therefore be heated in $\frac{14,000,000}{6,000,000} = 2{\cdot}3$ hours.

On this basis of design, we can see that the possibilities in connexion with 1,000 baths being used over a continuous period are as follows:

	B.T.U.
Heat in storage	14,000,000
Total requirements	25,000,000
Deficit	11,000,000

Therefore, to provide this amount of heat would require the heating system to work for $\frac{11,000,000}{6,000,000} = 2$ hours approximately, and the period over which bathing would need to be spread would be $2\frac{1}{2}$ hours.

In view of the diversity of people in an hotel of the size considered and the many ways in which their days are likely to be spent, this basis of design can safely be employed.

Taking yet another example, this time for a hospital colony, in this instance we would be able by consultation with the hospital officials to determine the periods allocated to bathing and the number of patients dealt with in that period. Schemes of this nature should be designed to provide comparatively small storage, capable of being heated by the calorifier equipment in a maximum of 1 hour. In view of the general nature of the boiler plant, usually capable of dealing with fluctuating demands, this procedure is not likely to result in uneconomical design, for the heating power required for the sudden hot-water supply demand will be required for other purposes when this demand does not exist, and the mechanical firing of the boilers, with either solid, liquid, or gaseous fuel, under thermostatic or steam-pressure control facilitates the operation.

Pipe Sizing for Specific Flow.

If there is a difficulty in deciding upon the possible demands for hot water and the periods during which it is required, there is an even greater problem in determining the sizes of distributing pipes.

In this case we have to decide upon the rate of flow for each particular class of draw-off and the number of draw-off points at any part of the building which will be in use at one moment. Obviously, if the system has only two draw-off points the minimum proportion of draw-off can only be 50 per cent., but as the system increases in size and the draw-off points are multiplied the risk of all taps being opened at the identical moment decreases. It has been the practice, in deciding the rate of flow for the various pipes, to assume that anything from 20 to 50 per cent. of the draw-off points would be in use at one time, according to the nature of the building and the total number of draw-off points.

The author prefers, however, to determine the proportion of the total number of draw-offs on any particular branch, which is to be used in calculating the capacity required in the various pipes, by means of an expression which takes into account also the head available for causing flow and the distance of the draw-offs from the cold-water tank. The expression advocated is the following:

$$Q_a = 5+\sqrt[3]{\left(\frac{T}{H}\times N\right)},$$

where Q_a = average quantity of water to be used for pipe sizing in gallons per minute,

T = travel in feet, measured from cold-water tank to tap,

H = height of water level in tank above tap, in feet,

N = number of baths on branch.

Where the fittings concerned are not baths, but basins, sinks, etc., these must be converted to equivalent numbers of baths by multiplying by the following factors:

Basins	0·5
Showers and sinks . . .	0·75

Considering the small system illustrated in Fig. 37, having 3 baths and 2 sinks, it is likely that the total possible draw-off if all points were in use at the same moment would be:

3 baths @ 5 gallons per minute =	15 G.P.M.
2 sinks @ 4 ,, ,, ,, =	8 ,,
Total	23 ,,

Let us see now what would be the effect of applying the expression just mentioned.

For the 3 baths, with the tank height $H = 4$ ft., and travel $T = 50$ ft., we have

$$Q_a = 5+\sqrt[3]{\left(\frac{50}{4}\times 3\right)} = 8{\cdot}4 \text{ G.P.M.}$$

As the maximum possible is 15 G.P.M., this represents $\frac{8{\cdot}4}{15} = 56$ per cent. capacity.

Similarly, for the 2 sinks, representing $2\times 0{\cdot}75 = 1{\cdot}5$ baths, with $H = 12$ ft. and $T = 67$ ft., we have

$$Q_a = 5+\sqrt[3]{\left(\frac{67}{12}\times 1{\cdot}5\right)} = 7{\cdot}0 \text{ G.P.M.}$$

This represents $\frac{7{\cdot}0}{8} = 87$ per cent. capacity.

In determining the capacity of main pipes these figures are used, the quantities for main pipes being as shown on Fig. 37.

The delivery of water from taps depends entirely upon the head of water represented by the height of the cold-water feed tank above the taps, and the distance which the water has to travel to reach the taps. In fact, the problem is very similar to the distribution of water for gravity hot-water heating, except that the head causing flow exists by virtue of the position of the cold feed tank, instead of by a difference in weights of opposing columns of water.

We have seen how, in designing the pipes for a heating system, they are firstly sized on an average pressure loss available for the radiator having the lowest value of circulating pressure divided by travel, resulting in a necessity for reducing some pipes in size where the travel is less or there is a surplus or unconsumed pressure. In the design of pipes for a hot- or cold-water draw-off system account should be taken of these factors, but the use of the expression for finding Q_a, resulting as it does in lower percentage capacity for draw-offs having lower values of T/H, naturally results in practice in the same or substantially similar results as if the whole series of

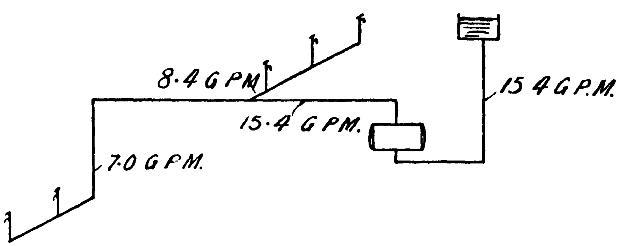


FIG. 37. Simple H.W.S. distribution.

branches to taps were sized on the same percentage capacity and those nearest the cold-water tank (having least travel) had smaller pipes to consume surplus pressures.

This method enables all draw-off and main pipes, once a value of Q_a has been decided, to be sized for the pressure loss available for that tap having the longest travel compared with the least tank head, that is, the highest value of T/H.

Delivery of Water from Pipes.

The delivery of water from the taps of a hot-water supply system is dependent upon the head available for causing flow which, unless resort is had to artificial means of increasing this, is provided by the pressure due to the height of the cold-water feed tank above the outlet or tap. This head or pressure is available for producing flow and causing a delivery of water from the tap, and it should be clearly understood that the underlying principle of the design of piping systems for hot-water supply is to provide pipes which will consume this available pressure when the design basis quantities of water are being delivered. It is therefore immaterial whether the pressure loss is divided equally over the whole length of pipe from the cold-water feed tank to the tap or is greater in some lengths than others, provided that the available head for each and every tap is not exceeded by that actually consumed to that tap.

The author finds the following formula, due to Neville, proved to be suitable for use in connexion with these problems:

$$v = 140\sqrt{(RS)} - 11\sqrt[3]{(RS)},$$

where v = velocity in feet per second,

R = mean hydraulic depth in feet $= \dfrac{\text{area}}{\text{wet perimeter}}$

$= \dfrac{d}{4}$ for circular pipes,

S = sine of slope $= \dfrac{H}{T}$,

H = head in feet,

T = length of pipe in feet,

d = diameter of pipe in feet.

To simplify the application of this formula to practical problems the nomographic calculator in Fig. 38 has been designed, enabling the carrying capacity of various sizes of pipes to be determined, in gallons per minute for different values of T/H, within the extremes likely to be reached in practice. The formula given refers to the flow of water in iron pipes, but in view of the increased use of copper and similar smooth tubes, the calculator is adjustable accordingly.

Investigations conducted by the Copper and Brass Extended Uses Council have determined the relative loss of head due to friction of water in iron, copper, and brass tubes to be as follows:

Iron = 1·0.
Copper and brass = 0·81.

For the same frictional resistance a copper or brass pipe would pass 1·11 times as much water as an iron pipe of the same internal diameter. The actual figures vary slightly with the diameter of the pipe, but it is sufficiently accurate for design to assume 11 per cent. increase, on which basis the nomographic calculator has been designed.

The difference in delivery comparing hot with cold water is so little as not to warrant taking into account, so that the nomographic calculator can be applied equally well to problems involving the sizing of cold-water services.

Circulation of Hot Water.

In all but the very small system, the piping arrangements are such as to enable circulation of hot water to take place throughout the system. It is therefore necessary to be able to design the piping system not only to be capable of delivering certain quantities of water at the taps but also for this water to be circulated without undue loss of heat.

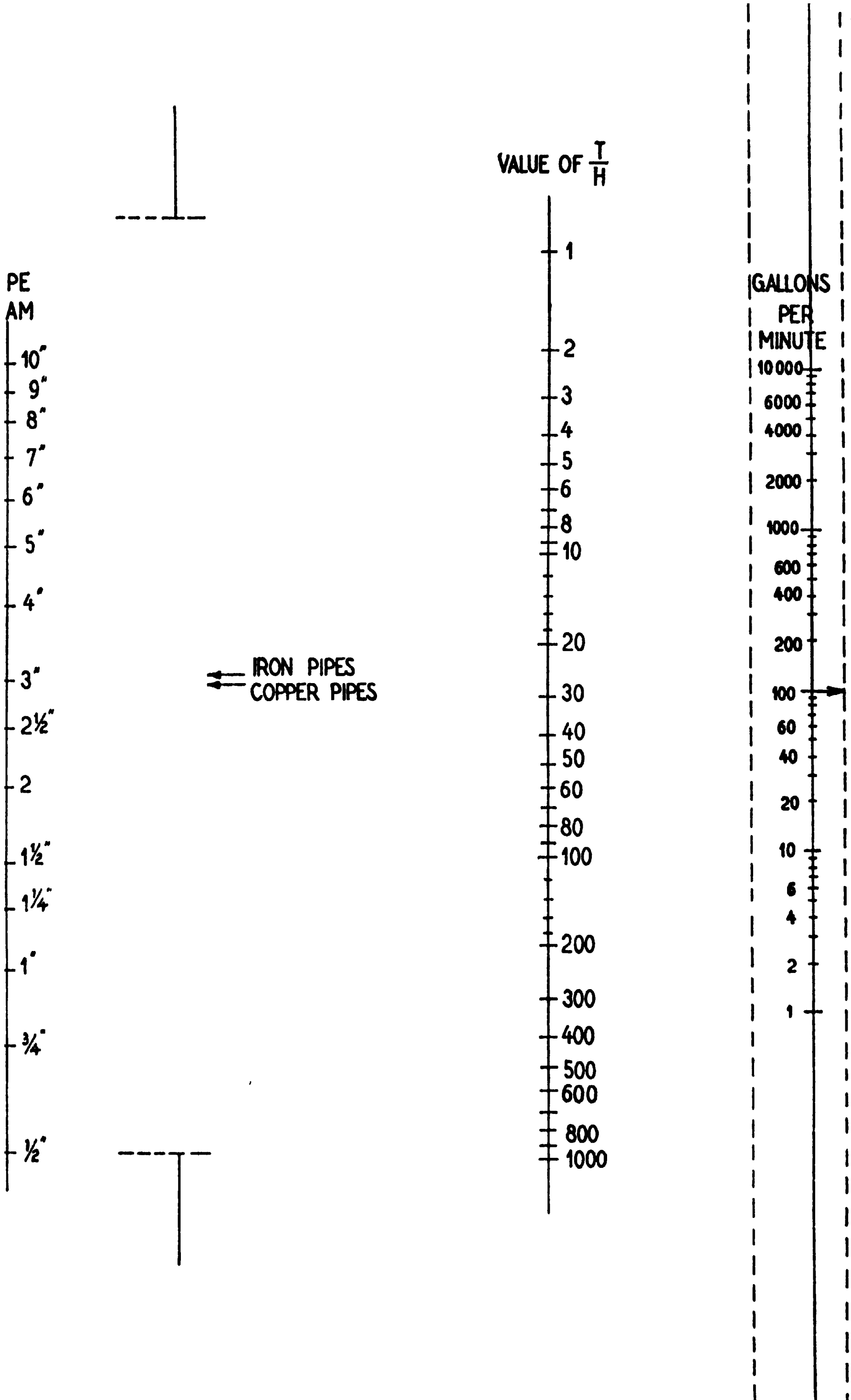


FIG. 38. NOMOGRAPHIC CALCULATOR FOR HOT-WATER SUPPLY PIPES

Having, therefore, decided upon sizes adequate for delivery at taps, the next step in design is to decide upon the quantity of heat given off by the pipes for the sizes determined, assuming any return pipes to be of similar size to the flow, the necessary quantities of heat being decided for every pipe in the system. At the same time, losses from coils in linen cupboards, towel rails, and similar apparatus are added in. The circulating pressure for various sections of pipe being determined by the methods given in Chapter I for a maximum temperature drop in the system of 10° F., and the resistance of the flow sections of the piping, which of course are fixed in size according to the delivery of water required at the taps, having been found, it remains only to choose return pipes of such size as will consume the remaining pressure for each circuit. Subsequent examples will explain this more fully.

Where the installation is of such large proportions that the length of pipes is out of all proportion to the pressure available for circulation, it becomes necessary to resort to pump accelerated circulation, the temperature drop in such cases being as low as 5° F., and the pump head 5 to 25 ft. With this system it is possible to have very large flow pipes and returns many sizes smaller.

Design of Piping System for Small Building.

As a simple example in the detailed application of the proposed methods of calculation we will deal with the system illustrated in Fig. 39, which is intended to serve 30 baths.

It will be seen that the general arrangement of this system is two-pipe underfeed, the baths being grouped on 6 risers. For the purpose of finding the value of Q_a each riser will be considered as a group of draw-off points. The calculation of Q_a for each riser then becomes as follows:

Riser	T	H	T/H	N	Q_a
A	118	11	10·7	5	8·8
B	129	11	11·7	5	8·9
C	139	11	12·6	5	9·0
D	148.	11	13·5	5	9·1
E	157	11	14·3	5	9·1
F	166	11	15·1	5	9·2

These values of Q_a are those for the bottom sections of the risers, and by adding back to the main feed the requisite values are obtained for all flow pipes, as indicated.

The sizes of pipes are now to be determined by use of the nomographic calculator, the cold feed pipe being taken on the basis of the sum of Q_a for all risers. In choosing sizes, the size larger is to be chosen if the capacity cannot quite be obtained on any particular size.

Knowing the size required for that portion of the flow riser having the greatest capacity, the sizes between this and the top tap, which would be sized for a flow of 5 G.P.M., can be decided empirically by gradation.

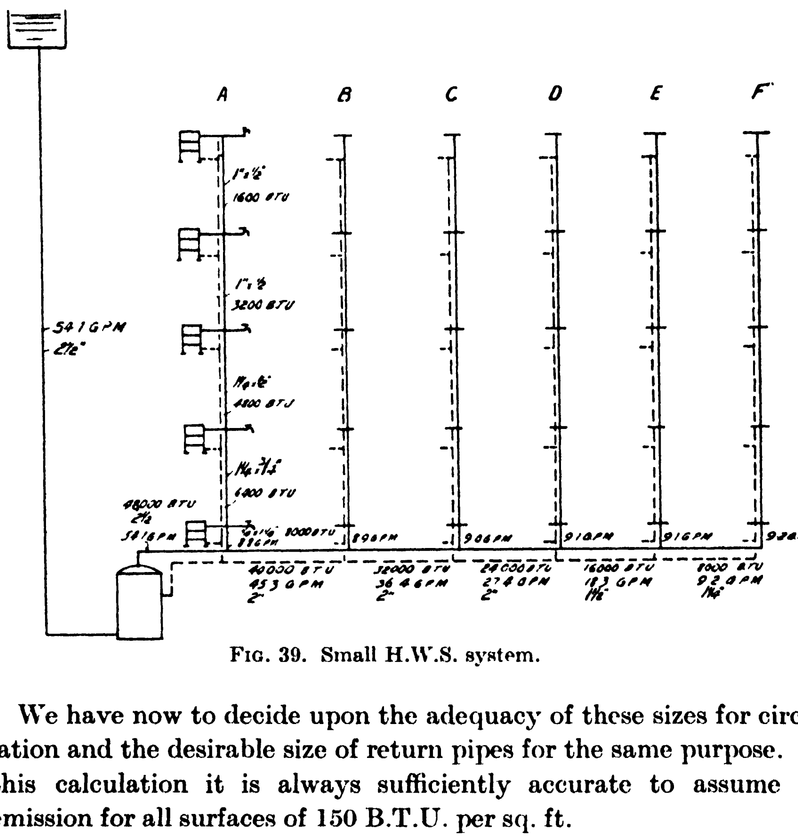


FIG. 39. Small H.W.S. system.

We have now to decide upon the adequacy of these sizes for circulation and the desirable size of return pipes for the same purpose. In this calculation it is always sufficiently accurate to assume an emission for all surfaces of 150 B.T.U. per sq. ft.

The total lengths of various sizes of flow pipes together with the approximate heat emission are tabulated as follows:

Diameter	*Length*	*Sq. ft.*	*B.T.U. Heat loss*
in.	ft.		
2½	10	7·5	1.130
2	30	18·6	2,800
1½	9	4·5	680
1¼	130	56	8,400
1	110	36·3	5,450
		Total	18,460 B.T.U.

30 towel rails = 120 sq. ft. 18,000

For approximate purposes the loss of heat from return pipes may be taken as 60 per cent. of that from the flow, namely,

$$18{,}460 \times 0{\cdot}6 = 11{,}000 \text{ B.T.U.}$$

The total loss is therefore $18{,}460+18{,}000+11{,}000 = 47{,}460$ B.T.U., and this must be approximately divided between the six branch circulations, that is 8,000 B.T.U. approximately to each.

From these figures the totals for the main circulation and its various sections can be determined and noted on the diagram.

In deciding now upon the sizes of return pipes, it would be pedantic to attempt to size the different risers according to the distance they are from the cylinder, in view of the fact that the whole system is so small. For determining the circulating pressure, the flow temperature will be taken as 150° F., if possible with a temperature drop of 10° F., the available pressure per foot of height being found from Fig. 2 to be 0·036 in. of water. The available pressures for the various flows are as follows:

Floor	*Height*	*Available Pressure 10° drop*	*Available pressure 20° drop*
	ft.		
Grd.	4	0·144	0·28
1st	13	0·47	0·91
2nd	22	0·79	1·54
3rd	31	1·12	2·17
4th	40	1·44	2·80

The next step is the calculation of losses in the flow pipes up to the base of the farthest riser, by the aid of the nomograph, Fig. 1, tabulated below.

Size	*Approx. length*	*B.T.U.*	*Loss of pressure*
in.	ft.		
2½	16	48,000	0·18
2	10	40,000	0·27
2	10	32,000	0·18
2	10	24,000	0·11
1½	10	16,000	0·22
1¼	14	8,000	0·17
Towel rail			
¾	8	600	0·01
		Total	1·14 in.

It is obvious, therefore, that the system would not in itself circulate at a sufficiently low temperature drop, with all pipes bare, for the loss in the flow alone exceeds that available for flow and return.

If all circulating pipes are insulated it is likely that the loss of heat from them will only be one-third of the loss when bare, so that the total loss could thereby be reduced to 27,500 B.T.U. approximately. In this case the total loss in the flow pipes would be

$$\left(\frac{27{,}500}{47{,}460}\right)^2 \times 1{\cdot}14 = 0{\cdot}39 \text{ in.}$$

Even now the loss is too great, but if we decide to allow a 20° drop this loss becomes $\left(\frac{10}{20}\right)^2 \times 0{\cdot}39 = 0{\cdot}10$ in. approximately.

The circulating pressure for 20° drop, in the case of the lower towel rails would become, however, $0{\cdot}07 \times 4 = 0{\cdot}28$ in., so that $0{\cdot}28 - 0{\cdot}10 = 0{\cdot}18$ is available for return pipes, having a total length of approximately 80 ft., allowing an average loss of $\frac{0{\cdot}18}{80} = 0{\cdot}0023$ in. per foot. The sizes of returns for this loss are ascertained from the nomographic calculator, Fig. 1, being actually similar to the flow diameters. For the vertical returns, owing to the higher circulating pressures, it is likely that they will be considerably smaller in diameter than their respective flows.

Still considering riser F, for which the B.T.U. of the various sections have been corrected for the revised conditions for insulated pipes, we may calculate in a similar way the additional losses in the flows to the various floors, adding these to that for the main flow and returns, to determine the total pressure available to be consumed in each particular section of return. These data are tabulated as follows:

Size of flow	*B.T.U.*	*Length*	*Available for return*	*Size of return*
in.		ft.		in.
$1\frac{1}{4}$	4,000	10	0·618	$\frac{3}{4}$
$1\frac{1}{4}$	3,000	10	0·623	$\frac{1}{2}$
1	2,000	10	0·62	$\frac{1}{2}$
1	1,000	10	0·63	$\frac{1}{2}$

In the case of the towel rails on the higher floors the diameter of connexions may accordingly be reduced to $\frac{1}{2}$ in.

This example has been of use in demonstrating that with systems of the type considered the difficulty is to obtain adequate circulation to the towel rails on the lowest floor, but for upper floors, return pipes several sizes smaller than the flow pipes are adequate. The example taken is, moreover, more compact than a system serving 30 baths would be likely to be in practice, so that the difficulty with the lower floor may sometimes be even greater than here shown. In such cases, the advisability of mechanical or accelerated circulation is apparent.

Design for Hotels and Blocks of Flats.

There are several special problems peculiar to hotels and large blocks of flats, relating to the position and apportionment of storage capacity and the general means of distributing hot water to the taps, which are worthy of consideration.

There have been many discussions in particular regarding the question of whether the hot-water storage should be concentrated at one central point adjacent to the heating equipment, or should be divided into units situated at different points to deal with the local requirements. The method in general use at one time, of running steam to small calorifier units, has little to commend itself, as it is without doubt an uneconomic system of distribution. The only occasions when this method should be employed are those where blocks of buildings are situated at a considerable distance from a large steam plant, in which cases distribution of heat by steam would perhaps be found more convenient and cheaper in capital cost.

With hotels and blocks of flats there is much to be said in favour of providing a proportion of storage at the heating point, the remainder being situated at high points over the building. It may be argued that this method is open to the risk of having storage available at points where it may not be of use during a heavy demand in another part of the system, but it is in the case of those buildings where the head of water provided by the cold-water storage tank is insufficient to give adequate discharge without unduly large pipes that localized storage of hot water is advocated. In such cases the hot-water storage vessel provides its own head of water, and the travel to be taken into account in sizing the delivery pipes to the taps in any particular section is only to be measured from its respective storage vessel.

Such a method of distribution is illustrated in Fig. 40.

There is another point of special interest with hot-water distribution in flats and hotels, namely the method of arranging pipe systems. If the designer prefers to centralize hot-water storage at the heating point, and there is no technical reason why a plant so designed should not give satisfactory results; almost any general system of distribution may be employed. If the two-pipe underfeed system is used, it is usual to depart from the general practice of connecting all draw-off points to the flow pipe, and to arrange for a proportion, perhaps a third, to be connected to the return, on the assumption that use is then being made of the return pipe for feed purposes; for on opening taps it is not possible to ensure the whole of the delivery taking place through the flow pipe, unless a non-return valve be fitted on the main return.

If all draw-off points are connected to the flow it is desirable that such a valve be fitted.

Another method of distribution usually found to be the cheapest, and certainly effective, is the one-pipe drop system, of particular value where ranges of baths or other draw-offs are repeated on several floors at similar positions. With this system use can be made in design of the fact that draw-off will take place from the flow (top) and return (bottom) ends of the drop pipes, and a non-return valve should not be used.

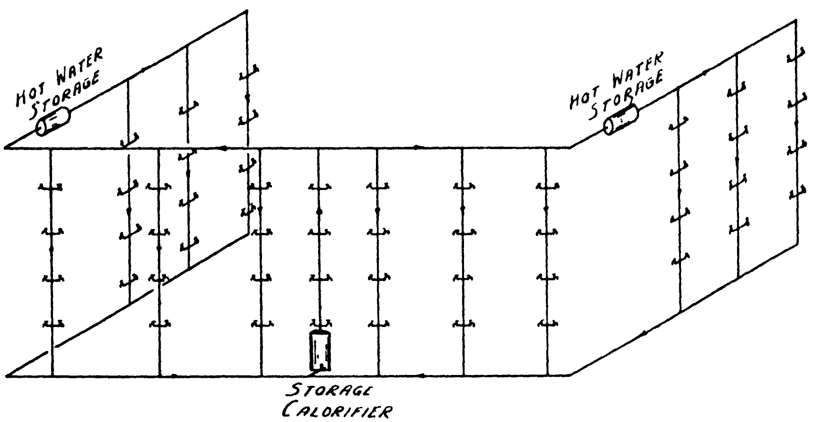


FIG. 40. Large H.W.S. system with split storage.

It is this system of distribution which is usually applied to installations having hot-water storage vessels in the roof of the building, as may be seen from Fig. 40.

Although reference has already been made to the theoretical considerations affecting storage capacity it should be noted that the usual capacity provided in practice varies between 12 and 30 gallons per bath, basins and sinks being ignored, and the boiler plant being sufficient to heat up the whole storage in 2 to 4 hours, proportionate to the storage allowed. That is, if only 12 gallons of storage per bath is allowed, it should be heated in 2 hours, whilst the heating-up period would increase with a higher storage.

Design for Hospitals and Mental Colonies.

With buildings in the nature of hospitals or mental colonies each particular installation calls for special treatment which can only be decided upon the special circumstances attached to the particular job.

Dealing firstly with hospitals, it will be appreciated that these may vary considerably in planning as far as building layout is concerned and requirements for hot water. Invariably it will be better with a compact building or group of buildings to centralize hot-water storage, and a fundamental requirement is that rather than provide large storage, rapidity of heating up should be a primary consideration. Similarly, the need for accurate thermostatic control will be obvious, for hot-water demands are made at intermittent and well-known periods. A consultation with the hospital authorities will always provide sufficient data upon which to base design.

With mental colonies, however, there is scarcely one which does not consist of many isolated blocks such as administration, villa blocks for patients (particularly if of modern design), boiler house and kitchen block, and various semi-industrial blocks. For such installations the wisest design provides for separate hot-water storage in each block, the central station consisting of hot-water calorifiers from which mains are run to each block, to connect to the storage vessel.

In cases where it is decided that it is desirable to have all hot-water storage at or near the heating point, there is need to decide upon the general method of arrangement of which the following alternatives are available:

(1) Heating calorifiers with separate storage vessels;

(2) Storage calorifiers;

(3) Storage calorifiers with heating calorifiers to deal with peak load requirements.

Of these three methods there is nothing in distinct favour of using any one in preference to another, but the author suggests that method (2), the use of storage calorifiers, is one which generally lends itself best to a compact arrangement of the whole of the plant. For buildings of the nature of those being discussed, a central steam boiler plant should be provided and calorifiers should be arranged either with double heating batteries, one of which should be served by exhaust steam from turbines driving pumps and other units, or it may be so arranged that the maximum hot-water demand is provided by several calorifiers of which one or more is served only by exhaust steam.

Combined Mains for Heating and Hot-Water Supply.

Although for many years it has been general practice to provide separate calorifiers and main distributing pipes for heating and hot-water systems, a practice which in the small installation becomes almost essential, the large plant for mental colonies offers a fine opportunity for the application of combined distribution systems. In the past, the only serious objections to employing such systems have been either the fact of pipes becoming furred or the risk of disturbing heating circulation when hot water is being drawn off from the system. We will discuss these objections in order to see how they may be overcome. In the first instance the question of furring of pipes is one which, even with separate hot-water supply systems, still exists and is likely to prove of real trouble unless soft water is available.

With regard to the possibility of circulation to radiators being disturbed by water being drawn off from the system, we can only say that whilst theoretical considerations may lead to objections, several plants on a large scale have proved that the use of combined distributing mains is practicable. It must not be thought that it is possible to arrange for draw-off points to be connected indiscriminately to the heating system, for this procedure has been proved useless. If, however, it is arranged that mains common to hot-water heating and hot-water supply services are run from a centralized calorifier station to the various blocks, there splitting into two separate and independent systems one to serve the heating and the other the draw-off system, no trouble will be experienced.

Even if a temporary stoppage of heating circulation occurs on heavy draw-off, it should be appreciated that this disturbance is taking place for only a few minutes, and as circulation is in such systems always by means of a pump, rapid recovery of circulation is certain. Moreover, such disturbance for a few minutes at intervals is insufficient to affect in any way the temperature of the rooms being heated.

In districts where the water supply is naturally soft the only precaution which needs to be taken is to carry out all pipework in materials suitable for resisting corrosion. In such cases, this remark applies only to the common mains and the separate hot-water supply system within the blocks. The separate heating systems within the blocks need not have piping suitable for resisting the corrosive action of soft water, for provided the system is arranged substantially in accordance with that in Fig. 41, the water in the heating system will remain unchanged in spite of draw-off from the separate internal hot-water system.

In hard water districts it is essential that a water-softening plant be provided to soften all water entering the system, to prevent furring up. In any case, with hospitals and similar institutions in hard water districts it is usual to provide water-softening equipment.

We are faced now with the problem of designing the common distributing system, which must be suitable for circulation for heating requirements and yet of such dimensions as will satisfy draw-off.

It is not possible to define a rule which will give the answer to this problem, for every installation which is to be designed will differ greatly as regards the relation of hot-water supply to heating requirements. Similarly, the head provided by the cold-water feed tank is that causing delivery of water from the taps, whilst on the other hand the head or pressure causing circulation is flexible within a wide range according to the duty of the accelerating pump.

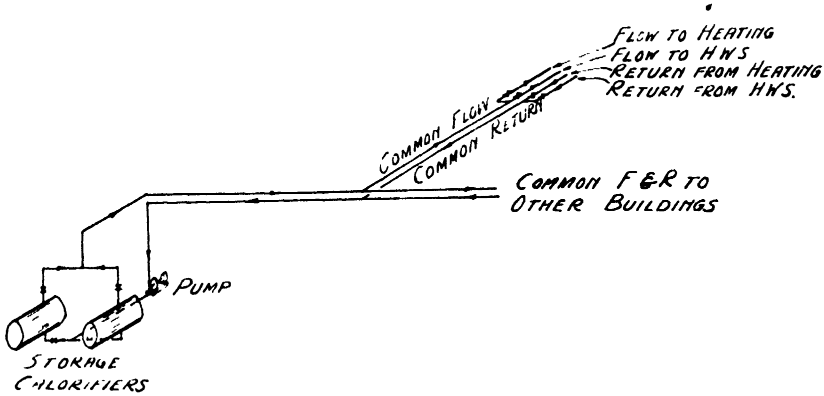


FIG. 41. Heating and H.W.S. with common mains.

It therefore becomes necessary with installations of this type to decide firstly upon the approximate dimensions of the main pipes to give adequate delivery of water from the taps, with the head available. Having done this, we are then in a position to determine whether the sizes so found are sufficiently large to permit of circulation of enough water for radiator heating, and if they should not be so they must accordingly be increased in diameter.

There are several points in distinct favour of employing combined mains and calorifiers, chief amongst which are saving in initial and maintenance cost of mains, and considerable economy through decreased heat loss from them.

In deciding upon the heating power of calorifiers to serve the combined system, the necessary power will be the sum of the maximum heating B.T.U. and that which would normally be required for hot-water supply services if served by separate calorifiers.

After studying the possibility of the combined or common main system, the advantages to be gained by sizing pipes on the reducing pressure loss system, outlined in Chapter I, will be more apparent.

Pressure Boosting for Large Systems.

Instances arise in practice where existing cold-water feed tanks give insufficient head to produce adequate delivery at all taps, owing perhaps to long travel and, in the case of isolated buildings, large differences in ground levels. The normal procedure has been to increase the height of the tank, but often structural reasons do not permit of this.

For the larger systems in particular, the use of a pressure-boosting pump capable of maintaining a constant pressure in the system of 20–50 lb. per sq. in. is a practical proposition. There are several means by which this may be put into effect. One method consists of a pumping set comprising the usual accelerating pump and a further centrifugal type of pump, both driven from a common motor situated between the two, the whole being mounted on one base plate. The cold-water feed connexion from the supply tank is taken to the second or booster pump and delivered by it to the hot-water system. This booster pump would be in operation continuously, with the accelerating pump.

With most large installations, however, it is advantageous to employ instead a separate steam-driven reciprocating pump of the type usually in service for boiler-feed purposes, and this pump is arranged to maintain a predetermined pressure in the system and controlled so that on this pressure being reached the steam supply to the pump is shut off and the pump becomes inoperative, only to start again when the pressure in the system falls, which would be caused by taps being opened. The flexibility of this type of pump and the ease with which the maximum pressure may be varied by control of the steam supply make it most suitable for the purpose of pressure boosting.

There is perhaps a tendency to apply pressure-boosting pumps indiscriminately without due thought as to the actual increased head required to produce adequate water delivery, but this should not be so, for calculation is extremely simple. If from an inspection of the nature of an installation with combined mains it is definite that intensive pressure boosting will be required, the wisest course to take in designing the system is firstly to size pipes for heating requirements. Having done this, the quantities of water required to be delivered by all pipes being known, it is possible by reference to the hot-water supply pipes nomographic calculator (Fig. 37) to take each length of delivery pipe on the longest run, to read off from the calculator the value of T/H which would be required to give the necessary flow through the pipe of the diameter in question. If this value is then divided by the length of the pipe the result is the head in feet required to produce such flow, and the sum of these separate heads consumed in the longest delivery circuit indicates the total head in feet which is required of the booster pump.

As the quantities of water decided upon for the various pipe runs will not represent the total required if all taps were in use simultaneously, it is wiser to instal the booster pump capable of providing, if required, double the calculated pressure when delivering a maximum of twice the total quantity of water calculated to be required under normal conditions.

Chapter Three

VENTILATION

Kitchen and Restaurant Ventilation Systems.

Ventilation systems applied to kitchens are as diverse as the kitchens themselves, ranging as they do from a small room containing a gas cooker to larger organizations capable of providing thousands of meals each day. Whilst few modern kitchens of any size have so important a factor as the ventilation neglected, the kitchens of the older hotels and restaurants have for many years been unbearable, due to faulty ventilation equipment, if indeed any had been applied.

Apart from any question of comfort for kitchen staff, the fundamental of kitchen ventilation has always been the necessity of keeping kitchen smells strictly to the precincts of the kitchen, for there is nothing calculated so to spoil a good appetite as a premature smell of a mixture of foods.

In many cases the need for keeping kitchen smells from adjoining rooms immediately shows the necessity of considering the ventilation of a kitchen in conjunction with that of the surrounding parts of the building.

It is scarcely possible to give in an orderly manner the different types of plant with which one must deal, but firstly we will refer to a typical large restaurant and its adjoining kitchen and discuss the ventilation problem.

If we were considering the design of air-conditioning equipment for theatres and cinemas, the matter would be comparatively simple owing to the fact that any cinema or theatre is designed to provide a definite amount of space per seat, but restaurants and dining-rooms vary so considerably that no two can safely be designed on the same basis and each must, therefore, be considered strictly on its own merits. For the smaller type of dining-room seating possibly 100 people with an adjacent kitchen, nothing is actually required in the nature of air-conditioning equipment, the problem in these cases merely resolving itself into adequate ventilation of the room, care being taken that there is a definite flow of air arranged to pass from the dining-room into the kitchen.

When, however, we get to the large restaurants seating possibly 1,000 people, the problem becomes difficult. It must be remembered that with all restaurants, wherever possible, the building owner tries to utilize either a basement or some other parts of the building which cannot be usefully employed for other purposes, and invariably these parts are found to be the most difficult to deal with from the air-conditioning and ventilation point of view; one reason being that they are particularly badly placed with regard to access to the external atmosphere and in many cases have a large floor area and very low ceilings, presenting difficulties in air distribution.

Restaurants may be classified under several general headings. We have in the first instance the large hotel restaurant or dining-room which, although it may have a large seating accommodation, nevertheless allows a floor space of anything up to 20 sq. ft. per person and is invariably very lofty with ceilings 20 to 25 ft. high. In this type of restaurant, therefore, not only is the building more suitable for the introduction of air-conditioning apparatus but at the same time the air-conditioning problem is easier because the amount of heat which must be removed compared with the volume of the room is also lower. Moreover, patrons are more inclined to remain for a long period for meals.

The popular type of café or restaurant, whilst in many cases still providing for a seating capacity of possibly 1,000 persons, does not provide for so great a floor space per person; in some cases this does not amount to more than 7 sq. ft. of actual restaurant floor per person. In this type of restaurant people are constantly moving about, and the average time occupied in a meal would rarely be more than 20 minutes.

The closeness with which the tables are arranged in this type of restaurant presents a difficulty in air distribution, for invariably the whole of the wall space is occupied by tables, and the owner is loth to part with any of the centre floor space in order to provide for ventilating inlets or outlets.

With a cinema or theatre system it is invariably known that the building will be almost fully occupied for 8 hours continuously, but with a restaurant of the popular type, the first heavy demand would exist between 10.30 and 11.30 a.m., representing the 'coffee period', another heavy demand being between 12 noon and 2 p.m. Teas will occupy a further period between 4 and 6 p.m., whilst the dinner period would occur between 7 and 9 p.m., practically running into the period for suppers until midnight or later. It may be seen, therefore, that there is an intermittent demand for meals between 10.30 a.m. and midnight, with very brief periods in between so that if, in the consideration of the operation of the air-conditioning equipment, care is to be taken to ensure economical running, very close observation or control is required to ensure that the plant is not run wastefully during the quiet periods.

Another factor which requires consideration with this type of restaurant is the fact that there may possibly be three restaurant floors, each wholly or partially independent of the others, in which case the economy problem becomes simpler, for it is usual at certain periods to close down one or more floors to concentrate service rather than to allow a straggling service over the whole of the building. Where this procedure is followed, control of the air-conditioning equipment becomes relatively simple, as the complete plant could be arranged in such manner as to be split into separate units serving each restaurant floor.

Again, apart from any question of the engineering side of the air-conditioning equipment, architectural features are of considerable importance to this class of building, as indeed they should be in any modern building, but unfortunately the restaurant does not offer the same opportunities that the lofty type of building would do, and therefore calls for very close co-operation between the engineer and architect in the preliminary stages in order to evolve an air-conditioning scheme that is not only efficient in dealing with the atmosphere of the room, but at the same time is, from the layman's point of view, invisible. A further point which is even more vital is that draughts must be non-existent, and that no noise emanating from the air-conditioning and ventilating equipment should be heard in the restaurant. It might be argued that noise would not be of much importance in a popular type of restaurant owing to the noise of general conversation, but on the other hand, during the quieter periods the few patrons then in the building would invariably be of such a type as would notice the slightest irritating noise.

Before proceeding with the detailed design for a typical restaurant we must review briefly the various items which must be considered in connexion with air conditioning and ventilation in large buildings of this type.

In its fundamentals, whatever equipment is employed must be capable of dealing not only with the heating of the restaurant during cold weather, but at the same time, an even more important point, the cooling during extreme summer conditions. This cooling, as will be shown later, is not only necessary during warm weather but to a large degree is of importance during the whole of the time that the restaurant is partially or wholly filled with patrons. In addition to these obvious problems we have the others of determining how the heating and cooling effects are to be obtained in a practical and economical manner, having due regard at the same time to the fact that floor space occupied by engineering plant represents a potential financial loss unless this space is devoted to equipment which is operated efficiently.

Our problem should take us also to the consideration of the many means by which cooling may be obtained and the merits of the various systems which can be used, whilst the control of the equipment by automatic means should also receive our attention.

The running costs of the plant must be capable of being set out at a definite price on each meal that is supplied, as they form an overhead charge which must be met before any profit can be shown, just as staff salaries, rates, lighting, and other charges.

A further problem which will lead to indefinite controversy with every job is the distribution of the air in the building, and a decision must be arrived at as to whether it is possible to obtain either theoretically efficient upward or downward ventilation or whether cross ventilation must be employed, and if none of these systems is possible a practical combination must be evolved to meet the circumstances. The large number of air inlets and outlets that must be provided suggest possible collaboration with the lighting engineers in order to produce some combined form of fitting which will be efficient from the points of view of both lighting and air-conditioning experts.

In a restaurant building, moreover, it is impossible to ignore the requirements of the kitchen ventilation system, and the two systems, whilst being practically independent as far as equipment is concerned, are nevertheless closely allied in connexion with air quantities or rate of air change; and one of the first problems that must be decided is the relation between the quantities of fresh air and extracted air in the kitchen and restaurant respectively, in order to eliminate the slightest possibility of kitchen smells or vapours entering the restaurant. Invariably with the type of restaurant which we are to consider the kitchen and service is immediately adjacent to the dining-room, so that any basic fault in design would ruin the complete scheme.

Example in Design of Restaurant Air-Conditioning Equipment.

In order adequately to illustrate every phase of the fundamental arguments and detailed calculations in connexion with the design of the air-conditioning and ventilating plants, we will take as an example the large restaurant and kitchen illustrated in Fig. 42. The restaurant in this case is approximately 100 ft. long and 70 ft. wide, the height being 16 ft., so that the total floor area is 7,000 sq. ft., and the cubic contents of the room 112,000 cu. ft. The space devoted to kitchen equipment and service is approximately 75 ft. by 75 ft.

by 16 ft. high, the floor area being 5,600 sq. ft., and the cubic contents 90,000 cu. ft. This restaurant, it will be assumed, has a seating capacity of 1,000 persons, and it will be taken that it is situated in a basement with fairly easy access all round. In addition, it will be assumed that the roof, several floors above, could be employed for accommodating any desired plant.

Although some County Council Regulations would call for a minimum quantity of air of 1,000 cu. ft. per person for a new building of this type, the nature of the building and the closeness with which

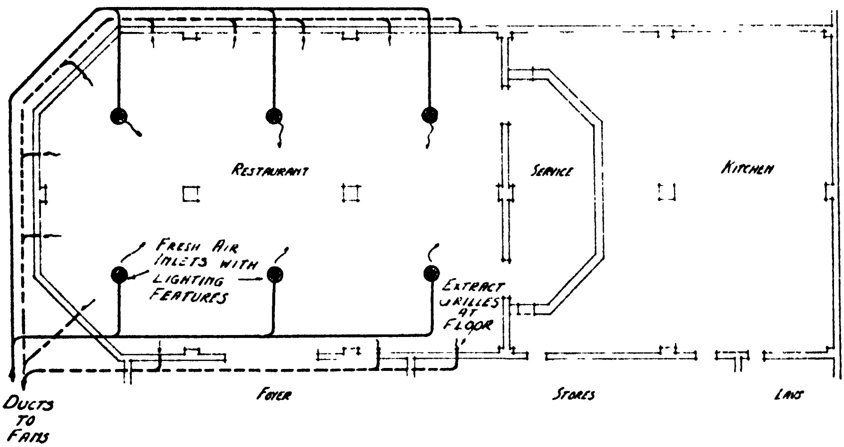


FIG. 42. Large restaurant air conditioning.

the seating accommodation is arranged makes it imperative to employ at least 2,000 cu. ft. of air per person, for unless this is done, as the author has shown,† the temperature at which the air would have to be introduced for cooling purposes would be so much below the temperature required in the building that cold draughts would exist. It will be noticed that we are referring firstly to the question of air volume and cooling, the reason for this being that in a plant of this nature, where heating and cooling will be required on different occasions, if the plant is designed for such a capacity as will best meet the cooling demands it need have no fundamental alteration to make it suitable for heating.

On the basis of air volume suggested, therefore, the amount of air to be provided would amount to 2,000,000 cu. ft. per hour, representing a capacity of 33,300 cu. ft. per minute. We must determine now the fundamental sources of heat tending to a rise of temperature in the restaurant. The first source of heat is that transmitted through

† 'The air-conditioning requirements of public buildings, theatres and cinemas', by A. T. Henly, *Heat and Vent. Engr.*, vol. v, pp. 164 et seq.

the wall adjacent to the kitchen, and it will be assumed that this is only the heat actually transmitted through the building structure owing to the fact that the restaurant is in a basement and is not in contact with the outer atmosphere. The next and largest source of heat is that emanating from the occupants themselves. This in the design of equipment of this nature may be calculated on the basis of 400 B.T.U. per person.

It will be seen that the situation of this particular restaurant is advantageous in the fact that it is not exposed to the rays of the sun, and therefore this source of heat need not be considered.

Another source of heat arises with this building, however, which does not exist with any other, and that is the heat represented by the food which is being consumed. It will be appreciated as far as the design of the air-conditioning plant is concerned, that the maximum conditions for which the plant is calculated exist during the summer months when the temperature and moisture content of the external air would be relatively high. During these months it would not be usual for the same amount of hot food to be consumed, and it is suggested, therefore, that calculations should be based upon an allowance of 100 B.T.U. per person to cover both the actual heat rising from the food and any margin required to cover the amount of moisture emanating from it.

Finally, a further source of heat exists in the lighting units. If we assume an average intensity of lighting equivalent to 2 watts per sq. ft. of floor surface, we find that this amounts to 14,000 watts, which actually is dissipated as heat.

We may, therefore, now sum up the various quantities of heat tending to a temperature rise as follows:

		B.T.U.
(*a*) Transmission through kitchen wall, assuming the area to be $70 \times 16 = 1{,}120$ sq. ft. The coefficient of heat transmission being 0·25 B.T.U. per sq. ft. per degree difference, and the temperature difference between the restaurant and kitchen being 15° F. The amount of heat, therefore, being	$1{,}120 \times 0{\cdot}25 \times 15$ =	4,200
(*b*) Heat emanating from people on the basis of 400 B.T.U. per hour for each of 1,000 persons	$= 400 \times 1{,}000$ =	400,000
(*c*) Heat emanating from food on the basis of 100 B.T.U. per person	$= 1{,}000 \times 100$ =	100,000
(*d*) Heat equivalent of lighting on the basis of 14,000 watts, each watt being equivalent to 3·4 B.T.U.	$= 14{,}000 \times 3{\cdot}4$ =	47,800
Total sensible heat tending towards rise of temperature B.T.U. per hour	=	552,000

Before it is possible to decide how exactly this amount of heat is to be absorbed from the room, we must first determine what internal temperature is to be maintained for the basic conditions on which the plant is to be designed.

We will assume that the plant is to be designed so that under all conditions the interior temperature of the restaurant will in no case be less than 12° F. below the external temperature of the atmosphere. If this difference were greater the feeling of chill experienced by anybody entering the restaurant would invariably lead to some complaint, and apart from this would be undesirable from a medical point of view.

It will be decided that the plant must deal with the various cooling loads with maximum external conditions equivalent to a dry bulb temperature of 90° F. with a relative humidity of 50 per cent., when the various data corresponding to this condition as read off the psychrometric nomograph, Fig. 59 would be as follows:

Dry bulb temperature	90° F.
Relative humidity	50 per cent.
Wet bulb temperature	75° F.
Dew-point temperature	69° F.
Moisture content	106 grains per lb.
Total heat content	37·7 B.T.U. per lb.

Knowing these details to be the basis upon which to work, we can decide that the internal conditions must be 78° F., for which temperature experience has shown that the most desirable relative humidity for comfort conditions would be a maximum of 40 per cent. The characteristics of the internal atmosphere can, therefore, be stated as follows:

Dry bulb temperature	78° F.
Relative humidity	40 per cent.
Wet bulb temperature	62° F.
Dew-point temperature	51·5° F.
Moisture content	57 grains per lb.
Total heat content	27·5 B.T.U. per lb.

As the total amount of air to be introduced into the building has already been determined as 2,000,000 cu. ft. per hour, and knowing that at whatever temperature this is introduced it will have to rise to the desired internal condition of 78° F. before leaving, and that in rising it will absorb the total of 552,000 B.T.U. per hour, which is the heat tending to a rise in temperature, we may now easily determine what rise in temperature is taking place after the air has been introduced into the restaurant, as follows:

$$\frac{552{,}000}{2{,}000{,}000 \times 0{\cdot}02} = 14^\circ \text{ F.}$$

It should be mentioned that the figure 0·02 represents the amount of heat required to raise 1 cu. ft. of air at normal temperature through 1° F.

As the air is to rise 14° F. due to absorbing the whole of the heat in the restaurant, it must, therefore, be introduced at a temperature equal to $78-14 = 64°$ F. This temperature is well above the dew-point temperature corresponding with the desired internal conditions, which means that the condition of the air entering the restaurant will be such that it is not fully saturated and is actually found to be, by reference to the psychrometric nomograph, 65 per cent. relative humidity, the wet bulb temperature corresponding to this condition being approximately 57° F. and the heat content 24·25 B.T.U. per lb.

Total Cooling Load.

It must not be thought that the 552,000 B.T.U. represents the total amount of cooling which is to be done by the plant. We have seen that the heat content of the entering air is equivalent to 37·7 B.T.U. per lb., and it must be remembered that to ensure the correct amount of moisture being present in the air in the restaurant it must be cooled by some means which may be determined later, to 100 per cent. saturation at the dew-point of 51·5° F., corresponding to the desired internal restaurant conditions, at which point the total heat is 21 B.T.U. per lb.

It may be seen, therefore, that for every pound of air passing through the plant $37{\cdot}7-21 = 16{\cdot}7$ B.T.U. must be removed by some convenient method.

Assuming that the air has a density equivalent to 13·7 cu. ft. per lb., we may calculate the total weight of air handled as follows:

$$\frac{2{,}000{,}000}{13{\cdot}7} = 146{,}000 \text{ lb. per hour.}$$

The total amount of heat which must, therefore, be removed in cooling is:

$$146{,}000 \times 16{\cdot}7 = 2{,}450{,}000 \text{ B.T.U. per hour approximately.}$$

This is the figure upon which the cooling plant must be designed.

As an aid to grasping the various stages of the problem, Fig. 43 is given showing diagrammatically the conditions existing between the point of entry of the air and the point at which it leaves the restaurant, having served for cooling purposes.

Is Air Recirculation Practicable?

Those who have had any experience of refrigeration equipment combined with air-conditioning plants will realize that to remove

2,450,000 B.T.U. per hour calls for a plant which, compared with the cost of the remainder of the air-conditioning equipment, is very expensive, and a primary consideration would be how to reduce, if possible, this cooling demand without in any way diminishing the cooling effect in the restaurant itself, and the first question that prompts itself is whether it would be possible to employ recirculation of the air, and if it is, how recirculation could be of any benefit.

We can best consider this matter by noticing that if the whole of the air has to be taken from an external source 16·7 B.T.U. must

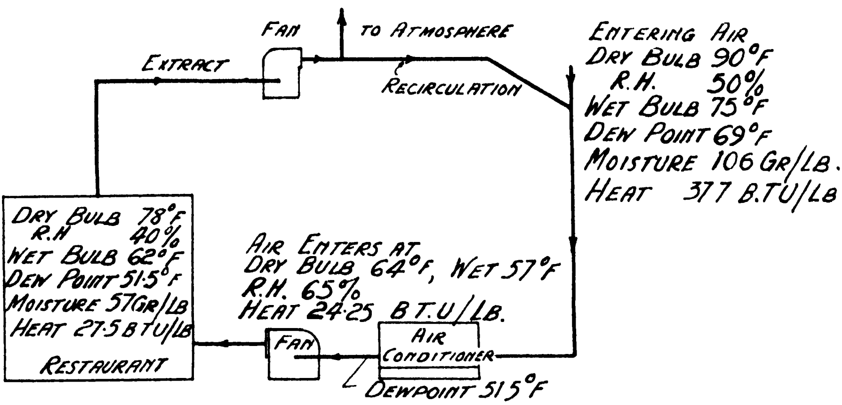


FIG. 43. Air-conditioning process diagrams.

be removed from every pound that is handled. Whilst, on the other hand, if recirculation is employed, that is to say, some air is taken from the restaurant by means of an extract system and mixed with a proportion of fresh air taken from an external source before again being passed into the restaurant, the amount of air which is recirculated will only have to be reduced in heat content from 27·5 B.T.U. per lb. to 21 B.T.U. per lb. = 6·5 B.T.U. per lb., so that for every pound of air that is recirculated there is a distinct saving of

$$16{\cdot}7 - 6{\cdot}5 = 10{\cdot}2 \text{ B.T.U. per lb.}$$

It has been established that for ventilation purposes, providing that an amount of air equivalent to 500 cu. ft. per person is taken from a fresh external source, any balance may be made up of recirculation, so that in our case where we are supplying 2,000 cu. ft. of air per person, 75 per cent. of this may be recirculated air.

To determine now the revised cooling load for employing 75 per cent. recirculation, we will first find what approximately the heat content of the mixture of recirculated and fresh air would be. This could be found as follows:

	B.T.U. per lb.
One part fresh air at 37·7 B.T.U. per lb.	37·7
Three parts recirculated air at 27·5 B.T.U. per lb. . .	82·5
Total	120·2

Dividing this by the total number of parts, namely 4, we find that the average heat content of the mixture of air would be approximately 30 B.T.U. per lb., so that the amount of heat to be removed per pound for cooling to saturation at the desired internal dew-point would amount to:

$$30-21 = 9 \text{ B.T.U. per lb.}$$

We will now revise the total cooling requirements in the ratio of the total amount of heat to be removed per lb. for the two conditions as follows:

$$\frac{2,450,000 \times 9}{16{\cdot}7} = 1,320,000 \text{ B.T.U. per hour.}$$

In other words, by employing 75 per cent. recirculation we have nearly halved the cooling requirements.

As the relation between the volume of fresh air and the extraction system for this class of building would be such that the extraction system would remove only 75 per cent. of the volume handled by the fresh-air system, it will be obvious that the system will be arranged for the whole of the air actually extracted to be recirculated, remembering that these remarks apply only for extreme conditions.

In a similar manner, in connexion with the ventilation of the kitchen, the relation between the fresh-air input and the amount of extracted air would be such that the fresh air was only 75 per cent. of that extracted. By maintaining these ratios of air volumes we ensure that there is a definite flow of air from the restaurant into the kitchen.

Air Volume Required in the Kitchen.

Opinions differ considerably upon the amount of air which must be supplied to or exhausted from a kitchen to provide adequate ventilation. For this particular building it must be remembered that over a long period the kitchen will be working at high pressure in providing varied meals, and there must therefore be a large amount of heat present.

Many authorities have stated that air changes should be anything up to 20, but the author's experience has shown that this number of air changes per hour is far from sufficient and that for efficient ventilation at least 40 air changes per hour are required.

As the total cube of the kitchen amounted to 90,000 cu. ft., it may be seen the amount of air to be extracted would amount to

$$90{,}000 \times 40 = 3{,}600{,}000 \text{ cu. ft. per hour.}$$

As the equipment will be designed for 75 per cent. of this amount to be introduced as fresh air, the amount of fresh air would amount to

$$3{,}600{,}000 \times 0{\cdot}75 = 2{,}700{,}000 \text{ cu. ft. per hour.}$$

As the restaurant system is introducing 2,000,000 cu. ft. per hour of fresh air and removing only three-quarters of this amount or 1,500,000 cu. ft. per hour in the extract system, there is an excess of air in the restaurant of 500,000 cu. ft. per hour which would go towards the difference of 900,000 cu. ft. per hour required in the kitchen. The margin would have to be made up in the kitchen from other sources if they existed. It would be desirable to leave the plant designed on this basis, and if any trouble was found to be experienced in practice, the capacity of the kitchen extraction system could be reduced slightly by the equivalent of 400,000 cu. ft. per hour, which would then still leave an extract air change of 36 times per hour.

Methods by which Cooling may be Accomplished.

There are many systems which could be introduced for cooling the air before it entered the restaurant.

We will consider, firstly, that we are employing an air-washing system of the ordinary type, in which the air is passed through a water-spray chamber. In this case, we would have the possibility of accomplishing a certain amount of cooling even when the spray water is recirculated, whilst if further cooling is desired, the spray water could be obtained from a cool well, and after use in the air washer employed for domestic purposes, or alternatively the spray water could be refrigerated by suitable means. These might be described as indirect methods of cooling.

We have other methods available for direct cooling: one system which has been considered in the past is the use of ice in solid form, the air being brought into contact with this before being passed into the building.

Yet another method is found in passing the air through a tubular or similar battery, the cooling medium for this battery being either well or refrigerated water. Each of these methods must be considered in turn to see which of them would be practicable and at the same time economical in running costs.

Another possible alternative is the use of ice for cooling the spray water in an indirect cooling system. It should be mentioned here that the continuous use of ice is rarely practicable.

It will be appreciated that either ice cooling or cooling by means of a refrigerating battery cannot give anything like the flexibility that could be obtained from a dehumidifying or air-conditioning plant, because with either of these systems provision is made for cooling only with the object of condensing out surplus moisture. It is possible, however, during certain periods of the year that moisture must be added to give the required standard of humidity, so that if either of the two cooling systems we have at present described are employed, an additional humidifying unit is also essential for use generally during the winter months.

Cooling by Contact with Spray Water.

In the general form of dehumidifying apparatus employed in conjunction with public building ventilation equipment the air entering the building is passed through a spray chamber filled with a finely divided mist of atomized water which is introduced to the spray chamber through a series of nozzles fed by water under pressure. The air before leaving the spray chamber passes through a series of corrugated baffles for the purpose of removing any entrained particles of moisture.

It will be understood that the provision of cool water either from an artesian well or some refrigerated source can easily be arranged for.

We have previously seen that the fundamental requirement of the air-cooling system is that air at a relatively high temperature and humidity should be reduced in temperature and have a large amount of moisture removed to convert it to the actual conditions required in the room.

It may seem strange that this should be possible when the incoming air is itself brought into contact with atomized water, but when we remember that a heat balance must always exist the matter is easily explained. The entering air being at a higher temperature than the spray water will first tend to absorb more moisture, for which purpose heat must be obtained from some source, the only source from which heat may be taken being the air itself, resulting in a temperature fall in the air and to complete the balance a temperature rise of the spray water. The incoming air will actually cool to a temperature nearly approaching its incoming wet bulb temperature, at which temperature it will be almost fully saturated with moisture. If the spray water employed in the air washer is recirculated by means of a pump, then the heat balance is such that spray water and air become nearly similar.

We are endeavouring in our cooling process to reduce the total heat content of the air in passing through the plant, and from a knowledge of hygrometrical relations we know that provided the wet bulb temperature of any particular sample of air does not alter, its total heat remains the same irrespective of whether the moisture content and dry bulb temperature vary. What actually happens with an air washer employing recirculated spray water is that the dry bulb temperature naturally becomes lower owing to the heat being abstracted from the air for evaporating moisture, a process which cannot be stopped when partially saturated air is in contact with moisture. We wish to point out, therefore, that the cooling accomplished with recirculated spray water, or cooling by evaporation as it is normally known, does not give any control over humidity conditions, and moreover, the cooling effect is apparent only, and in use for guaranteed temperature and humidity purposes is of no value. The only occasions where cooling by evaporation can have any value are those where the wet bulb temperature of the incoming air is either equal to or less than the dew-point equivalent of the desired internal condition, in our case 51·5° F.

The Use of Well Water or Refrigerated Water.

Where the water in the air washer is supplied at a definite temperature below that of the entering air there is the tendency that we have mentioned for the air when first brought in contact with the water to absorb moisture until it is cooled to within one or two degrees of its entering wet bulb temperature, at which temperature it is nearly saturated; but owing to the fact that a heat balance does not then exist, heat is transferred from the air which is still warm to the cool water, resulting in moisture being condensed out of the air, which finally cools until calculated conditions are reached. These two stages definitely occur in the length of the spray chamber.

Considering now the cooling which may be accomplished by the use of water taken direct from an artesian well, in the first instance it must be noted that during the summer the water temperature will be approximately 55° F. as a minimum, often rising to 60° F., which immediately shows us that for our particular problem it is not possible to do the whole of the cooling by means of well water, because even at 55° F. the water is still considerably higher in temperature than the desired dew-point of the air leaving the humidifier, namely 51·5° F. Assuming that the maximum amount of water that can be usefully employed in the air conditioner is approximately 10 gallons per thousand cu. ft. of air, then the total quantity which can be passed through the air conditioner will amount, for 2,000,000 cu. ft. of air per hour, to 20,000 gallons per hour. When determining what actual cooling may be done we must now construct a heat exchange diagram as Fig. 44, the curve representing the heat removed from the air at various wet bulb temperatures between the entering wet bulb temperature of 75° F., where no heat would be absorbed, and 62° F., wet bulb temperature equivalent of the desired room condition, at which temperature the total required cooling of 2,450,000 B.T.U. per hour would be accomplished, this assuming no air

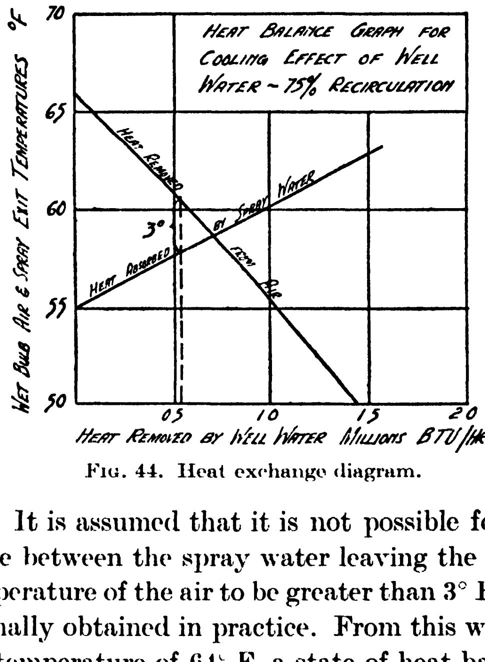


FIG. 44. Heat exchange diagram.

recirculation. It is assumed that it is not possible for the temperature difference between the spray water leaving the washer and the wet bulb temperature of the air to be greater than 3° F., the condition which is normally obtained in practice. From this we see that at an air wet bulb temperature of 64° F. a state of heat balance exists for this 3° difference, and for this to be possible the water rises from a temperature of 55° F. to 61·5° F., the total amount of heat removed in this cooling process being 1,300,000 B.T.U. per hour. The point of intersection of the air and water curves represents the point where the air and water leave at the same temperature, a condition which only obtains in theory.

Throughout a large portion of the year it would be found that well-water cooling would give sufficient refrigerating effect to ensure reasonable comfort conditions, but at the times when this is not so the only possible alternative is refrigeration, which might be applied by means of ice or direct refrigerating batteries, as previously explained. The proportion of the total refrigerating requirements not removed by well water amounts to 1,200,000 B.T.U. per hour.

Use of Well Water when Air Recirculation is Employed.

We have already seen that where air recirculation is employed the total amount of heat to be removed is reduced to 1,320,000 B.T.U. per hour, the total content of the air mixture entering the conditioner being 30 B.T.U. per lb. We must now find what wet bulb temperature this represents, which can be read directly from a psychrometric chart as 66° F. In this case it can be found by calculation that, if it were possible for the water to absorb the whole of the desired amount of heat, it would need to rise in temperature approxi-

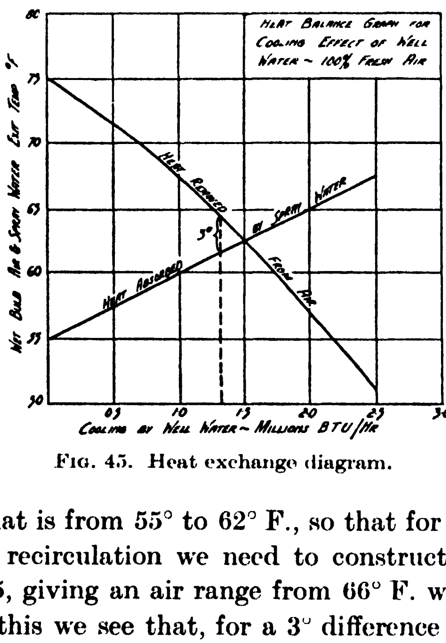


FIG. 45. Heat exchange diagram.

mately 7° F., that is from 55° to 62° F., so that for these conditions existing for air recirculation we need to construct a heat balance curve as Fig. 45, giving an air range from 66° F. wet bulb down to 51·5° F. From this we see that, for a 3° difference between air and water leaving temperatures, it is possible to remove 520,000 B.T.U. per hour, the water rising from 55° to 57·5° F., and the air cooling from 66° F. wet bulb down to 60·7° F. wet bulb. In this case, therefore, the balance of the heat required to be removed not taken out by well water is 800,000 B.T.U. per hour, which needs to be dealt with by refrigeration.

Refrigeration in Conjunction with Use of Well Water.

We will consider in this case the use of refrigeration in conjunction with well water for the condition requiring the removal of 1,320,000 B.T.U. per hour, that is, with recirculated air. It will be assumed that the 20,000 gallons of water per hour is being taken from the artesian well, delivered through one bank of spray nozzles in the air conditioner, and then made use of for some purpose such as kitchen supplies, for as we have seen from the heat balance curve it has only risen just under 3° F. in temperature by its use in the air conditioner. In order to deal with the balance of 800,000 B.T.U. per hour, an additional bank of sprays could be provided in the spray chamber, the water employed for this bank of sprays being recirculated through the cooling section of the refrigerating equipment. In this bank of sprays it is suggested that half of the quantity of water employed in the other should be used, namely 10,000 gallons per hour, in which case to remove the necessary heat it would need to rise through the following temperature:

$$\frac{800,000}{10,000 \times 10} = 8^\circ \text{ F.}$$

Let us assume that the first stage of cooling has been well-water cooling and that, having passed through this section of the air conditioner, the air then meets the refrigerated water section. As the air has ultimately to be cooled to 51·5° F., dew-point temperature, and it would be 100 per cent. saturated, having passed through a double-bank air washer, the dry bulb temperature leaving the air washer would also be 51·5° F., so that, assuming again a 3° difference between air and water leaving temperatures, the spray water in the refrigerated section would leave at 51·5—3 = 48·5 and would, therefore, need to enter the sprays at 40·5° F. These temperatures would be such as to ensure economical working of the refrigerating equipment.

There might appear to be another alternative method of combining the use of well water and refrigerated water by taking the water direct from the artesian well to the refrigerating plant and thus introducing it to the air washer in a single bank of sprays. In this case the rise of temperature which would take place would amount to:

$$\frac{1,320,000 \text{ B.T.U.}}{20,000 \times 10} = 6\cdot6^\circ \text{ F.},$$

so that allowing for the necessary air and water temperature difference of 3° F. it would be necessary to refrigerate the water to 51·5—6·6—3 = 41° F. approximately, but we must remember that the building which we are considering is a large restaurant, and therefore it would not be desirable to distribute water at so low a temperature as approximately 48° F. for kitchen use so that this alternative combined method must be abandoned. Moreover, it will be observed that although the quantity of water handled only requires to be cooled through 6·6°, if it were taken in from the well at 55° and cooled to the necessary temperature of 41° F., this would represent a total cooling of 14° F.

Use of Recirculated Water and Refrigeration.

It has been shown that there is nothing to be gained by using an air washer with one bank of sprays employing recirculated water in order to obtain cooling by refrigeration because, during the whole of this process the total heat of the air does not alter and no real refrigeration is accomplished. We could employ, however, a system in which the whole of the spray water were recirculated through a refrigerating plant. For this purpose a single bank of sprays could be provided with water on the basis of 10 gallons per 1,000 cu. ft. of air.

With this system it would be necessary to absorb the whole of the 1,320,000 B.T.U. per hour, necessitating a rise in temperature of the water at:

$$\frac{1,320,000}{10,000 \times 10} = 13° \text{ F. approximately,}$$

so that allowing the usual 3° difference the refrigerating plant would need to deliver water to the sprays at 36° F., and this temperature would be the practical lower limit of spray water temperature.

Heating During Winter.

Having based the general design of the air-conditioning equipment upon the summer requirements which invariably call for the greatest output from the plant as far as air volumes are concerned, we must now turn our attention to the requirements of the system during winter months. Firstly, however, it must be very obvious that heating will only be required in a restaurant of the type which we are considering at very rare intervals, because as we have already seen the heat equivalent of the people in the room is far greater than the heat loss during the winter months. Although at first sight it might be thought desirable to reduce the total volume of air supplied during heating periods, further consideration would show that there is very little to be gained, and it is suggested that in order to reduce the total heat-requirements of the plant at such times, recirculation up to 75 per cent. should be employed.

The capacity of the plant has been based on 2,000 cu. ft. of air per person, so that 75 per cent. recirculation would still mean that 500 cu. ft. per person is completely fresh air from an external source, the remaining 1,500 cu. ft. being recirculated, a rate of supply which is ample to meet requirements.

Considering the times at which heating will be required, we may see that the only call for heating will be for raising the temperature of the restaurant to a minimum temperature of 60° F. in readiness for use by the public.

Temperature at Re-Heater.

The necessary heat for this purpose is provided by the Re-Heater or final heater, in which the air must be raised to such a temperature that in cooling to the desired room condition of 60° F. sufficient heat is given off to balance exactly the heat loss from the structure. This heat loss would, of course, be calculated by the usual accurate methods, but for the purpose of this example we may assume it to be 200,000 B.T.U. per hour.

The total quantity of air handled by the plant is 2,000,000 cu. ft. per hour, so that the increase of temperature required above room temperature will be:

$$\frac{200{,}000}{2{,}000{,}000 \times 0{\cdot}02} = 5^\circ \text{ F.},$$

so that air would need only to be reduced to 65° F.

We may decide that during the winter months when cooling is not called for, the internal conditions will be 60° F., and 50 per cent., relative humidity, corresponding to a wet bulb temperature of 50° F., and a dew-point temperature of 41° F., whilst the basic external conditions are 32° F. and 50 per cent. R.H., corresponding to a wet bulb temperature of 28° F. and a dew-point temperature of 18° F.

During the winter months, therefore, air will be leaving the humidifier at 41° F., so that the re-heater needs to raise air from 41 to 65 F.

Requirements of Pre-Heater.

The Pre-Heater in this case will be assumed to have to provide sufficient heat not only to raise the temperature of the air but also to evaporate the necessary moisture to give the correct internal humidity. At a dew-point temperature of 41° F., the total heat of air is 16 B.T.U. per lb., whilst at the entering condition of 18° F. the total heat is only 6 B.T.U. per lb. At the internal condition of 50° F. wet bulb temperature, the total heat is 20 B.T.U. per lb. As we have decided to recirculate 75 per cent. the average heat content of air entering the air-conditioning plant will be as follows:

$$\frac{(75 \times 20) + (25 \times 6)}{100} = 16{\cdot}5 \text{ B.T.U. per lb.},$$

which, as may be seen, is already higher than the desired total heat content leaving the humidifier, namely 16 B.T.U. per lb., which would seem to indicate that where recirculation up to 75 per cent. is provided there is no need to employ a pre-heater.

On the other hand, it must be remembered that if the building has been allowed to become very cold, its condition will nearly approach at some time the basic external condition, so that as a precaution the pre-heater should be provided, although under most conditions it need not be used. The conditions referred to immediately suggest a simple means of controlling the humidity by having dampers in the fresh-air intake and recirculating ducts operated in a differential manner by means of a wet bulb thermostat situated in the restaurant, the proportions of fresh and recirculated air being varied to give the desired relative humidity.

It is extremely unlikely that the internal temperature would fall below 40° F. with a humidity of 40 per cent., the corresponding wet bulb temperature being 32° F., at which condition the total heat is 12 B.T.U. per lb., so that it would be quite safe to design the pre-heater on the assumption that the minimum average heat content of the entering air would be:

$$\frac{(75\times 12)+(25\times 6)}{100} = 10{\cdot}5 \text{ B.T.U. per lb.,}$$

so that the pre-heater would need to add $16-10{\cdot}5 = 5{\cdot}5$ B.T.U. per lb.

Assuming 14·5 cu. ft. of air per lb., the total heat to be added by the pre-heater will be:

$$\frac{2{,}000{,}000\times 5{\cdot}5}{14{\cdot}5} = 760{,}000 \text{ B.T.U. per hour.}$$

The number of degrees above the humidifier dew-point temperature to which the air must be raised in the pre-heater will, therefore, be:

$$\frac{760{,}000}{2{,}000{,}000\times 0{\cdot}02} = 19^\circ \text{ F.,}$$

so that the pre-heater must raise the air to $41+19 = 60^\circ$ F. The inlet temperature to the pre-heater under the conditions of recirculation assumed will be somewhere between 40° F. and 32° F., and will be approximately 38° F.

Upward or Downward Ventilation.

We are now faced with the problem of deciding definitely upon the method of distribution of the air in the restaurant, and in the first instance we see that there are three distinct systems which might be employed:

(1) Downward ventilation.
(2) Upward ventilation.
(3) Cross ventilation.

Considering firstly cross ventilation, what is meant by this is a system in which the air is introduced at one side of the room and extracted at the other, and a very brief consideration will show that this method will not be practicable because, when cooling is being accomplished, it means that large volumes of cool air are introduced at one side of the room, passing gradually across the room until it leaves at about the desired room temperature, whilst in the same way when heating is being accomplished warm air is introduced at one side, gradually cooling as it passes across to the extraction point at the other.

Considering now a choice between upward and downward ventilation, in the first instance it must be remembered that during the greater period of occupation cooling is necessary, and without doubt for such conditions it is better to introduce the cool air at high level and allow it to follow the natural tendency of falling on entering the warmer atmosphere.

Moreover, it must be remembered that during such periods as heating is required, the entering air heat is so very little above the desired internal temperature that there would be no trouble due to stratification as might be the case if lower air volumes had been used and the air introduced at a higher temperature. Again, as far as inlets or outlets at low level are concerned, the only feasible position is round the walls, and if air were introduced in such a position it is extremely unlikely that it would disperse evenly over the whole of the restaurant before leaving any high level extraction points unless it were injected at high velocity which would cause draughts, and extracted at very low velocity.

Another point to be considered is that in a restaurant of the type we are considering trouble might be experienced from smoking, and extraction at high level could not but add to any possible difficulties from this source, so that considering all things it would be decided to employ the downward ventilation system.

To obtain a system of this type, theoretically correct, is not possible in this instance, and the only sound practical method of introducing and extracting air is to arrange for the whole of the fresh air to be introduced through inlets arranged flat in the ceiling, suitably concealed to harmonize with the architectural features of the room.

Thus, for instance, it is possible to combine the fresh-air inlets with air diffusers and concealed or indirect lighting fittings.

As far as the extract system is concerned, it is not desirable to have outlets actually over the floor area, principally because of the dirt and dust that would accumulate, so that in practice it would become essential to place these outlets around the outer walls of the restaurant. These might well be arranged as almost continuous grilles, leading into suction boxes, from each of which several extract ducts are taken.

The Automatic Control of the Air-conditioning Plant.

There are many systems by means of which the air-conditioning plant may be placed under close automatic control as far as temperature and humidity are concerned, but one recent development of the Drayton systems is of particular interest as it is of use for all public building systems.

Whatever method of control has been used in the past has suffered from the one difficulty that, when the air is saturated at the temperature equivalent to the required relative humidity corresponding to the temperature in the rooms that the plant has been designed to maintain, and the plant is first started up to deliver air into a relatively cold building, condensation invariably takes place. In the case of the system being described, this difficulty is overcome by putting under the control of a thermostat in the building the valve controlling the warming of the spray water, or the pre-heater, as the case may be, in such a way that the air is not allowed to be brought to the desired dew-point condition until the required room temperature is first reached.

This is accomplished by providing the control valve with a second diaphragm top which may be done at very little extra expense. A further advantage of this system is that it is designed to overcome the trouble that might be experienced due to the occupants being discomforted by the relatively cold air entering the room, when it has become overheated.

Figs. 46, 47, and 48 show this system of control as applied to air-conditioning plants, employing the various well-known systems of evaporating moisture into the air.

Fig. 46 shows air-conditioning apparatus with a pre-heater. The thermostat F controls the dew-point by operating the valve D, which is fitted with a double top. The thermostat and valve are of the direct-acting type. Under normal conditions, the valve is operated on the lower pressure chamber by the thermostat F. The room regulator B, which is of the reverse-acting type, holds the valve closed by maintaining pressure in the upper pressure chamber, until normal room temperature is reached, thus eliminating any possibility of deposition of moisture in the room due to condensation. The room temperature is maintained constant by room regulator E, which is of the reverse-acting type and operates on the upper pressure chamber of valve A, which is a double-topped valve controlling the admission of steam to the final heater. The lower pressure chamber of valve A is under the control of thermostat C, of the direct type, and opens the valve in case the air temperature falls to a given minimum, maintaining the air temperature at this minimum,

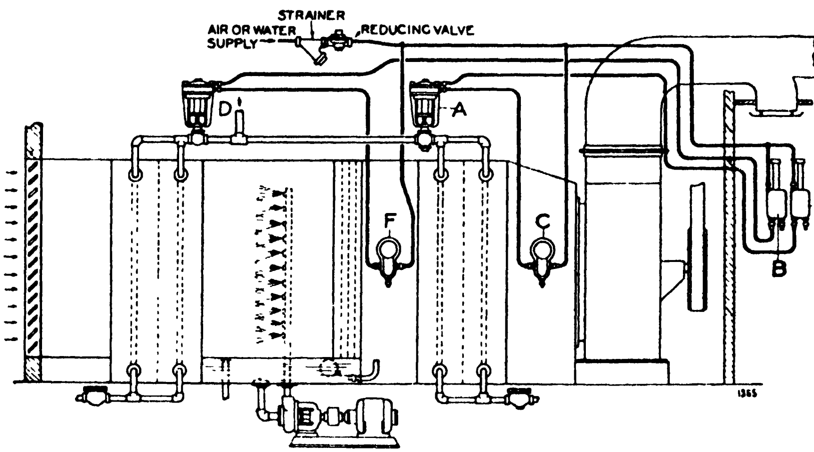


FIG. 46. Drayton control system.

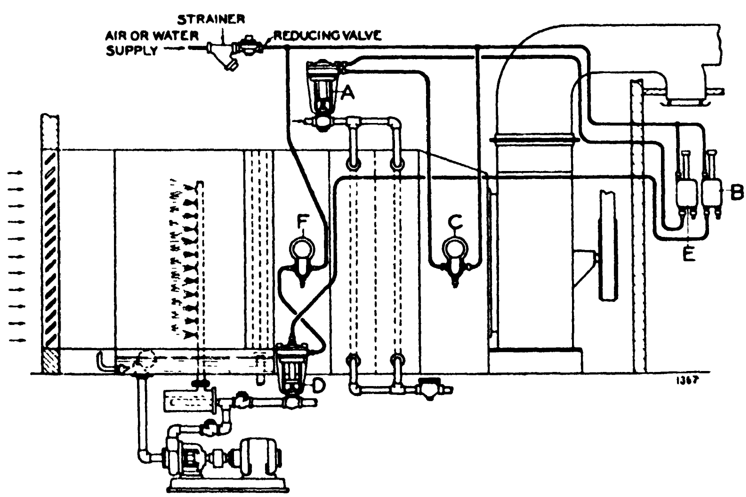


FIG. 47. Drayton control system.

all the time the room regulator E remains inoperative by reason of the room being at the correct temperature. This prevents any cold draughts of air being felt.

Fig. 47 shows the system applied to an air-conditioning plant in which moisture evaporation is accomplished by circulating the spray water through a Drayton water blender, an apparatus by means of which steam is mixed with the water.

Similarly, Fig. 48 shows the system controlling a plant, where evaporation is secured by injecting steam into the air-washer tank.

Damper control for recirculation of the air can be combined with the conditioning control which has been described.

We should mention that the regulators shown in the diagrams, which would normally be used, are the Drayton A.M. regulators for the pre-heater control and the low limit control of the final heater, and the E.M. regulators for the normal room control and the pre-heater control from the room. These regulators are of a simple relay

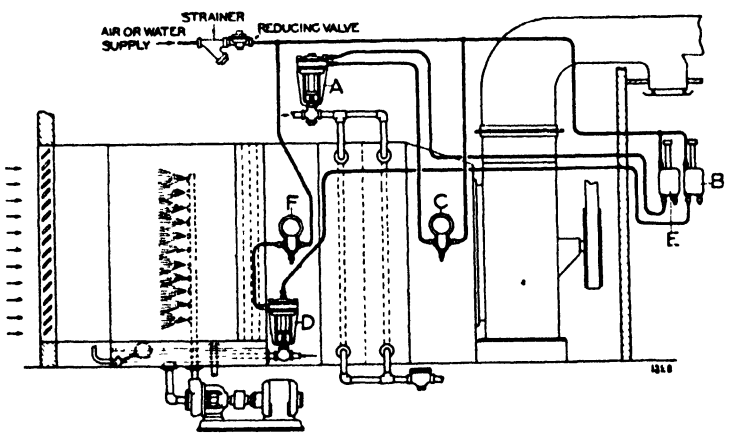


FIG. 48. Drayton control system.

type, expansion stem operated, the expansion stem being copper in the case of the E.M. regulator. Both regulators can be obtained direct or reverse acting, that is, they can be arranged to close the pilot valve with increasing temperature or to open it. Either type can be used in conjunction with direct-acting valves, that is, valves that open on failure of pressure in the pressure chamber or those that close under these conditions.

Advantage is taken of the fact that these pressure-operated valves can be fitted with two pressure chambers, one above the other. One pressure chamber is used for the master or cut-out control, and the other for the normal control. For instance, in the case of a direct-acting valve with two pressure chambers, the valve will remain closed all the time; either of the pressure chambers is under pressure regardless of what may be happening in the other chamber.

Layout of Kitchen Ducts.

In considering the arrangement of ductwork for supply and extract in connexion with the kitchen ventilation, it must be remembered that the system is installed for the dual purpose of removing steam or vapour and also for keeping the kitchen reasonably cool. With a large kitchen, for the restaurant which we are considering, it would be desirable to arrange for extract hoods over large ranges, fish fryers, boiling pans, and similar apparatus, whilst in addition openings will be provided at high level in the connecting ductwork to take care of general extraction from the kitchen. As far as the fresh-air system is concerned, it is obviously better to introduce the greatest volumes of air in those places where excessive heat is likely to be felt. At those points it is suggested that inlet ducts should be brought down to breathing level and provided with outlets which are adjustable as to volume and direction of air flow. In addition, outlets in the ductwork at high level are necessary for general fresh-air ventilation of the whole area in the kitchen.

It must be remembered that some form of air filtration must be provided on the fresh-air system, and it is suggested that either a dry plate or viscous type of filter is most suitable for this purpose. Unless refrigeration is available, the use of an air washer is not advisable, as it is likely to result in too humid conditions of the air.

In some localities it is inadvisable to discharge air from the extract system in the condition it would be in, taken straight from the kitchen, for not only is the air tainted with kitchen smells, but it is also heavily laden with grease. It is possible with some installations to pass the air discharged from the extract fans through an air-washing device to remove grease, whilst similarly ozone treatment has been found to be beneficial in neutralizing smells. The use of these refinements is obviously costly, and would only be resorted to where discharging the air in an untreated condition would be likely to interfere with neighbouring property.

Ventilation of Smaller Kitchens.

With smaller kitchens, such as those attached to staff canteens, hotels, cinemas, and theatres, the problem of ventilation is a little more simple, although the fundamental principle remains the same, and there is still the definite need for powerful extraction and consideration of where the fresh air is to come from to replace this extracted air. Also, with a small kitchen, a decision is required on whether extraction shall be local by means of hoods over apparatus producing heat or fumes, or general extraction from the room.

The system illustrated in Fig. 49 is typical of a ventilation system for a kitchen where the ventilation is of general type, with a local hood over the central range. With this particular layout, the fresh-air and extract fans are situated on an adjoining roof, and the extraction system also serves an adjacent hall.

The system of ventilation applied to the kitchen of the Hotel Moderne in Paris is of particular interest, as it makes provision for

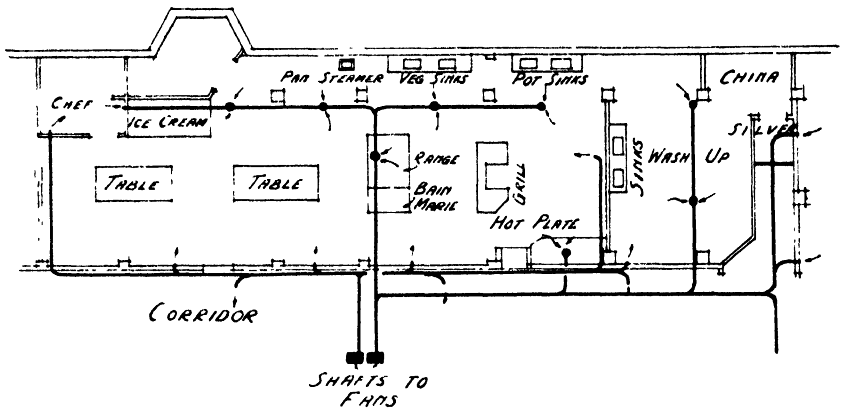


FIG. 49. Kitchen ventilation.

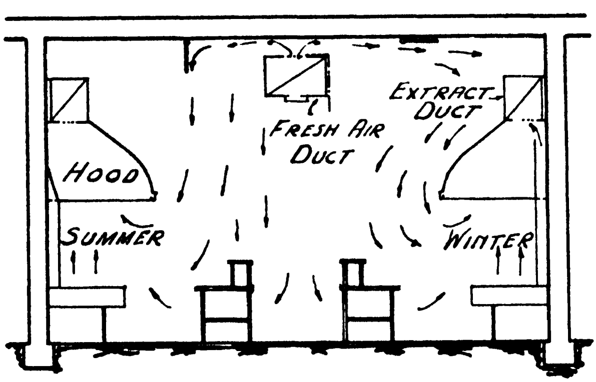


FIG. 50. Kitchen ventilation system.

adjustment of the ventilation for winter and summer conditions. This is illustrated in Fig. 50, which is a cross-section through the kitchen. It will be seen that extraction takes place through hoods arranged over the cooking apparatus against the walls, whilst a fresh-air inlet duct runs at ceiling level along the centre of the kitchen. This fresh-air duct has outlets on the bottom and top, and suspended deflectors at either side of the duct enable the air to be deflected downwards during the summer when cooling effect and draughts can be tolerated as giving a measure of comfort. During the winter, the entering air would be at a temperature too low to allow it to be so discharged, and the deflectors are then fixed flat upon the ceiling, so that the air is projected against the walls, is heated by the extract hoods, and finally reaches the lower working level in a tempered condition.

A hospital or hotel kitchen of modern construction, with vitreous enamelled plant and tiled walls, calls for consideration in applying the ventilation equipment with a view to concealing it as far as possible from view. In such cases the ducts may be constructed as part of the building structure, or in metal covered with expanded metal to which tiles may afterwards be fixed.

It is important in first-class kitchens where ducts are exposed to insulate the cool fresh-air ducts to prevent condensation of moisture from the humid air.

Design Data for Kitchens.

Whilst silent-running ventilation equipment is of secondary importance in kitchens, apart from the effect in surrounding rooms, there is still a limit to the air velocities which can be employed. The following data are reliable as a basis of design.

	Velocity ft. per minute
Maximum in extract ducts	2,500
Maximum in fresh-air ducts	1,500
Discharge from fresh-air ducts for arrangement in Fig. 49	500
Discharge from fresh-air ducts for arrangement in Fig. 50	300
Velocity in fresh-air ducts for deflector system in Fig. 50	1,000

There are instances where the system in Fig. 50 may be applied using a fresh-air inlet without a fan, the extract fan in these cases having to overcome the resistance of the fresh-air duct.

Small kitchens adjoining cinema tea-rooms may be adequately ventilated by means of a propellor type of fan situated in a window.

It is the practice in some of the better-class hotel kitchens to subdivide kitchens into departments and to ventilate the kitchen and all adjoining rooms, the following being desirable rates of air change:

Nature of room	*Air changes per hour*
Ice cream, general preparation	5
Wine dispense, chef office, pantry, staff mess-rooms, vegetable cleaning	10
Fruit, glass, and silver	15
Patisserie, fish cleaning, poultry plucking	20
Dish washing, pot scullery, still room	30
Ranges and grills	40

Ventilating the Canteen.

Figure 51 shows a typical works canteen† with kitchen separated by a dividing wall or partition, and access door or doors. The proportion varies according to accommodation. For a small seating the size or area of the kitchen will be greater in proportion than in a canteen holding a much larger number.

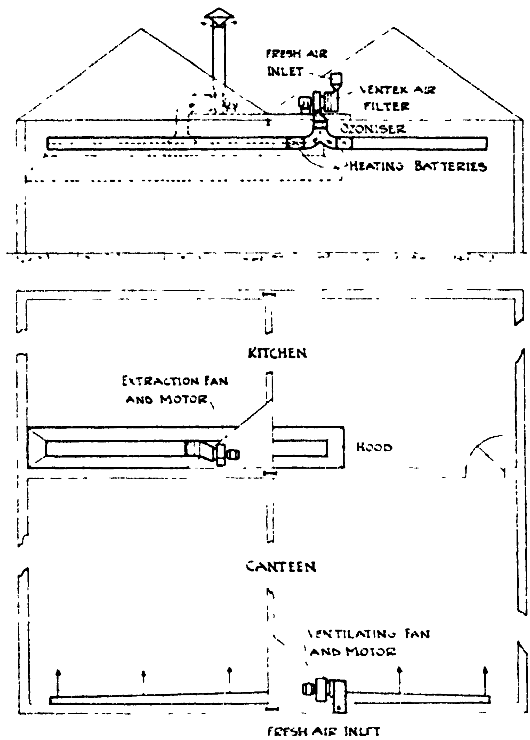


FIG. 51. 'Ozonair' canteen ventilation system.

The kitchen ventilation is by extraction alone. By this means kitchen smells are confined to the kitchen itself, and even there they are not confined for long because the extraction system is so rapid that any smells disappear as soon as they are created, for the air of the whole of the kitchen is renewed every two or three minutes. In other words, twenty or thirty complete changes of the kitchen atmosphere take place every hour.

The fresh air is drawn from as clean a source as possible at a high level. This is high enough to escape the dust from the road or path, and not so high as to catch the chimney smuts. The inlet fan driven by a motor, directly coupled or by belt, is carried on a light steel platform fixed to the end wall of the canteen.

† See *Industrial Welfare and Personnel Management*, June 1934.

The air surrounding any works is usually more or less contaminated with solid matter in the form of dust, and it is desirable to pass it through a filter before distributing it through the canteen. An air filter, preferably of the viscous type, can be quite easily fitted to the air inlet, as shown in the top part of the drawing. A galvanized metal duct is fixed right and left of the fan. This duct is fitted with several openings through which the air is distributed evenly over the whole width of the canteen.

The whole of this air, after accomplishing its work of ventilating the canteen, finds its way into the kitchen through the door opening or through several openings made in the kitchen partition wall. These openings are fitted with non-return flaps which prevent any kitchen odours coming back to the canteen.

The kitchen extraction plant will cause a steady and equable flow of fresh air throughout the whole of the canteen merely by means of the simple distributing system described.

The inclusion of an ozonizing apparatus in the position shown will purify and invigorate the whole atmosphere.

Heating coils for warming the inlet air are fitted in the distributing ducts. There is usually to be found in most works a handy supply of steam which can be tapped, and if necessary reduced in pressure. The general warming of the canteen is taken care of by the usual system of radiators.

The ventilation of the kitchen is effected by extracting from a hood fixed over cookers and similar places, and connected with an extraction fan discharging into a shaft carried well above the roof, as shown in the illustration.

An alternative to a ventilating scheme for the canteen only, apart from the kitchen, is to make use of one or more small ozone apparatuses. These will purify the air of cooking and other smells, tobacco smoke, and so on, and add an invigorating quality to the atmosphere.

Ventilation of Large Office Buildings.

Large office buildings call for varied treatment in applying ventilation systems, and it is only in new buildings that it is possible for constructional reasons to apply extensive systems. Occasionally opportunities arise for the remodelling of plants installed twenty-five years or more ago, and the problem is then even more intricate than for a new building. We will consider some of the matters arising in the design of systems of this nature.

The preliminary step consists in acquiring by survey and examination of old records accurate details of the whole of the existing equipment. The following details would be obtained:

(1) Dimensions, make, and arrangement of fans.
(2) Dimensions and name-plate details of driving motors.
(3) Dimensions, positions, and construction of all ducts and grilles.
(4) Details of air heaters, filters if any, ozone apparatus, etc.
(5) Details of observed condition of all apparatus.

It is probable that much of the metal ducting and smaller apparatus would be found to be in bad condition, whilst only fans, motors, and similar equipment would prove suitable for use.

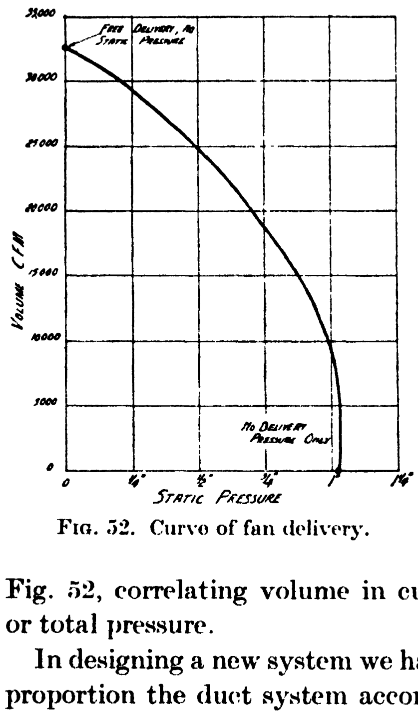


Fig. 52. Curve of fan delivery.

Having a fan of the centrifugal type, coupled direct to an electric motor running at constant speed, provides a problem, owing to the fact that according to the ducting system used and its resistance or pressure loss, so the volume of air which the fan can deliver will vary. Knowing the type and size of fan enables details to be obtained from the manufacturers, which could usefully be applied in the form of a graph as in Fig. 52, correlating volume in cu. ft. per minute with either static or total pressure.

In designing a new system we have, therefore, to accept the fan and proportion the duct system accordingly. As it is likely that the air changes required would be higher than when the plant was originally installed, it points to the need for having a duct system designed for very low pressure loss so that the fan can be run on that point on the curve giving the necessary high volume.

The power consumption must also be carefully considered, remembering that this will vary, even with a fixed fan speed, according to the volume and pressure. Usually, however, the characteristic of the fan would be such that a wide variation in duty at a fixed speed would cause little variation in power consumption. On the other hand, certain early types of centrifugal fan have rising power characteristics which may provide difficulties by pointing to motor overload.

Having decided upon the volume and pressure which can be used, further decisions must be made on the proportions of the total pressure which can be used in ducts, heaters, filters, and other apparatus, these points being decided by experience alone. No rules can be defined to cover such matters, which differ with every system.

In new office buildings, of modern construction, ventilation systems are extensively applied. Invariably extract ventilation is the system applied, giving 2–4 air changes per hour, except in board rooms and similar areas where crowded occupation is likely and up to 8 air changes is required.

Ventilation of Hotels.

Modern hotel buildings in this country are usually planned so that the minimum of mechanical ventilation is required. It is unusual to have mechanical ventilation in all rooms. Bath-rooms, W.C.s, and similar places are served by extract systems. For the ball-rooms, dining-rooms, and similar areas a complete air-conditioning system, comprising fresh-air and extract ventilation, providing a minimum of 1,000 cu. ft. of air per hour per person, should be provided, and this figure could be usefully increased to 2,000 cu. ft. for the ball-room, in order to be able to cool the room effectively.

The extraction system for the bath-rooms and W.C.s should be capable of providing 6–10 air changes per hour.

Hospital Ventilation.

The application of ventilation and air conditioning in hospital buildings is a matter receiving increasing attention in view of the general rebuilding and extension schemes still proceeding.

For wards, open-window ventilation still prevails, but before many years mechanical ventilation will be general in new buildings. At the moment, operating theatre and X-ray suites are the only departments being provided with such equipment. The following air changes should be provided:

Room	*Air changes per hour*	
	Fresh air	*Extract*
Operating theatre . . .	10	8
Anaesthetic	8	12
Sterilizing room . . .	10	20
Recovery	8	8
Doctor's wash . . .	nil	8
X-ray	6	8

There are, in large hospitals, subsidiary rooms attached to operating theatre suites which call for extract ventilation at the rate of 4–6 air changes per hour.

Opinions differ on the disposition of fresh-air and extraction openings. In some theatre suites, fresh air is introduced at high level and extraction takes place at low level, whilst with others the distribution is reversed. There is no definite support in favour of either method, for building arrangement is the guide to the best practicable scheme.

Indeed, some public hospitals have centralized extraction systems, served by one fan, for a group of operating theatres, the switch-gear operating the driving motor being so arranged that the fan is still running if only one theatre is in use, control being effected from a push-button station in the theatre. Small theatres for minor operations are usually served by a propeller extract fan, situated in an external wall.

Some theatres are provided with a centralized air-cooling plant for distributing fresh air to the rooms, air washers and heaters being used, all motors being controlled from a theatre push-button station having indicator lamps to show when the various parts of the plant are in use.

Other operating theatres, where the building arrangement provides for a students' gallery, have utilized the space under the gallery as a fresh-air fan chamber, a fan, heater, and air filter being arranged so that air is blown into the theatre in sufficient quantity not only for the theatre but the adjoining rooms of the suite.

It is desirable that the extract and fresh-air plants should be electrically interlocked so that both must be running together. If an air washer is used for the fresh-air plant, thermostatic control is essential to keep the humidity of the air in the theatre, usually maintained at a temperature of 75° F., at such a level that condensation does not occur when the external temperature is low.

Silent-running equipment is of importance, and where possible the extract and air-conditioning plants should be situated some distance from the theatres to ensure this.

Design for Sound-Picture Studios.

The necessity for absolute silence in sound-picture studios calls for special measures in design to eliminate all noises from ventilating equipment.

Firstly, metal construction should be avoided entirely for ducts, and brickwork, concrete, timber, and similar construction should be used, whilst sound-absorbing material of the acoustic board type should be used for lining ducts.

The maximum air velocity in the ducts should be 500 ft. per minute and through inlets or outlets 150 ft. per minute. The maximum total water gauge at which the fans should be operated should be no greater than 0·75 in., equivalent, with most makes of centrifugal fan, to 3,500 ft. per minute peripheral velocity.

The air changes desirable for picture studios is 10 per hour, for anything less than this will not enable the studios to be kept cool. Large quantities of heat are generated over short periods by lighting units, and the cooling problem is very difficult. To ensure the best conditions, a refrigeration plant is necessary in conjunction with the air-conditioning equipment. The cost of refrigerating plant generally leads to its not being adopted on the initial stages, but the system can be designed so that it may easily be applied later if required.

Apart from ventilation requirements, the acoustic problem should be given attention, and if necessary a sound-baffling chamber should be used to provide a large area of sound-absorbing material.

Ventilation of Churches.

Ventilating and air-conditioning equipment has been infrequently applied to churches, but there are perhaps some notable exceptions in the branches of the Church of Christ Scientist, built some ten years ago. It is proposed to deal at length with the design of such a system as applied to the church building in Fig. 53.

From a general inspection we see that there is a main church, a side chapel, and vestries to be dealt with. It will be found with buildings of this nature that the most suitable scheme provides for a radiator heating system to deal with the heat losses from the structure and to maintain 55° F. internally ignoring air change. The ventilation side of the equipment should be able to keep the internal temperature up to 60° F., when used in conjunction with the radiator system.

Seating capacity should be used as a guide to deciding air quantities which should certainly be no less than 1,000 cu. ft. per hour per person, giving in our case for 530 seats, $530 \times 1{,}000 = 530{,}000$ cu. ft. per hour.

We will assume that the heat loss from the building when 32° F. externally is 300,000 B.T.U. per hour, calculated by the method previously explained, to maintain 55° F., a further 54,000 B.T.U. being required when 60° F. is maintained.

The air entering the building must therefore be at such a temperature that in cooling to 60° F. it can give up 54,000 B.T.U., so that the rise in temperature above 60° F. will be $\dfrac{54{,}000}{530{,}000 \times 0{\cdot}02} = 5°$ F. approximately. The figure of 0·02 is that amount of heat required to

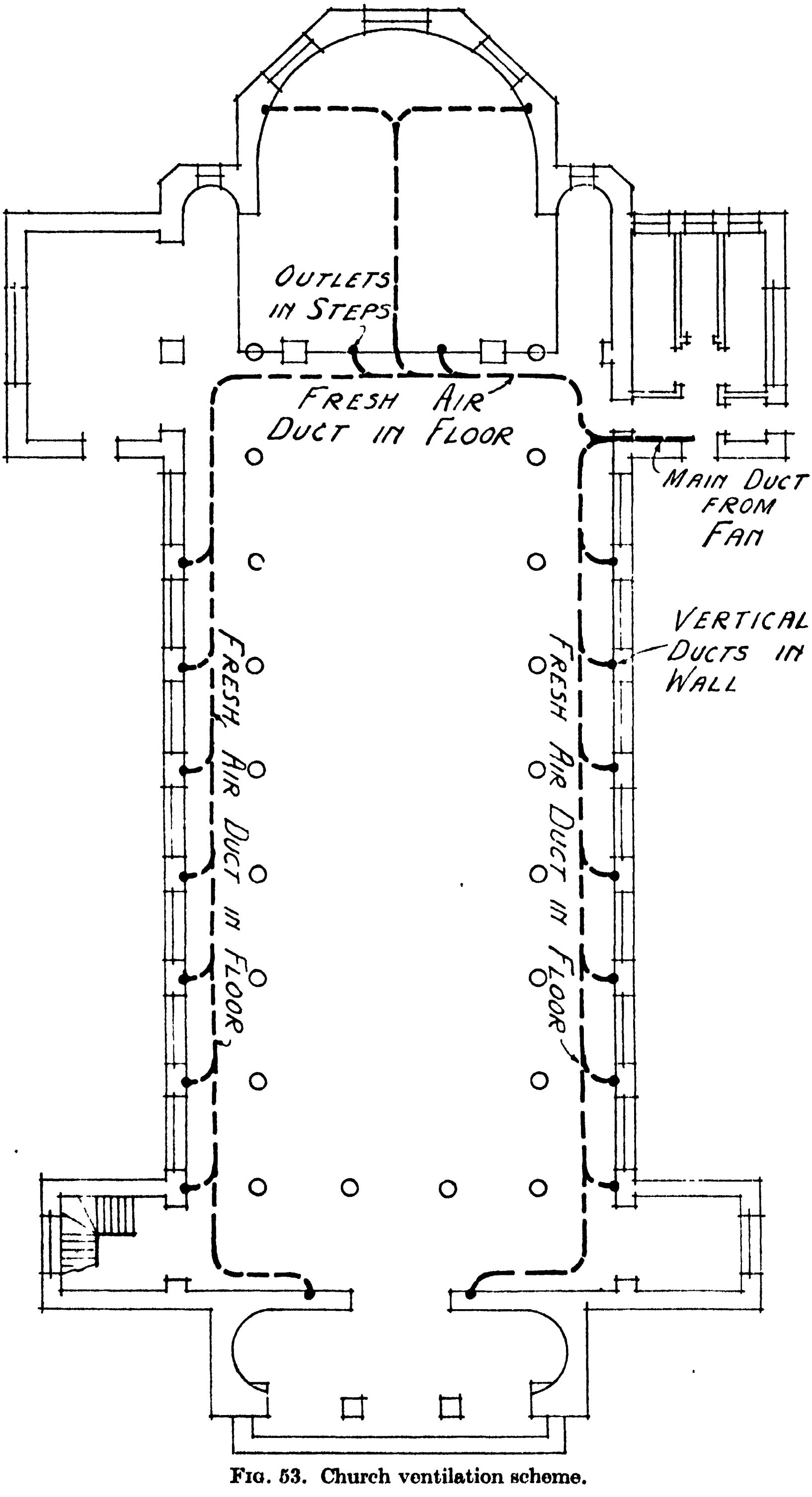


FIG. 53. Church ventilation scheme.

raise 1 cu. ft. of air at 70° F. through 1° F. The air must therefore enter at $60+5 = 65°$ F. It must be remembered that a further cooling would take place due to heat losses in ducts under the floors, which might amount to 4° F., so that in practice the air heater would raise the air to approximately 70° F.

The ventilating plant would consist of an electrically driven fan, an air filter or washer, and an air heater, the air being discharged through underground ducts to suitable rising ducts at the sides of the building.

In designing the ducts, the maximum velocity should be 1,000 ft. per minute, dropping to 300 ft. per minute in the smaller ducts communicating with grilles. The velocity through the free area of the grilles should not be greater than 60 ft. per minute.

The general arrangement of the scheme is shown in Fig. 53.

The boiler power required would be as follows:

Heat for raising air

(assuming no air washer)

$530{,}000 \times (70-32) \times 0{\cdot}02 = 403{,}000$ B.T.U./hr.

Radiator heating system $= 300{,}000$

Total 703,000 B.T.U./hr.

To this must be added any losses from pipes, and a margin to ensure easy stoking. If an air washer is used, advantageous for summer cooling, during the winter evaporation of moisture will be required under certain conditions, and this will represent a minimum further temperature rise to be imparted to the air of 6° F., with a correspondingly larger requirement to be added to boiler power.

The Heating and Ventilating Problems of Underground Car Parks.

The extreme congestion of traffic in the London streets has prompted many to put forward ideas for overcoming the traffic problem by various means. In considering the fundamental needs, the question of car parking is one which is actually causing far more trouble and discussion than that of traffic control. At definite hours in the day it is known that various parts become congested owing to what one might call a 'peak' traffic load. It seems, therefore, that the only sound suggestion for alleviating the congestion of traffic in the streets of London lies in entirely prohibiting the influx of certain classes of road traffic during what are now the 'peak' traffic periods.

The problem has been carefully studied by Mr. Thomas Spencer, his suggestion being to create a ring of underground car parks at strategic points round London. Naturally, as car parking at present takes place in the principal squares and similar places, those positions suggest themselves as suitable sites for the creation of underground car parks.

We are indebted to Mr. Spencer for permission to reproduce his pictorial drawings illustrating the proposed arrangement of such underground car parks, and Fig. 54 shows the plan suggested for serving the City of London, actually conceived to be placed under Finsbury Square. This underground car park would accommodate 500 cars on each floor, that is, 1,000 cars in all if the two floors were both constructed.

From the architectural point of view, apart from actual garage accommodation, the park provides for a general office for the issue of tickets, waiting-rooms, lavatories, and parcels office, whilst a refreshment room, dressing-room, book and paper stall, accessories show-room, and minor repair shops are also contemplated. The arrangement of exits and entrances is specially arranged to avoid interference with traffic passing along Moorgate, these being arranged on the side of the square farthest from the main street. The layout of the square which forms part of the scheme does not in any way spoil the amenities of the City, as it is designed in sympathy with the surrounding buildings. The architectural pavilion is available for use as shelters or for the accommodation of any ventilating, air-conditioning, or heating plant, whilst the square has generally been conceived as an Italian garden with paths, flower-beds, trees, and tennis courts.

As a site for a super underground car park to cope with the requirements of the West End of London, Mr. Spencer chooses as his site the Marble Arch entrance to Hyde Park, his scheme being shown in Fig. 55. This garage, whilst planned on substantially similar lines to the other, is intended to accommodate 1,000 cars on each floor, that is 2,000 in all, the dotted lines indicate the site under which the garage extends; it is wholly under the roadway.

This garage again is arranged with its external features in sympathy with the surrounding district. It presents difficulties, however, in the arrangement or placing of above-ground architectural features, which might be necessitated in connexion with engineering equipment.

Mr. Spencer's scheme, therefore, in a few words, prohibits private cars from entering Central London, it being necessary for them to park at the various points arranged, their owners finishing their journey into town by the ordinary public vehicles, thus entirely eliminating the traffic congestion problem.

It may be seen that the heating and ventilating requirements of such garages present problems of first magnitude.

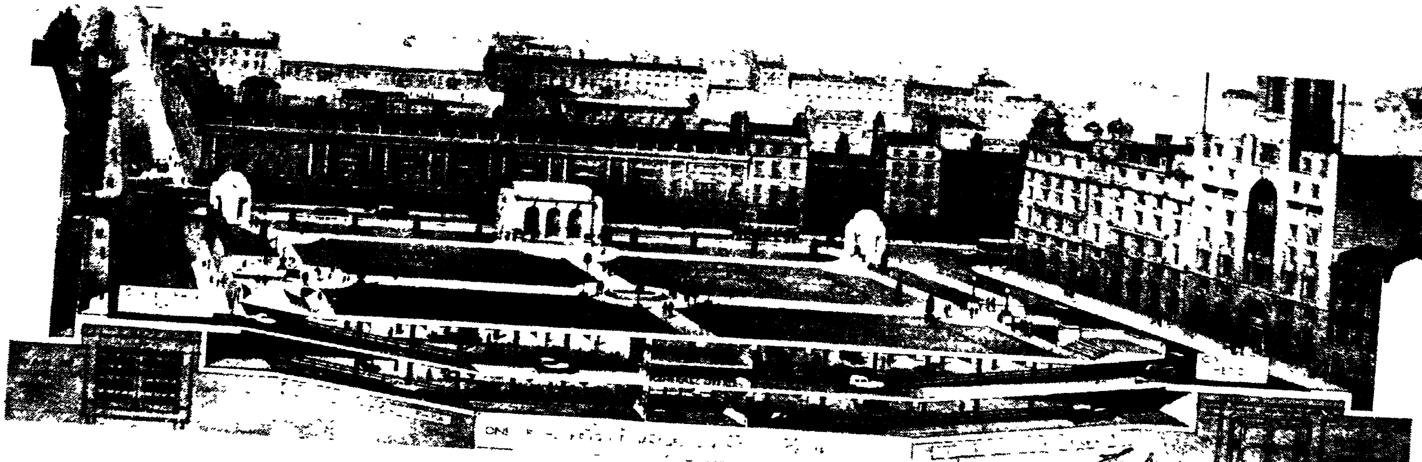


FIG. 54. Finsbury Square underground car park.

Firstly, it must be obvious that ventilation and air-conditioning is of primary importance. Although for long periods these garages might be full of cars, it would be extremely unlikely that more than a small proportion would have their engines running at any one time; on the other hand, there would be occasions, after theatres, for instance, when a large proportion of the cars would be moving out at the same time, and it is at such times that the need for efficient ventilation becomes apparent owing to the fumes emanating from the engine exhaust. Such fumes make it fundamentally important that large quantities of air should be moved in the garage, or, more accurately, that large quantities of fresh air should be introduced to absorb the fumes before passing out to the atmosphere. Not only must sufficient air be supplied for this purpose, but the quantity must be such that the concentration of injurious exhaust gases shall be at such a level as not to be deleterious to the health of drivers and the garage staff; the latter would, of course, be in the car park for hours at a time.

With the introduction of large quantities of air, a difficulty may immediately be foreseen in having to heat this air before introducing it into the park during winter months. Moreover, apart from any question of the amount of air which is entering, there is a definite heat loss through the structure if the internal temperature is maintained at a possible 55° F.

During the summer months an even greater problem is formed by the need of keeping the garage relatively cool, for apart from the fact that any air then introduced is likely to be at a high temperature, there is also a certain amount of transmission from the surrounding ground into the car park, and in addition there is the heat of the exhaust gases and engines to be taken into account.

During the whole of the year, any air introduced, particularly in the case of the car park situated in the City or similar districts, adequate provision must be made for filtering or washing the incoming air before it is introduced to the car park, as otherwise there would be an enormous amount of dust present to settle upon bodywork. All these factors present their own difficulties which can scarcely be appreciated duly until some idea of the air quantities is arrived at. When this point is settled it will be better understood what difficulties oppose the heating and ventilating engineer commencing upon a design of equipment for these underground car parks.

At the commencement of our discussion on the possible means of heating such parks, we would again repeat the architect's wishes that the scheme should be architecturally in sympathy with sur-

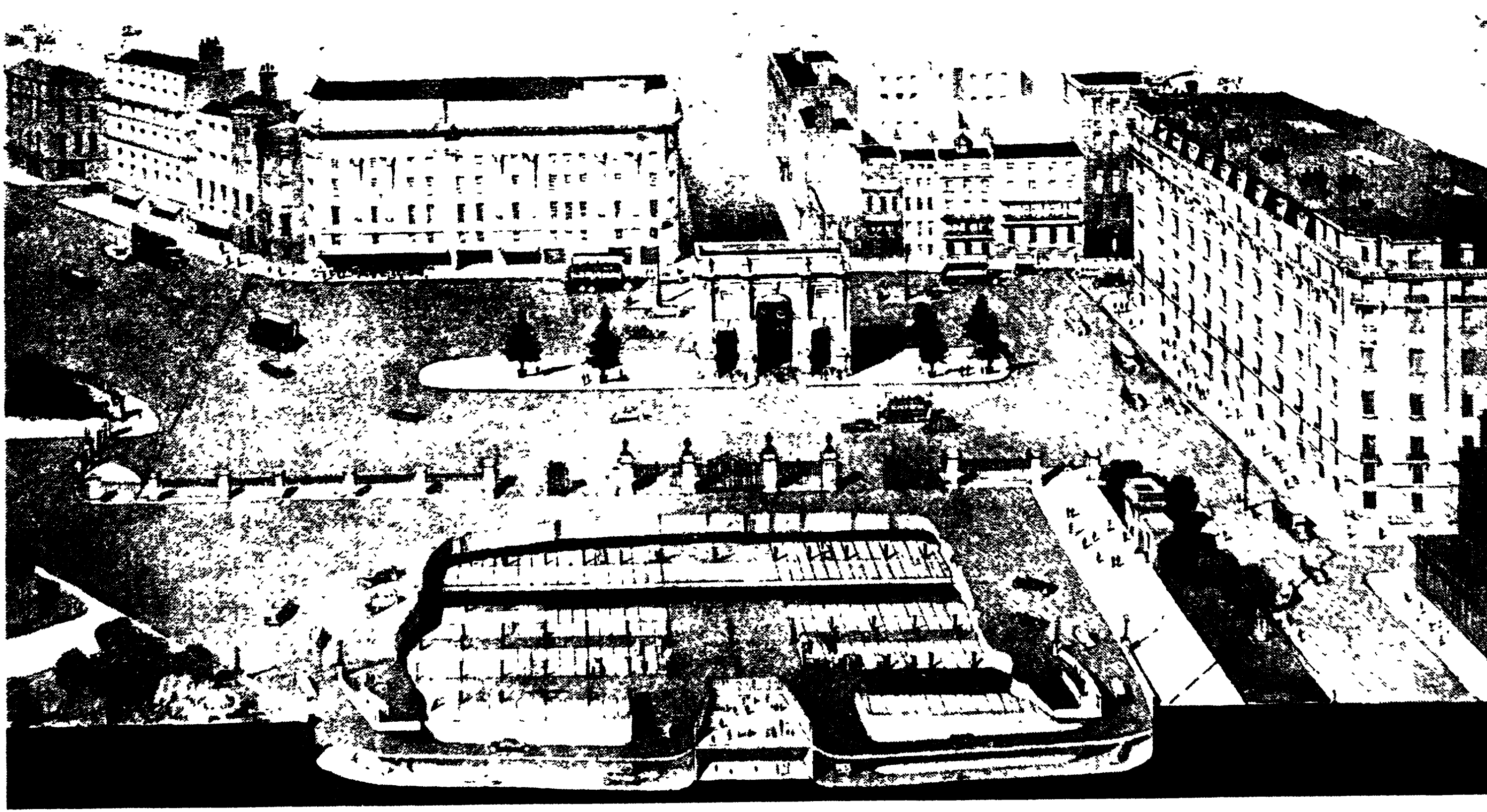


Fig. 55. Marble Arch underground car park.

rounding buildings. What, then, are we to do with the heating system, for obviously a chimney shaft is impossible? We can therefore suggest the two alternatives of gas or electric heating systems, but must, nevertheless, immediately rule out the possibility of electric heating because of the enormous cost of current required for such jobs, even if a thermal storage system were employed and current were obtained at specially reduced rates.

On turning to gas-heating systems, we immediately find a solution to the heating problem, which nevertheless provides many alternatives. We could employ ordinary gas-heated boilers of any known type, or alternatively use could be made of submerged combustion systems in which the gas is actually burnt in the heating medium. It is this latter system in the author's opinion which is ideal for the heating problem which is to be solved.

As a result of research on the air conditions of the Blackwall and Rotherhithe tunnels† it has been established that the rate of ventilation should be sufficient to keep the proportion of carbon monoxide down to 20 parts per 100,000 parts of air. It has been found that exposure to conditions where this proportion exists leads to injurious effect on health within 6 hours.

Henderson has found‡ that a relation exists between the product of the proportion of carbon monoxide and time of exposure, and the effect on a human being is as follows:

Time in hours × concentration of CO (parts per 100,000)

= 30 gives no appreciable effect;
= 60 gives just perceptible effect;
= 90 gives headache and nausea;
= 150 is dangerous.

The workmen in a car park would be occupied for perhaps 8 hours, so that allowing for the effect of exertion, 10 parts per 100,000 would be an acceptable figure.

The usual carbon monoxide content of exhaust gases reaches 5–9 per cent. of the whole.

It must be evident that it is almost impossible to give any definite basis upon which to decide the ventilation requirements of these underground car parks. Fundamentally, of course, it would be possible to calculate the quantities of various substances contained in the exhaust gases, and from a knowledge of the permissible degrees of concentration thus calculate the necessary air volumes, but

† *The Ventilation of Vehicular Tunnels with particular regard to those at Blackwall and Rotherhithe*, by C. J. Regan.

‡ *J. Industrial Hygiene*, vol iii, July 1921.

immediately there is a difficulty in knowing how many cars will have their engines running at any one time and how long these engines will be running. It would not be desirable to assume that the whole of the cars were moving at any one time, any more than it would to assume that half or a quarter were doing so. Again, one could not tell the type and horse-power of every car in the park, and even to assume an average would not prove an easy matter. From experience, however, of various classes of garage the author is led to believe that no system would be satisfactory providing for extracting less than the equivalent of 15 air changes per hour, and it is suggested that the fresh-air system should deliver the equivalent of 13 changes per hour.

In considering the problem in more detail, we will restrict our remarks to the larger type of garage, such as that contemplated at Marble Arch, which would have two floors each 400 ft. $\times$ 400 ft. $\times$ 8 ft. 6 in. high = 1,400,000 cu. ft. The total amount of air to be handled by the extraction system would, therefore, amount to:

$$1{,}400{,}000 \times 15 \times 2 \text{ floors} = 42{,}000{,}000 \text{ cu. ft. per hour,}$$

that is 700,000 cu. ft. per minute; this alone shows that the ventilation equipment required is on a very large scale.

It is inevitable that any car parks in this scheme should combine a petrol-filling station, and it is therefore proposed that the ventilation of such areas, being the only ones where petrol vapour is likely to exist in any serious concentration, should be completely isolated from the main ventilation equipment.

Moreover, particular care should be taken in the arrangement of the equipment to bear in mind the greater possibilities of fire-risks in these areas. The fire-risk problem must, of course, be also considered when designing the major ventilation equipment of the car park.

Without doubt serious difficulties would be encountered in London when the approval of Public Authorities is looked for in connexion with the mechanical equipment of these underground car parks, and it is likely that serious objection would be raised to the existence of discharge points from the extraction system at comparatively low level, a point possibly not of such great consequence at Marble Arch as it would be at Finsbury Square, which is within the Fire Brigade Danger Zone. In the author's opinion, however, there is no sound reason for prohibiting discharge at low level from the major portion of the car park, because not only is the amount of petrol vapour or other inflammable gases present likely to be existing in small quantities, but the large amount of air employed for carrying them away would give concentrations ensuring no danger. Moreover, if the suggestion of isolating the petrol area as far as ventilation equipment is concerned were carried out, various means could be provided for ensuring that no danger were present. Even so, the treatment of air inlets and outlets in a manner architecturally sound would provide no easy task.

We might mention that an underground car park has recently been constructed at Hastings, and in this particular instance the scheme evolved for the discharge of the extract fan is the use of a shelter surrounding the discharge shaft. Such a scheme might be considered for our particular problem. Let us consider, however, at this stage, what the area of a single discharge shaft would need to be. Assuming a velocity through the shaft of 1,000 ft. per minute, the area required would amount to:

$$\frac{700,000}{1,000} = 700 \text{ sq. ft.}$$

It will be seen, therefore, that a single outlet would become impracticable when the ideas of the architect and engineer are considered together, so that it would seem that a better scheme would be to split the building up so that it is served by two distinct plants, one for each floor. This suggestion would also be of merit in the event of fire commencing on any one floor, as there would be no direct connexion by way of ventilation ducts between the two floors.

Considering in further detail the petrol area, it is apparent that no public authority under any circumstances would allow the discharge of air taken from such areas at only a few feet above pavement or road level, and as such air quantities from this particular area would be low, any shaft required would architecturally tend rather towards a chimney, and is therefore impossible if of any height. It is suggested that instead of employing the normal slow air-discharge shaft, the air from which would leave at only approximately 1,000 ft. per minute, conventional practice should be abandoned completely. The discharge from this system should be arranged so that the air leaves at a velocity of 10,000 ft. per minute, equivalent to a final velocity pressure at the discharge of 6·25 in. W.G. Compared with the volume of air handled the necessary horse-power would be high, but this scheme would be a practical one for overcoming the discharge difficulty. It might be argued that winds would upset this system by blowing the discharge horizontally and dispersing it before it has any chance to reach comparatively high level, but it must be remembered that the discharge velocity is equivalent to a speed of 114 miles per hour, so that it is unlikely that a wind at 50 miles per hour could prove troublesome, for even so, the combined forces would only result in the high-velocity air column leaving the top of the shaft at an angle of approximately 45°. The discharge from this system might very conveniently be arranged in the centre of an existing gate-post, for the Marble Arch scheme.

In connexion with the petrol vapour problem, an alternative might be considered for the absorption of the vapours before discharging by means of a substance such as silica gel. (This, we might mention, is a hard, greasy material with the appearance of clear quartz sand made from sodium silicate and acid having the same chemical analysis as sand, but differing physically owing to its porous structure.) Silica gel is capable of absorbing, if placed above water in a closed vessel, vapour to the extent of 40 per cent. of its own weight, and it is, therefore, capable of absorbing substances such as petrol vapour, the efficiency of absorption depending entirely upon the concentration of vapour and the amount of water evaporated in the atmosphere which is being treated. It would appear, therefore, that there is a possibility of a silica gel absorption plant for serving the petrol area, and this method might therefore be kept in mind for detailed consideration should serious objection be raised to the high-velocity discharge method.

Yet another possible method exists in passing the air from the extract fan through a series of scrubbing chambers arranged on the weir principle, so that the petrol vapours are condensed and float on the surface of the water employed in the scrubbing chambers from which it may periodically be drained.

It may be easily found by calculation that during the winter months to maintain an internal temperature of 55° F. at any one of the large car parks necessitates balancing a heat loss of 3,500,000 B.T.U. per hour, this being principally represented by the losses through the walls, floors, and roof of the structure.

It must be apparent that, as an enormous volume of air is being introduced to the building for the purpose of ventilation, there would be no justification for putting forward any scheme for heating other than on the 'Plenum' system. We have already suggested that the Extract Ventilation Scheme should be capable of exhausting air at the rate of 15 changes per hour, representing a total volume of 42,000,000 cu. ft. per hour, and that the fresh-air system should introduce 13 changes per hour, the difference of two air changes being represented by natural infiltration for which allowance was made in the heat loss figure mentioned above.

On the basis, therefore, of 13 air changes we may calculate the temperature above the desired internal temperature which must be given to the air before it enters the car park in order to give up the necessary 3,500,000 B.T.U. per hour as follows:

$$\frac{3{,}500{,}000}{36{,}500{,}000 \times 0{\cdot}02} = 4{\cdot}8^\circ \text{ F.}$$

The temperature, therefore, at which the air must enter the car park must be $55+4{\cdot}8 = 59{\cdot}8$, or shall we say approximately 60° F.

At this stage it must be clearly apparent that such quantities involve heating equipment of very considerable capacity, for assuming that the whole of the fresh air is taken from an external source at 32° F. and heated to a temperature of 60° F. before entering the car park, the amount of heat which must be imparted to raise it to this temperature would be:

$$36{,}500{,}000 \times 0{\cdot}02 \times (60-32) = 20{,}500{,}000 \text{ B.T.U. per hour.}$$

Air recirculation is not normally approved of by various authorities in this country, but is nevertheless perfectly sound, provided adequate means are arranged for cleaning the air before it is reintroduced to the building. It is suggested that 75 per cent. of the air should be recirculated, under which conditions instead of the air entering at 32° F. it would in effect enter at 54° F. To raise 36,500,000 cu. ft. of air per hour from 54° to 60° F. requires only 4,400,000 B.T.U. per hour, which immediately brings the capacity of the heating equipment to a reasonable level, particularly as this capacity is easily arranged for by means of the submerged combustion gas-heating system to which we have previously referred, very little floor space being required for the equipment. If this system were employed, the flue area for exhaust gases need only be approximately 50 sq. in., owing to the fact that the exhaust takes place under high pressure from the plant itself. As far as flue height is concerned, this need only be high enough to satisfy official requirements, and it is suggested that it should only be as high as any of the architectural features arranged for by the architect. In any case, if there is any objection to the exhaust gases being passed to the atmosphere at such a low level, it would be a comparatively simple matter to provide for washing and chemically treating them to remove or neutralize any undesirable impurities.

Just as some means must be provided for heating during the winter months, so must some provision be made during the summer for keeping underground car parks reasonably cool. It is not suggested that complication of the air-conditioning plant should be looked for for this reason. In the author's opinion, it would be impracticable, and certainly not a commercial proposition, to design cooling equipment to maintain low temperatures by the use of a powerful refrigeration plant, firstly, owing to the initial cost of the plant, and secondly, owing to its high running cost.

It is suggested that the only provision that should be made for cooling is the employment of an air washer arranged either for the use of recirculated water or cold water run to waste or employed for the general purposes of the car park. It must be remembered that the gases in motor-car exhausts consist mostly of carbon monoxide and dioxide, and, as we are proposing recirculation, some provision must be made for absorbing the bulk of such gases from the air before it is returned to the car park.

With the volume of air which it is proposed to employ for this scheme, it is not thought that the amount of gases present would be any considerable proportion, so that the mere fact of introducing 25 per cent. completely fresh air during the whole of the running of the ventilation plant should give a condition of sufficient dilution to prove satisfactory in practice. It is difficult to support this statement by any sound argument, owing to the impossibility of knowing the number of cars which would be running their engines at any particular time, the horse-power of these cars, and the period for which they would be running together.

Should, on closer inspection of the problem, it prove necessary to make other provision for absorbing some of these gases, the only practical solution would consist in the provision of an additional scrubbing chamber other than the air washer itself in which the air, that is the recirculated portion, is subjected to a chemical spray before passing into the room. Whilst it would be possible to design such a scrubbing chamber on a theoretical basis following on calculations, it is suggested that it would be far wiser, in the event of it proving necessary, to carry out small-scale laboratory experiments in order to judge of the efficiency of the different methods of coping with the problem. At this stage I would suggest the strong advisability of providing ultimately for such equipment even if it were not installed in the preliminary stages. It would be quite a simple matter to install a simple air-conditioning equipment in such a way that the purifying plant could be added at very short notice. In any case, the conditions existing without it would not prove to be so bad that the car park could not be used.

Referring again to the question of cooling, as must be well known, by means of an air washer it is possible to cool the air to a point very close to its entering wet bulb temperature, even when the spray water is recirculated. Air movement is likely to play a far more important part in the apparent coolness of the car park than actual temperature conditions, and for this reason high-speed air inlets might well be provided for.

On closer examination of the problem it might be thought better to pass the whole of the extracted air to the atmosphere instead of recirculating any, in which case an alternative method of heat economy offers itself, consisting in the use of an air economizer through which the exhaust gases are passed before entering the atmosphere. This economizer would comprise a chamber with cross flues, the exhaust gases passing through the flues and the fresh air brought into the air-conditioning plant passing through the chamber itself, so that the heat contained in the exhaust air is employed for partially heating the fresh air. We can obtain some idea of the efficiency of this method as follows:

Let x = temperature to which incoming air is raised; then we may state the following equation:

Heat gained by incoming air rising from 32° to x° F. = heat loss by extract air in falling from 55° to x° F.

Then

$$36{,}500{,}000 \times 0{\cdot}02 \times (x-32) = 42{,}000{,}000 \times 0{\cdot}02 \times (55-x)$$
$$-23{,}250{,}000 \times 728{,}000x = 46{,}200{,}000 - 840{,}000x$$
$$1{,}568{,}000x = 69{,}450{,}000$$

whence $x = 44°$ F.

The amount of heat required to raise 36,500,000 cu. ft. of air from 44° to 60° F.

$$= 36{,}500{,}000 \times 0{\cdot}02 \times (60-44)$$
$$= 11{,}650{,}000 \text{ B.T.U. per hour}$$

as compared with 20,500,000 without heat transference.

We see, therefore, that approximately half of the heat required is saved by the use of a heat economizer. No allowance for moisture content was allowed, of course, in the above calculation. This method only saves half of the heat as compared with approximately three-quarters saved by recirculation.

We have now to decide upon the major problem of air distribution for the car park as a whole. Reviewing the problem, we see that there are certain structural limitations which tend to make the distributing system conform to certain fixed lines. The network of beams in roof and floors practically rules out the possibility of extensive network of ducts in the floor or at roof level. In our opinion the most successful system of distribution would be that of introducing the fresh air at as high a level as possible and at high speed along all sides of the car park, extraction being arranged underneath at as low a level as possible also round all sides, but extraction should take place at as low a velocity as possible. This, therefore, will ensure, firstly, that the fresh air should be thrown as far as the centre of the car park, and secondly, low-speed extraction

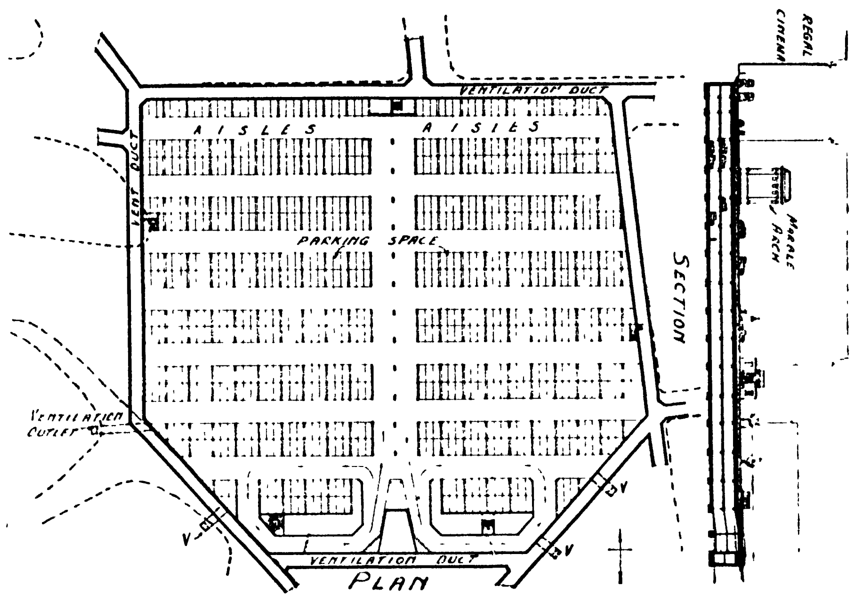


FIG. 56. Plan and section of Marble Arch car park.

at low level ensures extraction from as wide an area as possible at low level where the bulk of any fumes would collect. There is no reasonable justification, in the author's opinion, for the practice of putting in extraction openings at high and low levels as has been customary in several large garages recently; this is not necessary. This system of air distribution would require large ducts running right round the car park on each floor which might be formed as large corridors easily accessible for all purposes.

Referring again to the introduction of fresh air at high velocity, it is suggested that the actual velocity should be in the neighbourhood of 4,000 ft. per minute, and that in order to obtain such a velocity the inlet openings should be continuous around the car park and as shallow as possible, resulting in a thin sheet of air being thrown towards the centre of the park, and owing to the action of the extract system at low level gradually descending in horizontal strata.

One final point which must not be overlooked is the adequate control of all ventilation units in the case of fire, and such control should be fed by the suitable employment of large fire-dampers and electrically operated remote controls operating upon the fan motors. Such controls should be capable of being operated in the first instance by contact with high temperature, and secondly as ordinary remote controls under the supervision of the car-park attendants. This latter control should be so arranged as to do immediately whatever is required in the control of the fans by merely pushing one push button. Fig. 56, which is Mr. T. Spencer's layout of the Marble Arch Car Park, indicates the position of these ducts, which incidentally can also be employed for cables and similar purposes. The branch ducts indicated might be employed for connecting to the plant chambers containing extract and fresh-air plants in positions exterior to the car park itself. The installation of this plant well apart from the main park is desirable in view of the high fire-risk of such a building. These branch ducts serve to link up also with main cables and other services in surrounding streets.

Ventilation of Laundries.

Every laundry, small or large, has some serious problem either of ventilation or steam removal, the solution to which is essential to smooth and efficient working of the plant.

In all laundries there are two serious factors which affect production, namely, excessive heat and moisture. Unfortunately, very few laundries have installed equipment to overcome these difficulties, although the plant required is by no means elaborate or costly to run.

When considering the ventilation equipment the laundry may be divided into three distinct groups:

(1) The wash-house.
(2) The ironing room.
(3) Sorting and packing departments.

It is the wash-house, perhaps, which provides one of the most serious problems, owing to the large amount of steam that is likely to be given off from the machines. It is true that the presence of this steam does not directly affect the washing, but it must be remembered that the exceedingly humid atmosphere which will exist is not only unhealthy to the operatives, but owing to condensation on the cold roof and walls of the building is bound to lead to undesirable conditions in the room.

There are several means by which we can overcome the difficulty of excess free steam or vapour. The first method that suggests itself is to remove the steam at its source, so that it has become customary to provide large hoods over the washing machines, connected by ductwork to a centrifugal extract fan arranged to draw away the whole of the vapour from its source. There is another point, however, that requires even closer attention.

It must be remembered that the cause of condensation of steam is the fact that air at any particular temperature has a definite capacity for moisture. For instance, the temperature of the air in the wash-house may be about 70° F., the air being practically fully saturated with moisture at this temperature. When air which is fully saturated is lowered in temperature its capacity for moisture is less, so any moisture in the air over and above that which it can hold at a lower temperature must be condensed out during the cooling. It is quite likely that the temperature of the external atmosphere will be in the neighbourhood of 30° to 40° F., so that there is a possibility of any air in contact with the external building structure cooling a very large amount.

Where, therefore, an extract ventilation system is provided to withdraw vapours from their source, careful thought must be given to the means of taking in the incoming air, for if this is merely allowed to filter in from outside the building through the structure it will do so at a low temperature and will have very little capacity for moisture. This results in the extract system losing much of its efficiency, for a considerable proportion of the vapour is bound to find its way into the body of the room, and unless heating batteries are provided at defined air intake points a foggy or steam-laden atmosphere will result.

It is often possible with a small laundry, where steam is only coming off at one particular point, to treat this as a separate zone, arranging an air heater and fan near this point, taking in air from outside and heating it to about 100° F., blowing it into the zone where the steam prevails, ordinary extraction cowls being provided in the roof. This system ensures freedom from a steam-laden atmosphere.

In the ironing room the great difficulty is to maintain a reasonably cool atmosphere, which is not an easy problem, for a large amount of heat is generated and needs to be absorbed. Again, at this point another important fact to be considered is that the room should be perfectly clean, and that any air entering should be thoroughly washed, so that a ventilation system comprising an electrically driven centrifugal fan drawing air from an external source through an air washer is necessary. This consists of a sheet-metal casing in which water-spray nozzles form a very fine mist through which the air has to pass, and also a bank of corrugated eliminator plates to take out any free moisture not absorbed by the air and to ensure removal of more impurities.

From the air-conditioning plant, the air is taken through a series of sheet-metal ducts, and in the case of the ironing room is blown down branch ducts to outlets arranged at low level where a reasonably cool atmosphere is to be maintained. The air-washing plant is capable of removing 95 to 98 per cent. of the solid impurities contained in the air, so that very little dust can actually pass into the building. When it is desirable to try to maintain an even higher degree of purity, a viscous air filter is arranged on the inlet side of the air washer.

In the case of the up-to-date laundry the ventilation equipment is capable of changing the air 20 times per hour in the wash-house, 16 in the ironing room, and 12 to 13 times in the sorting and packing rooms.

Boiler-House Ventilation.

The need for efficient ventilation is felt not only in the large boiler-houses such as one finds in connexion with power stations, but even in the smallest of boiler-houses containing only one small cast-iron sectional type of boiler. Although every care may be taken in the insulation of pipes and boilers, wherever possible, with efficient non-conduting composition, there nevertheless remains a considerable proportion of heat which tends to cause a temperature rise, whilst in many cases the boiler fronts are responsible for the emission of relatively high temperature radiant heat. It might be thought possible to calculate exactly what amount of heat is likely to be present, and on this basis to determine the quantity of air that must be introduced from an external source to ensure cool conditions, but unfortunately, whilst this would be possible, the conventional basis of design tends rather towards a stereotyped practice which dictates a fixed number of air changes per hour, varying between 15 and 40 according to the designer's taste and the amount of money available for the ventilating equipment.

It must be here stated that few boiler-houses can be successfully ventilated unless the air changes are approximately 20 per hour.

Apart from actual ventilation requirements, it must be remembered that a large volume of air must enter for combustion purposes. For average coal the theoretical quantity of air is 12 lb. per lb. of fuel, and in actual practice the quantity of air employed will vary between

18 and 30 lb. per lb. of fuel according to the type of fuel and the method of combustion.

A ventilating system for the boiler-house will, therefore, as a fundamental basis have to provide at least sufficient air for combustion purposes, but it must in addition to this provide air for ventilation and cooling, which must find its way both in and out of the building either by mechanical or other means.

There are a number of factors controlling the total amount of air to be supplied, for, firstly, if the boiler-house is large in relation to the boiler plant, it is evidently not necessary to supply so much air as one would in the case of a very cramped boiler-house giving little head room.

It is possible to determine a suitable volume of air by an empirical formula, namely:

$$V = k(C+R),$$

where V = volume of air required to be introduced in cu. ft. per minute,

C = cubic contents of boiler-house in ft.,

R = rating of boiler in thousands of British Thermal Units (B.T.U.),

k = coefficient varying between 0·2 and 0·3.

We may see how this formula adjusts the quantities of air by taking firstly as an example a boiler-house 50 ft. by 40 ft. by 20 ft. high, containing two Lancashire boilers each having an evaporation of 8,500 lb. per hour.

The volume of the boiler-house = $50 \times 40 \times 20$ = 40,000 cu. ft.

The approximate combined rating of the boilers = 17,000,000 B.T.U.

Substituting in the formula, in this case using the coefficient of 0·2.

$$V = 0{\cdot}2(40{,}000+17{,}000)$$
$$= 11{,}400 \text{ cu. ft. per minute.}$$

As a second example this same boiler-house might only have been 15 ft. high, and in this case the volume of the boiler-house would have been 30,000 cu. ft., so that substituting in the formula in this case a less advantageous value for k of 0·3 would then give:

$$V = 0{\cdot}3(30{,}000+17{,}000)$$
$$= 14{,}100 \text{ cu. ft. per minute.}$$

Apart from natural ventilation systems comprising air inlet and outlet shafts and grilles, with which we do not propose to deal, the mechanical ventilation system may be applied in several ways, of which the most efficient is without doubt that comprising a fresh-air ventilation system introducing cool air into the boiler-house and, in addition, an extract ventilation system extracting and delivering to the external atmosphere that proportion of the fresh air not used for combustion purposes. In some cases, to lower the cost of installation, the extraction system may be arranged on gravity principles, assisted by a slight additional pressure on the fresh-air inlet fan, but this, of course, has the inherent defect that prevailing winds may tend to disarrange the working of the system.

Fig. 57 shows a typical arrangement of fresh-air and extract-ventilation equipment for a comparatively large boiler-house, having

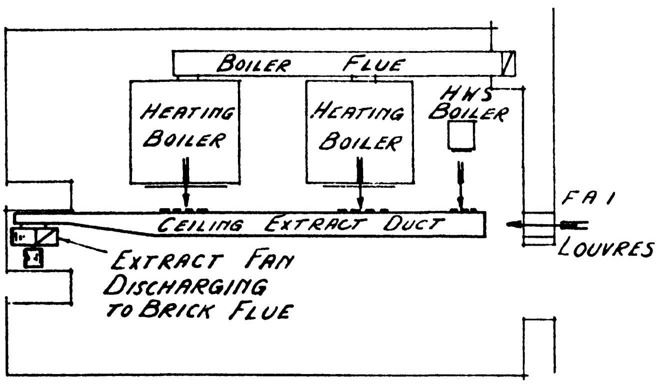


FIG. 57. Small boiler-house.

in this instance an air washer for cooling the entering air. Fig. 58 is of interest, as it shows a simple method of providing mechanical ventilation in a boiler-house of the smaller type such as we might expect to find in schools and similar buildings. In this case a mechanical extraction system is provided, exhausting air from the front of the boilers and delivering it by a main brick flue to the external atmosphere. A fresh-air inlet louvre is provided at the one side of the boiler-house through which air for ventilation purposes and combustion is drawn.

Whilst it is possible to design a system sufficiently accurately to be sure of introducing any desired quantity of air, one must remember that the only source of air is the external atmosphere.

During the winter months, when the external temperature is low, such air is sufficiently cool for any purposes of boiler-house ventilation, but during the summer months it must be remembered that the outside atmosphere may reach a temperature of 90° F., and for this reason would be comparatively useless for maintaining cool boiler-house conditions. It is possible by the use of an air washer to provide adequate cooling.

The amount of cooling obtainable by the use of an air washer when the spray water is recirculated, as is the normal practice, is limited to the extent that the air can at the most only be cooled to the wet bulb temperature at its entering condition, which means that if the external temperature is 90° F. and the relative humidity of the atmosphere is 50 per cent., the air could be introduced to the boiler-house at approximately 75° F., which would give an appreciable cooling effect.

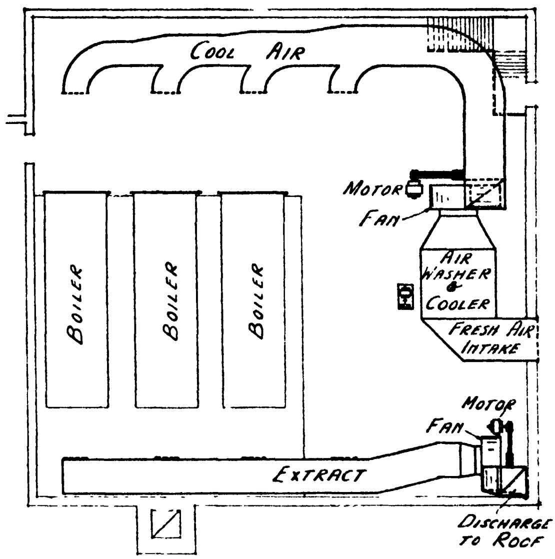


Fig. 58. Large boiler-house ventilation scheme.

For the larger building employing large quantities of cold water continuously for general purposes, it is possible to dispense with the spray-water pump on the air washer and to run water from the main to the sprays at a temperature of 55° to 60° F., the water afterwards being taken from the tank under the air washer to service-storage tanks in any part of the building. When this arrangement is adopted the air can be cooled, depending upon the quantities of water normally used, to approximately 65° F., and in such cases the subsequent boiler-house temperature could be in the neighbourhood of 70° F.

At this stage we must refer to the value of air movement as a factor affecting the efficiency of ventilation. Whilst large quantities of air may be introduced to the boiler-house at a comparatively low temperature, if it is allowed to enter at low velocities the full effect cannot be obtained, and therefore fresh air inlets and ducts should be designed to throw the air downwards on the basis of it leaving at a velocity of 1,000 ft. per minute, the air outlets preferably being adjustable for direction. A cool current of air blowing across the body definitely gives greater comfort effect than a still air condition at the same temperature. This explains why a higher velocity should be used in this case, a practice normally not possible in ordinary buildings where individual requirements will not tolerate draught.

The modern boiler-house with its oil firing and white glazed brickwork is intended to be a perfectly clean part of the building, and for this reason air must not be introduced straight to the boiler-house without some efficient means of filtration to remove dust, a necessity very apparent in busy cities or factory areas.

The air washer previously referred to would remove over 90 per cent. of dust in the incoming air as part of its normal duty, but in instances where the cost of air washer for cooling purposes is not justified, the use of a viscous filter can be recommended.

We need hardly mention, in connexion with the motors driving fans, that for boiler-house work it is very desirable to employ a motor of the totally enclosed type, preferably pipe-ventilated, because the amount of dust which is in many cases present invariably leads to trouble with an ordinary ventilated type of motor. An ordinary totally enclosed type will not be too satisfactory because of the high temperatures which are in some cases likely to be experienced. Arrangements can be made for passing a small amount of air delivered by the ventilating fan through the motor windings for ventilation purposes, an arrangement which proves to be very economical.

Chapter Four

INDUSTRIAL AIR CONDITIONING

As we shall use many expressions peculiar to the science of air conditioning, both when referring to particular industries and when giving typical calculations, we intend now to give some attention to the meanings of the various expressions.

Humidity is the moisture in the form of water or vapour which is mixed with air. The maximum amount of vapour which a given space will hold depends upon the temperature and not upon the presence of air or any other gas. That is to say, a vacuum at a given temperature would contain the same amount of vapour as if it were filled with air at the same temperature.

Air is said to be saturated when it has mixed with it the maximum amount of moisture permissible at its particular temperature.

The actual humidity of the air may be stated to be the number of grains of moisture contained in one pound of a mixture of air and vapour.

The relative humidity is the ratio of the actual amount of moisture contained in one pound of air to that which one pound of air would contain if fully saturated at that temperature.

The expression 'R.H.' will be used to indicate relative humidity, and it may be mentioned that this will be expressed as a percentage.

Dew-point temperature is the temperature corresponding to saturation or 100 per cent. R.H. Air cannot be cooled below its dew-point temperature without producing a contraction of volume, a certain amount of moisture being condensed out, this amount being actually the difference between the amount originally contained and that permissible at the lower temperature to which the air has been cooled.

Knowing the temperature of the air and the weight of water contained in one pound, it is possible to determine the actual R.H. equivalent to this condition.

For example, at a temperature of 60° F., one pound of air can contain a maximum weight of moisture of 77 grains. If it is found that air contains only 32 grains of moisture, then the 'R.H.' will be

$$32 \times \frac{100}{77} = 41 \text{ per cent.}$$

One says that air is humid or dry according to whether the relative humidity becomes nearer to or farther from 100 per cent. It therefore follows that, as the degree of saturation varies enormously with the temperature, the absolute humidity, and consequently the 'R.H.', varies also with the temperature. Air at a low temperature may be very humid when containing only a minute quantity of water vapour and at a high temperature very dry when containing a large amount of vapour. Thus the atmosphere contains a far greater quantity of moisture in the summer than in the winter, although the relative humidity is lower, because, the temperature being higher, the air is farther from its saturation point than in winter.

Nomograph for Air-conditioning Calculations.

There have been many graphs published correlating the various factors involved in air-conditioning and drying calculations, but all have been of too complicated a nature, so that the author feels that his nomograph for psychrometric calculations, Fig. 59, is of definite value in eliminating the complexity of curves of which the Harding, Carrier, Grosvenor, and similar charts are comprised.

It is an established fact that air in any condition of dry bulb temperature and humidity, provided its wet bulb temperature remains unaltered, has a constant total heat content above any datum. Similarly, air with a fixed dew-point has a definite moisture content, irrespective of dry bulb temperature and humidity. The psychrometric nomograph comprises, therefore, scale A, on which dry bulb temperature and dew-point temperature are read, and scale B, giving the moisture content in grains of one pound of dry air, both these scales being incorporated on one vertical line. Scale C, which is obliquely inclined to the others, on which relative humidity, henceforth referred to as R.H., is read, and a further line graduated with scale D giving the total heat in B.T.U. of one pound of dry air saturated with moisture, and scale E giving the wet bulb temperature.

EXAMPLE 1. If, for instance, we have a sample of air at 70° F. dry bulb and 50 per cent. R.H., by placing a straight edge across 70° F. on scale A and 50 per cent. on scale C we read at the intersection with scale E that this represents a wet bulb temperature of 59° F., whilst the same intersection reads on scale D a total heat content of 26 B.T.U. per pound.

It is observed from scale B that the moisture content of fully saturated air is 110 grains per lb. At 50 per cent. R.H. it will contain $110 \times \frac{50}{100} = 55$ grains, and against this figure on scale B the corresponding dew-point temperature on scale A is read as 51° F.

EXAMPLE 2. If dry bulb and wet bulb temperatures are known, a line from these on scales A and E respectively intersects scale C at

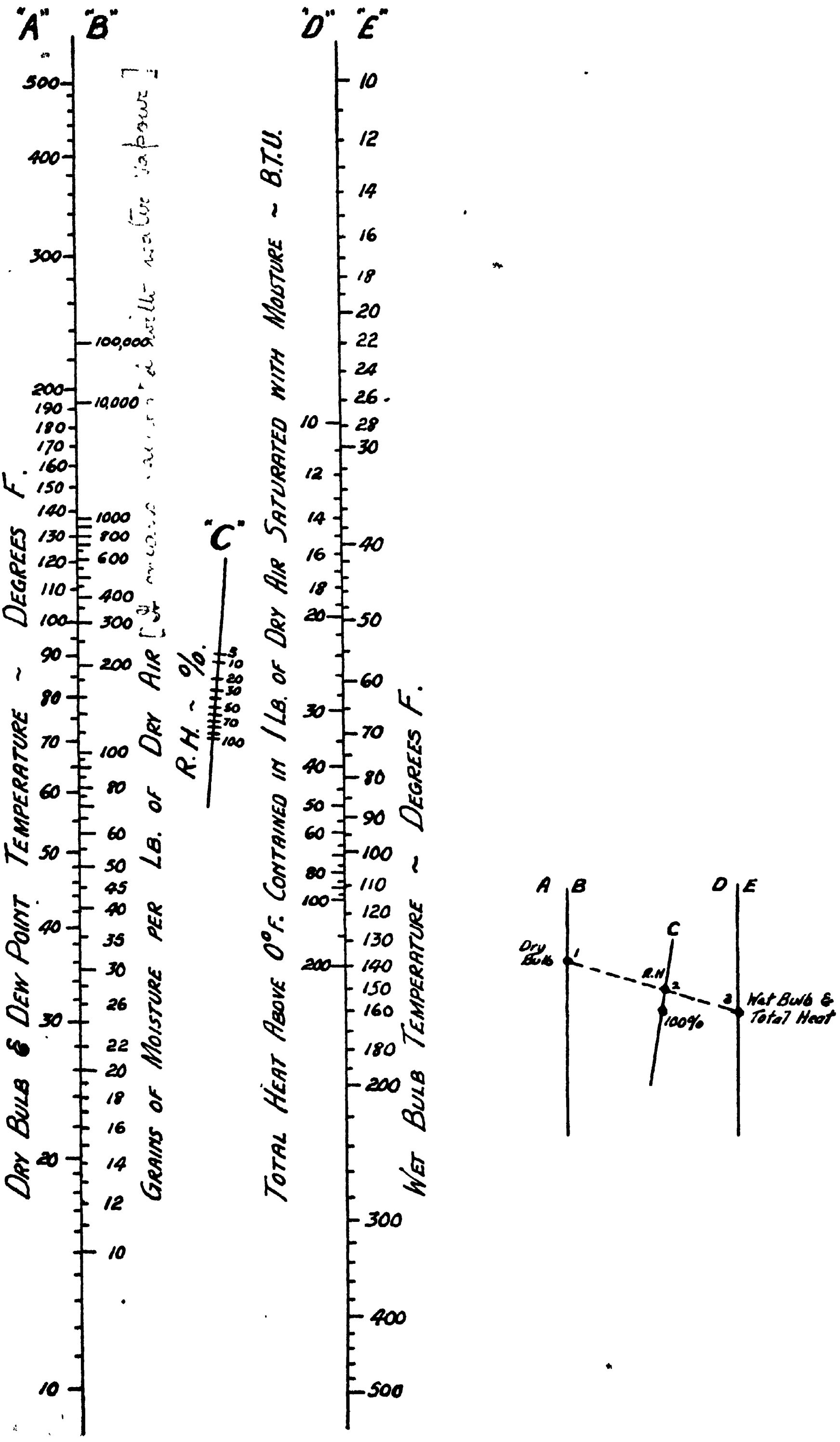


Fig. 59. Psychrometric nomograph.

the corresponding relative humidity. Thus 80° F. dry bulb and 60° F. wet bulb is equivalent to 30 per cent. R.H.

EXAMPLE 3. If air is introduced into a drying-room at 110° F., having been raised in temperature from 80° F. when it had a R.H. of 40 per cent., and it is not to leave the room with a R.H. higher than 70 per cent., we can trace out the conditions on the chart.

Firstly, the intersection of 80° F. and 40 per cent. shows the wet bulb temperature on E as 64° F. When the air is raised to 110° F. the moisture content will not alter. The maximum moisture for 100 per cent. R.H. at 80° F. is read as 160 grains per lb., but at 40 per cent. R.H. it would only contain $\frac{40}{100} \times 160 = 64$ grains per lb., with a corresponding dew-point on A of 55° F.

At 110° F. and fully saturated it could contain 440 grains per lb., so that in heating up the R.H. has dropped to $\frac{64}{440} \times 100 = 14\frac{1}{2}$ per cent.

The line through 110° F. on A and $14\frac{1}{2}$ per cent. on C shows the total heat as 36 B.T.U. per lb. on D. During the process of drying the total heat will remain unchanged, so that the line through 36 on D and 70 per cent. on C gives 81° F. on A, the temperature at which the air leaves the drying-room, when the maximum moisture content is 162 and the actual $\frac{70}{100} \times 162 = 113$ grains per lb. Each pound of air passing through the room can therefore absorb $113 - 64 = 49$ grains of moisture.

There are, of course, other factors involved in drying systems, but such points are dealt with in Chapter V.

Air Conditioning in the Artificial Silk Industry.

Every industry has its own requirements determined in many cases by years of patient study of the effect of various air conditions. It is not possible to deal with more than a few of these, but the information given is indicative of the general nature of the basis of air-conditioning problems.

Perhaps there is no industry in which air conditioning is more important than in the manufacture of artificial silk, and we propose, therefore, to consider the manufacturing processes and air-conditioning requirements of artificial silk mills in order to show how the manufacturing process has to be studied to evolve satisfactory air-conditioning equipment to deal with all conditions.

There are four principal types of artificial silk, the manner in which each is manufactured depending to a large extent upon the method used for forming the cellulose, which is the base of all types of material, into solution and making it solid again to form a thread. The cellulose used is obtained from cotton, cotton-waste, or wood-pulp, whilst straw and ramie have also been used. The manufacturing process, which is almost identical for each type of silk, may be classified under the following main headings:

(1) Cellulose, preparation.
(2) Cellulose, solution.
(3) Filtering of solution.
(4) Liquid forced through nozzles into a spinning bath.
(5) Coagulation in the spinning bath.
(6) A number of filaments twisted together to form a thread.
(7) Reeling of the thread.

The four types of art-silk are:

Chardonne or nitro-cellulose silk.
Cuprate or cuprous ammonium silk, sometimes called 'Glanzstoff' silk.
Acetate or acetyl silk.
Viscose silk.

Of these silks, viscose is the most frequently used, as it is the cheapest, having as its raw materials pitch-pine wood-pulp and cheap common chemicals.

The pulp is cut into sheets of appropriate dimensions, and after reaching the factory is conditioned to contain a definite percentage of water, and it is stored until required for use at a normal temperature, humidification being of little importance at this stage of the manufacture. These sheets are treated with caustic soda, forming 'alkali cellulose', which passes to hydraulic presses to remove the excess alkali. The 'alkali cellulose', in the form of a white cake, is disintegrated in a kneading and mixing machine, the cake being converted into a white fluffy mass which is put into small iron drums and stored for about two days, this process being known as 'the ageing of the alkali cellulose'. In the ageing-room for the 'alkali cellulose' the chemical treatment of the wood-pulp is well advanced, and colloidal changes are experienced.

In the next stage the white fluffy mass is rotated for three to four hours in a sulphidizing churn, in a room maintained at a fairly high temperature, and is then treated again with caustic soda in a quick-working paddle mixer to form the final spinning solution, after which the viscose, as it is then known, is left to mature for three or four days before spinning. The best place to store viscose is a warm, dry shed at a temperature of 68° F., and under no circumstances should a cold, damp atmosphere be obtained in the viscose store if the spinning process is to be satisfactory.

There are two methods of spinning viscose:

(1) The viscose is forced by means of small force-pumps at a pressure of five atmospheres through glass or platinum nozzles, or 'spinnerettes', into a bath containing a solution which has sulphuric acid as its main constituent, where it is solidified to form a thread. In this case, the silk leaving the bath is wound on bobbins, washed, dried, humidified, twisted on a twisting machine, and then reeled on a reeling machine into the form of hanks.

(2) The other system is known as the 'Centrifugal Pot' spinning system, combining in one operation the forming and twisting of the threads. The threads are cross-wound in the form of truncated cones into the spinning cans, and then reeled on reeling machines.

The final treatment from this point is identical for both systems, with minor modifications in the drying. All sulphur is removed from the silk in a sulphide bath, it is washed, soaped, whizzed off in the hydro-extractors, opened, dried, and humidified, and is then sorted and packed ready for dispatch.

From these notes it may be seen that artificial silk manufacture is more of a chemical than a mechanical process, and, consequently, temperature and humidity control must be of far greater importance than with any other textile. In the first processes a variation of temperature may mean a loss through spoilt material, whilst later, humidification difficulties may cause losses at a far more valuable stage.

Although the earlier processes may apparently have been successful, it is in the actual spinning that the faults become noticeable, and, moreover, faults are commonly developed here if the material has hitherto been perfect. In the viscose spinning-rooms humidity is required in order to eliminate the crystalline substances, which in dry air form upon the thread and machines, resulting in frequent breakages of the thread, while the temperature should be maintained fairly high. After the spinning it is usual to maintain a very humid atmosphere and a high temperature through the processes of twisting, reeling, sorting, and grading.

To the average person, perhaps, it may seem strange that the amount of moisture contained in the atmosphere can influence the finished artificial silk, but the expert knows that lustre and resistance to breaking depend to a great extent upon the control exercised over the atmosphere in which the material is worked. Apart from this factor, one must also consider the effect of the air conditions upon the operatives employed in the mill. The object to be achieved, by a combination of humidification and ventilation, is to obtain a uniform and regular temperature and hygrometric condition of the atmosphere in the rooms where the silk is manufactured, and to create by artificial means a special atmosphere which will be as favourable as possible to the working fibres and will at the same time ensure the best conditions of health and hygiene for those engaged in the rooms. The special atmosphere required must remain absolutely constant irrespective of the season and climate, a condition which cannot be obtained naturally in any country, so that air-conditioning equipment becomes an absolute necessity in the industry.

From the hygienic standpoint it should be remembered that air which is excessively dry has an exceedingly deleterious effect upon health, for the respiratory organs, which are normally lined with a slightly bactericidal secretion, are found to be dried up in such an atmosphere. Chronic troubles affecting the nostrils and larynx arise, and persist in spite of the strictest personal hygiene, and, moreover, because of the excessive evaporation taking place over the whole of the body, the mill-hands often have a feeling of cold, although the air may actually be very warm.

During the summer months there is a tendency for mills to become overheated, due to the heat from the sun and from the machinery in the mill, where air-conditioning equipment is not used, and in cases such as this the temperature of spinning-rooms may become as high as 87° F., with a relative humidity at this temperature of 55 per cent. In some mills the heat dispersed from the machinery is far in excess of that required for warming the mill on the most severe of winter days. It must be apparent that an efficient air-conditioning equipment must be capable of either heating or cooling the air and adding to or removing moisture from the air as required, the whole of the equipment being automatically controlled by any tendency to a change of temperature or humidity in the mill.

When the air from outside the mill is taken into the humidifier, during the summer it is cooled due to the evaporation of moisture, and when the spray water is recirculated by the pump this cooling effect is limited to such an extent that it is of very little use, unless large volumes of air are provided, so that in most plants it is necessary to cool the spray water by passing it over a cooling coil used in conjunction with a refrigerating plant. It is usual to cool the spray water to a temperature such that, in rising to the saturation temperature of the air leaving the apparatus, the water absorbs the whole of the heat which must be removed from the air to cool it to the degree required.

In many cases the air entering the mill contains an excess of moisture which must be condensed out in the spray chamber by contact with the cool spray water.

The humidifying equipment is also capable of removing and depositing in the apparatus dust or similar impurities contained in the air. It may be mentioned that the highest degree of solid impurity contained in the air for industrial towns is 2·0 lb. per million cubic yards, and if we accept this figure, it may be seen that in a plant serving an artificial silk spinning-room capable of producing 4 tons of silk per day, where the volume of air supplied might be about 170,000 cu. ft. of air per minute, the amount of solid impurities deposited as sludge would be over 6 cwt. per month.

Air Conditioning in Cotton Mills.

In the preparation, spinning, and weaving of cotton air conditioning plays a very large part in the equipment of an efficient mill. It will be interesting to consider some of the reasons why an air-conditioning plant should be necessary.

(1) The necessity for maintaining the humidity at the correct point to ensure that the thread is closely spun in the case of spinning processes.

(2) Humidity is necessary to overcome the tendency to the presence of static electricity, which in itself tends to produce a loose thread, bristling the fibres.

(3) The temperature must be maintained at that point found best for ensuring that the natural gums and dressings are in the correct state for working.

(4) The temperature must be maintained constant to eliminate any tendency to expansion or contraction of the machines, which would be the case if the temperature were always fluctuating.

(5) Air conditioning is necessary to compensate for the enormous amount of heat generated by the machinery.

(6) The health of workers must receive some attention.

(7) The necessity of complying with statutory orders relative to temperature and humidity. Moreover, the strength of the thread must obviously be considerably influenced by humidity variations. There is another point to be considered in most air-conditioning systems, and that is the extent to which the material being manufactured will absorb moisture.

Considering the necessity to compensate for the amount of heat emanating from machinery, let us take as an example a spinning-room, 130 ft. long, 120 ft. wide, and 16 ft. average height, that is to say, with a cubical content of 250,000 cu. ft. In such a room, assuming 20,000 spindles on the spinning machines, there would be 350 h.p. at least, of which, say 90 per cent., or 315 h.p. would be dissipated as heat, equivalent to, on the basis of 2,540 B.T.U. equalling 1 h.p.,

$$315 \times 2{,}540 = 800{,}000 \text{ B.T.U. per hour.}$$

During the winter, when the external temperature of the mill is down to 30° F., the heat loss will be perhaps only 650,000 B.T.U. per hour, so that it may be seen that the amount of heat available from the machines will be more than enough for the winter requirements. This fact may be taken full advantage of in a well-designed air-conditioning system by extracting air from the spinning-room and utilizing it for the heating and ventilation, after it has been purified, of other parts of the factory.

In the summer, however, the presence of the enormous amount of heat can prove serious, unless proper provision is made to absorb it by the introduction of cool air from the air-conditioning system. If, for instance, the amount of air passed into the mill were only equal to an air change of four times per hour, which many have advocated in the past, it may be seen that the air will be raised in temperature as follows:

$$\frac{800{,}000}{4 \times 250{,}000 \times 0{\cdot}02} = 80^\circ \text{ F.}$$

Even with the use of an air washer cooling the incoming air to 50° F., the internal temperature would be 90° F., so that it must be evident that very careful thought must be given to the problem.

Let us assume, therefore, that the condition required in the mill will be a temperature of 75° F. and 60 per cent. R.H. Any air-conditioning system, operating without refrigeration, will be able to cool the incoming air to the wet bulb temperature of the external atmosphere. Assuming this to be 85° F. and 40 per cent. R.H., equivalent to 67° F. wet bulb temperature, we may see that there is a margin of 5° F. between the incoming and outgoing air, so that the quantity required to absorb 800,000 B.T.U. per hour would be

$$\frac{800{,}000}{5 \times 0{\cdot}02} = 8{,}000{,}000 \text{ cu. ft. per hour.}$$

This is equivalent to an air change in the mill of

$$\frac{8{,}000{,}000}{250{,}000} = 32 \text{ times per hour.}$$

By supplying this quantity of air it would be possible to maintain the correct temperature, but we must also consider the R.H.

A condition of 75° F. and 60 per cent. R.H. is equivalent to a dew-point temperature of 60° F., whereas we have found above that the air-conditioning plant, operating on evaporative cooling, can cool the air to only 67° F., so that it is apparent that if the R.H. is to be maintained correct, some artificial cooling is necessary, in the form of refrigeration.

Statutory Requirements as to Humidity and Ventilation.

In view of the difficulties experienced in maintaining perfect conditions for the workers, and at the same time for the produce, it became necessary for some statutory protection to be extended to the workers to ensure that the conditions, as far as ventilation and humidity were concerned, were not deleterious to their health.

As a result, it has been established that in no case shall the wet bulb temperature in a weaving-shed exceed 78° F., and that there shall be no artificial humidification when the wet bulb temperature is 75° F. or more.

It has been decided also that hygrometers shall be fixed in all sheds, and that readings shall be taken and tabulated regularly.

Temperature and Humidity Requirement for Cotton Manufacture.

Many research workers have investigated the desirable temperatures and humidities for different cotton processes, and the principal information is derived from B. A. Dobson.

The principal points to be noted are that, according to the 'denier' or size of the cotton, so the temperature and humidity best suited to the working of the thread will vary, as also it will according to the nature of the cotton and the country in which it is produced.

In the earlier part of cotton-goods manufacture, the raw cotton is first thoroughly mixed, and then opened, and in these two processes, which are only preliminary to the spinning operations, humidity is of little importance, provided a reasonably equable temperature of about 70° to 72° F. is maintained. In the carding processes, however, a humidity up to 60 per cent. at a temperature of 75° to 77° F. is required.

These temperatures and humidities only give, of course, the general conditions required, and in any case it is advisable to discuss very carefully each proposition with the mill-owner before definitely designing the air-conditioning system, to arrive at the type and denier of thread which is to be produced.

Air Conditioning in the Manufacture of Wool and other Textiles.

The remarks which have been applied previously in regard to air conditioning as affecting artificial silk and cotton manufacture apply equally to all processes relative to the handling of wool, jute, hemp, flax, etc., with one or two obvious exceptions peculiar to the particular processes through which these textiles must pass.

Referring particularly to wool, which is probably next in importance to cotton, we have practically identical problems with which to deal. In the case of wool, however, what has particularly to be contended with is the greasy nature of the material, a factor which is not present to such a large extent with cotton. Again, with wool the presence of static electricity is far more likely to be felt in incorrect atmospheric conditions, whilst, unlike cotton, wool definitely loses its strength as the humidity of the surrounding air increases. It is for these reasons that combed wool is spun in very high humidities, but the reverse for carded wool.

Wool, as is generally known, undergoes two distinct carding processes, with the object of, as it were, combing out the wool and laying all the fibres in one direction, and this must necessarily cause the production of enormous quantities of static electricity, which can only be overcome in a humid atmosphere. The carding processes will generally call for a temperature of 72° F., and a relative humidity between 75 and 80 per cent.

This condition will hold good for the whole of the preliminary processes to wool spinning, with very little variation, any difference employed in practice being only due to the personal opinions of the different mill-owners.

In the continuous spinning process it is usual to maintain an atmosphere of 72° to 76° F., and a humidity of 75–90 per cent.

In the case of carded wool, which undergoes different preparations, the temperatures required remain about the same, but the humidity is considerably lower, in the region of 60–70 per cent.

In the weaving processes which follow, care must again be taken in the case of wool that the hygrometrical conditions are correct, it being usual to maintain a temperature of about 65° F., with 60–65 per cent. humidity in the preparation stage, 70–80 per cent. for ordinary weaving, and 60–70 per cent. for weaving on Jacquard looms.

In the case of flax and hemp, it is found that the material will spin far better on machines if it has previously been handled in correctly humidified rooms. Even when it is customary for jute to be at a humidity of anything from 75 to 80 per cent., this figure remaining more or less constant throughout the operations, the temperature varies from 65° F. in the preparation stages up to perhaps 75° F. during the actual spinning process. With hemp the temperatures are about the same, but it is advantageous to maintain a higher humidity of anywhere between 80 and 90 per cent.

Jute, which is, of course, prepared from specially grown grasses, is employed extensively in the manufacture of canvas, rope, and similar articles, and also in inferior quality clothing materials when mixed with wool and cotton. Jute is handled in a manner practically identical in other textiles, only differing from the technical point of view as far as the actual textile operations are concerned, and the condition maintained as far as temperature and humidity are concerned may be taken to be exactly the same as for flax and hemp.

We have dealt at some length with the question of air conditioning in the case of artificial silk, or rayon, as it is known, but so far have given no attention to natural silk, which, of course, differs entirely in constitution from the ordinary spun silk, and, needless to say, has to be handled in an absolutely different manner, as it is received in the form of a cocoon. With natural silk there is a primary possibility of employing air conditioning in maintaining equable conditions for the growth and propagation of the silkworms, which should lead to better production of silk and far better quality, but it must be obvious that any such installation would, at any rate for a time, be purely of an experimental nature, although without a doubt there are interesting possibilities.

We have discussed the question of the effect of temperature and humidity upon artificial silk, and have seen how static electricity can cause considerable trouble in the operations, so that it must be obvious that in the case of real spun silk these troubles are greatly magnified, and it is necessary throughout the various operations to maintain humidities varying between 65 and 85 per cent. with temperatures between 65° and 75° F., according to the particular operation, if this trouble of static electricity is to be overcome.

Again, one must remember that during the various operations and manufacture of natural silk there are many cases where the cocoons, both in their natural and subsequent conditions, must be washed, and hence there will be in some of the preparatory stages a large amount of moisture available without going to the extent of supplying actual humidification plant, and it will therefore be apparent that in such cases sufficient ventilation, by means either of forced air 'inlet' or a combination of 'inlet' and 'extract', will be all that is necessary.

Air Conditioning in the Dry-Cleaning Industry.

We have been discussing the question of the application of air conditioning to the textile industry. Static electricity, which can prove to have a bad effect during manufacture, can be overcome by the proper application of air conditioning, and in a similar way in dry-cleaning works air conditioning is used to neutralize the static electricity which is produced by different types of cloth being rubbed together. It is, of course, well known that, provided the clothing is damp, no trouble will be produced due to static electricity.

The practice has been for some years in the dry-cleaning industry to humidify the room by the use of steam jets somewhat on the same principle as that formerly employed for textile mills, but again, the question of excessive rise in temperature which is caused by the evaporation of large quantities of steam has a deleterious effect in that the room temperature might be brought too near to the ignition temperature of the volatile solvent employed for cleaning purposes. This, of course, points to the necessity for using a modern type of air-conditioning plant in order to maintain a comparatively low temperature with fairly high relative humidity.

The period of the year during which trouble is most likely to be experienced is that in which cold nights and warm days are common, as during the night the external atmosphere is reduced to a fairly low temperature, and consequently its absolute humidity is reduced to a very low point, with the result that during the day, as the temperature increases, the relative humidity is very low, which will result in the whole of the clothing in the factory being dried excessively. To bring this dry clothing into contact with damp clothes which may be coming in with the collecting vans would be very advisable.

Even with the application of correct air-conditioning equipment it is not possible to prevent the actual creation of static electricity, which is unavoidable when wool, silk, or similar materials are rubbed together on open belting or tables. By maintaining 70 per cent. relative humidity in the cleaning rooms it is possible to ensure that the clothing becomes dampened, and thus acts as a conductor for the static charges, which may be said to be dissipated in the cloth itself without any likelihood of sparks due to static discharges. It is advisable that all clothing entering a dry-cleaning works should be kept in a preliminary room maintained at this humidity for about one hour before they are handled in order to ensure correct moisture equilibrium. In the final process, when the clothing is drying, the humidity must again be maintained fairly high.

Air Conditioning and its Application to the Cures of Asthmatic and Bronchial Complaints.

Professor E. Küster, as the result of extensive experimental work in Germany, has published a considerable amount of interesting information on the application of air conditioning to the alleviation and care of bad asthmatic and bronchial complaints. General medical practice has shown that in the case of asthma sufferers attacks of the complaint may be warded off if the patient is transferred either to a well-arranged clinic, or by transfer to regions where the climate is more suitable, and he has divided asthma patients into two main groups, namely:

(*a*) Patients whose attacks are brought on by unsuitably conditioned homes.

(*b*) Sufferers who are domiciled in unsuitable climates.

Following on many experiences it was found that considerable relief could be given if the patient were kept in a space with an air change of 10 times per hour with a temperature of 70° F. and about 25 per cent. humidity. Professor Küster in his experiments has worked on far more practical lines than othér investigators, and in his laboratory has devoted a room to the subject, supplying it with conditioned air which is passed through an air washer, which it leaves at approximately 70° F. with 65 per cent. R.H., being discharged into the room at a very slow speed. It was found that germs common to asthmatic and bronchial complaints remained in suspension in the room when the relative humidity was 60 per cent. for about 5 hours, whilst at 90 per cent. R.H. they remained in suspension for only 2½ hours.

Several installations are in course of construction in Germany for air conditioning the rooms of sufferers, but no results are yet available as to the advantages in practice. It should, however, point to the desirability of carefully considering this class of equipment in hospitals and sanatoriums.

Air Conditioning in the Printing Industry.

In lithographic printing, air conditioning and ventilation is of particular use. In the case of ordinary printing works air conditioning may be required for several purposes, which may be summarized as follows:

(1) General air conditioning and ventilation throughout the works.

(2) Cooling press-rooms and foundry.

(3) Preventing evaporation loss of ink, which can mean a serious financial loss.

With regard to this latter item, the author recollects one instance recently where a large firm of printers were looking to an air-conditioning system to save them many tons of ink per year, and it was actually found that the saving on the price of the printers' ink would very nearly repay the cost of the installation. In this particular instance, also, it was found that a large amount of type-metal was lost each week, and as constant supervision and checking had been carried out to ensure that pilfering was impossible, it could only be assumed that this loss was wholly or partially due to the metal being oxidized and carried off in the form of dust.

Whether or not it would have been possible for an air-conditioning system to have overcome this difficulty in any way is largely a matter of conjecture, but nevertheless there is the interesting possibility.

In the case of press-rooms, it is usual to provide a system comprising fresh-air ventilation and extraction, an air conditioner being provided on the fresh-air side to ensure adequate cooling of the rooms. Similarly, the whole of the processing rooms, such as the linotype-room, are also thoroughly ventilated, air changes varying from 2 to 10 times per hour, according to the particular room which is being dealt with.

In lithographic printing, as must be well understood, the greatest point to be considered is that the paper upon which the impression is made must not vary in its dimensions to any appreciable extent, as otherwise the impression is likely to be both inaccurate and blurred. Paper is influenced by the presence of moisture in exactly the same way as textiles are, that is to say, it varies in dimensions and in strength according to variations in the relative humidity of the surrounding air, and therefore it is of particular importance that the whole of the processing rooms in a lithographic printing establishment should be air conditioned, with temperatures and humidities maintained constant by means of automatic control apparatus operating in conjunction with the air-conditioning plant.

Apart from the question of the paper stretching or contracting due to the variation in the surrounding atmosphere, it has been found that constant hygrometric conditions are more likely to ensure that paper breakages do not occur. If one considers the action of moisture on a high-class paper it will be appreciated that, should the relative humidity be too high, the paper will absorb moisture, with a result that the individual fibres of which it consists commence to swell, which immediately leads to an irregular face on the paper, thus affecting the impression.

Another point worth mentioning is the influence which variations of temperature and humidity can have in colour-printing. It must be remembered that in most colour-printing processes a large number of impressions are required to complete each picture, and in each impression a different colour is applied. It will be apparent that the various impressions must be absolutely identical in their form if a clear and well-defined picture is to be accurately produced.

Notes on the Tobacco Industry.

In the tobacco industry air conditioning enters right from the earliest stages of production apart from any use in experimental curing-sheds, which are sometimes air conditioned. As must be well known, tobacco grows in comparatively small plants, the green leaves of which are harvested and cured to make the tobacco that we all know, and in the beginning when the leaf is picked it contains anything from 70 to 80 per cent. of moisture. The most noticeable change which takes place during curing is the gradual reduction of the moisture contained in the leaf. It is the manner in which this removal of moisture is carried out that decides not only the quality of the tobacco but the proportion of green leaf converted to saleable product. It must not be thought that curing the tobacco leaves consists only in drying out the moisture, because if the drying or curing process is carried out too rapidly and at too high a temperature the finished tobacco possesses properties that are in many ways deleterious.

The curing or drying of the leaf is commercially carried out in several ways on tobacco estates.

There are generally four distinct ways of curing the tobacco leaf:

(1) Fire curing.

(2) Flue curing.

(3) Air drying or curing.

(4) Artificial curing.

It is, of course, in the last method that the scientific application of air conditioning can be of great use.

The first method, that of fire curing, consists in subjecting the tobacco to the fumes of charcoal or other fires ignited in the curing barns. This method, of course, gives to the tobacco a very strong flavour, and is therefore not of use for the general brands.

The flue-curing method is that in which drying- or curing-sheds are heated by means of the gases from fires passed through flues running round the drying-room. In this method the aroma of the tobacco is not contaminated or affected by the smoke as it would be in fire curing.

Air curing consists in having the leaves hung in drying-sheds in which there is ample ventilation, the drying simply being effected by natural means. Until comparatively recently, however, the flue-curing method has been considered to be the most scientific, but with the advent of air-conditioning equipment with efficient automatic control arrangements the conditions which are known definitely to be required may more accurately and constantly be maintained, whereas with the flue-curing system large fluctuations in temperature and humidity were unavoidable.

In the drying of the tobacco leaf it is necessary to heat the rooms continuously from the time curing is commenced until all the leaf in the drying-room is thoroughly dry. It is possible for good tobacco to be completely ruined during the curing process, whilst leaves of an apparently poor quality can be considerably increased in value by scientific curing.

In the curing of tobacco there are three stages to be observed, namely, yellowing of the leaf, fixing the colour, and drying the leaf and midrib. Where, however, the whole plant and not the single leaves are harvested, yet a further stage is required to kill the stalk.

During the yellowing process the barn should be filled with leaves of the same texture and the same stage of ripeness, so that all the leaves in the barn are yellow at approximately the same time. When the barn is filled it is tightly closed so that the necessary temperature and moisture conditions can be maintained. At the beginning of the drying care is taken that a hygrometer is provided, level with the first or lower tier of tobacco, to indicate the temperature and humidity of the air in the barn together with any fluctuations during this stage of curing. At first the temperature is kept at the normal atmospheric temperature, and then gradually increased until the thermometer registers about 90° F. It is necessary that the lower temperature should be maintained at first, as a high temperature would kill the leaf prematurely before it had a chance to change from the original green to a yellow colour, and any tobacco so treated is practically valueless.

The temperature of 90° F. is maintained until the leaf begins to yellow at the edges and tips, when it is raised to 95° F., and held until the colour begins to spread. The temperature is then increased to 100° F. until the yellow becomes more pronounced. During this time the atmosphere of the barn must be kept moist to prevent the leaf from drying too rapidly, and the atmosphere must be kept such that the temperature registered by the wet-bulb thermometer will not be more than 3° to 4° F. below the dry bulb. If the wet bulb depression could be kept at 3° F. the leaf would then yellow more rapidly and uniformly.

Finally, when the leaf begins to show a distinctly yellow colour the temperature is increased to 110° F., until the leaf is practically yellow, and then raised to 115° F., until the colour is taken. During this increase of temperature from 100° to 115° F. the amount of moisture in the atmosphere of the barn is reduced until the wet bulb registers from 6° to 7° F. depression.

Curing of Turkish Tobacco.

During the curing of Turkish tobacco the leaves are first hung on laths and placed in the wilting-room, a room generally cool, free from draughts and dust, and fairly dark. In order to obtain the best results the temperature and humidity of the wilting-room must be under complete control. If the temperature is too low the desired changes will not take place, and if too high the leaf will wither quickly and fail to take on the proper colour. It has been found from practical experience that a temperature of 70° F. is conducive to the best results.

With regard to humidity, if the atmosphere of the wilting-room is too highly humidified the leaf becomes damp and covered with condensation and there is a danger of the tobacco becoming damaged from mildew. Similarly, if the atmosphere is too dry the leaves wither prematurely and the green colour will remain in the cured product.

It is considered that the relative humidity of the wilting-room should be about 85 per cent., that is to say, the wet bulb depression should be 2½° to 3° F. During the curing of the Turkish tobacco leaves there is again ample scope for the application of scientific air conditioning.

'Black Rot.'

During the fermentation, in particular of cigar leaf, there is likelihood of a disease known as 'black rot' attacking the leaf, the disease actually being caused by a fungus growth. The three factors which influence the spread of 'black rot' in fermenting tobacco are temperature, humidity, and air supply. By the use of air-conditioning equipment to maintain constant conditions it is possible to control the disease by bulking the tobacco in a room maintained at 110° F. with a moisture content of 25 per cent. It is necessary that the tobacco should be bulked as solidly as possible to exclude air.

Tobacco Leaf Drying Schedule.

Research in the yellow curing process for leaf tobacco has enabled a complete artificial curing process to be carried out in 16 days, the tobacco losing 71 per cent. of its weight during the curing. The drying schedule is as follows:

Period	*Hours*	*T.*	*R.H.*
A	42 to end of yellowing	76–91	69–85
B	28 to end of browning	91–6	78–86
C	24 to development of odour	96–7	85–93
D	72 to end of sweating	93–9	92–6
E	216 to completion	98–108	95–100
	382		

Moisture Content of Tobacco.

Green leaves contain a large proportion of water, this proportion varying according to the type and quality, climate and the nature of the soil, and also the time taken in curing. According to von Babo, the water contents of tobacco are as follows:

Type	*Water %*	*Dry material %*
Amersfort	89·72	10·28
Dutton	85·45	14·55
Ohio	90·77	9·23
Goundi (1st quality) . .	91·45	8·55
Goundi (2nd quality) . .	89·71	10·29
Vilot	87·95	12·05

According to the neighbourhood in which the tobacco is cured it still contains a large amount of moisture, up to 28 to 30 per cent. in France, and down to 20 per cent. in the Dutch Indies. The humidity of tobacco makes a great difference during actual working and manufacture. Tobacco when it contains less than 12 to 14 per cent. of water cannot be easily worked without producing waste, and below 8 to 10 per cent. it becomes extremely friable. Certain yellow tobaccos cannot be worked without very great loss except in workshops where the air is maintained at a certain humidity by artificial humidification. The dearest and most delicate tobaccos are often those which are the most sensitive to humidity variations.

It is proposed now to give a brief description of the different processes through which the tobacco leaf must go from the time it enters a modern factory to the time it leaves as a finished product.

Acknowledgements are due to the Balkan Cigarette Co. Ltd. for much of this information.

Leaf Opening.

Tobacco leaf comes into the factory either packed in bales or casks. The leaf is opened by hand, and in the case of Virginian tobacco the leaf has to be moistened in order that it may be more easily worked.

In large factories the operation of opening both Turkish and Virginian leaf is done by a machine which employs steam to moisten or to soften the leaf so that it shall not be broken by the revolving cylinders into which it is placed.

Cutting.

In cases where the leaf is opened by hand it requires further moistening before it is in a condition to be cut. In this country the question of the amount of moisture to be added to render the leaf suitable for this operation has been generally left to the man in charge of the department. He adds moisture according to the state of the weather, and there seems to have been little attempt to gauge the actual percentage of moisture added to the leaf. After the tobacco is cut and dressed, in the case of Turkish leaf it is now ready for working on the machine, but in the case of Virginian leaf it is panned, that is to say, the tobacco is placed on a hot plate and turned over until a considerable part of the moisture added for cutting purposes has been extracted.

The amount of moisture now remaining in the tobacco also seems to be a matter of speculation in most factories. The tobacco is dried until it is in the right condition for working on the cigarette machine. The actual condition of the tobacco varies according to the different types of cigarette machine, and most factories have their own ideas as to what that condition should be. In the case of Turkish tobacco the position is somewhat different. This class of tobacco is more hydroscopic, and although when it is cut to-day it may be in good working condition, if the weather turns damp it is quite possible by to-morrow that it may be so damp that it cannot be worked on the machine at all.

Drying this tobacco by the same means used in the case of Virginian only results in killing its flavour and aroma, and it is therefore obvious that in this type of manufacture the humidifying plant would be of great benefit. In Germany and America a great deal of work has been done along these lines, but Turkish manufacture in this country is so small in comparison with the large amount of Virginian tobacco handled that no one seems to have paid very much attention to this side of the business. Whether the cigarettes are manufactured by hand or machine, it is necessary that they should be further dried in order that they may be in a condition for smoking. In some of the large factories, however, the Virginian tobacco is so dried that, when it comes from the machine in the form of cigarettes, it can be packed straight away, without any further treatment.

This process, in addition to removing any surplus moisture that the cigarettes may contain, also causes the shreds of tobacco to shrink, and so leaves air spaces in the cigarette, which assist combustion. When the cigarettes have been dried to the required amount they are in many cases allowed to stand in a cool room at a temperature of 60° F. and 64 per cent. R.H. for some time before they are packed in order that they may gain what is recognized as the natural moisture content of the leaf. Manufacturers have paid fairly strict attention to the question of moisture content, both in the leaf when it arrives in the factory and in cigarettes exported from the factory on 'draw-back', for the reason that the moisture content of the cigarette plays a considerable part in the rate of 'draw-back' paid.

The average moisture content of cigarettes exported from this country is between 11 and 14 per cent. This variation depends largely upon the destination of the cigarettes, the type of tobacco from which they are manufactured, and the effect of the moisture on the smoking propensities of the tobacco. A great deal more, however, is done with regard to moisture, humidifying, and conditioning in the case of pipe mixtures, many of which carry up to 22 per cent. moisture content.

The Balkan Cigarette Company have had a special gas drier constructed, and the temperature of this room is kept constant by means of thermostatic control, which regulates the supply of gas to the heater. As the greater part of their output consists of Turkish cigarettes, the best results may be obtained by keeping the temperature in the drying-room somewhere between 70° and 75° F.

A similar heater has also been fitted in the stock-room, which keeps the temperature in this room at about 60° F.

The temperature and relative humidity of the storage-room has also been confirmed by Messrs. Godfrey Phillips, Ltd., who mention that fixed humidities are adhered to as closely as possible, but conditions are varied considerably to meet the fluctuations in demand, loading, condition of the raw material, and to a certain extent external conditions, it being almost impossible to define any ideal conditions.

According to M. Max. Vinges† the technology of cigarette manufacture is a very wide subject, embracing many things, of which air conditioning is one particular subject influencing greatly the whole process of manufacture. It is essential that the leaf or cut tobacco should be in a flexible condition in order that it may be properly worked and the minimum of dust produced, which is only obtained

† *Chaleur et Industrie*, Sept. 1934.

by the correct humidity at all stages of manufacture. The necessary humidity not only depends on the stage of manufacture but also to a large extent upon the type of tobacco which is being used. At some stages, where tobaccos are being mixed, in order to ensure the minimum of waste it is essential for each particular type of tobacco to be at a different humidity. In practice, however, it is usual to fix a mean relative humidity, found by experience, of the particular blend of tobaccos. Reference is made to the detailed requirements of two classes of tobacco, namely, the dark tobaccos extensively used in France for 'Maryland' and similar cigarettes, and also the Turkish and similar types of tobaccos.

In the case of the dark tobaccos the leaves arrive at the factory with a humidity between 9 and 22 per cent., the average value being 12 to 14 per cent. In order to soften the leaf and to allow the stems to be removed, the leaf is sprayed with water until the humidity is between 30 and 40 per cent., when the leaf is ready to cut. After cutting the tobacco is dried in a rotary dryer, from which, after it has been cooled, it leaves with a humidity of 18·5 per cent., and finally, after pneumatic transport to another department, the humidity is 17 per cent., in which condition it is allowed to remain for 15 to 20 hours in storage, and it is during this storage that the tobacco matures and develops an aroma, the humidity decreasing to 16 per cent., which is the correct condition for cigarette manufacture.

For Turkish tobacco the humidity content of the leaf varies considerably. In the beginning the leaves have a humidity between 9 and 14 per cent., the average of a blend being 12·5 per cent. It is usual in the case of countries charging import duties on tobacco for the bales to be dried so that duty is not paid on moisture content. In such cases the average humidity of a blend would be 10 per cent. For de-stalking and mixing it is usually found that 15 per cent. humidity is sufficient, whilst for cutting 15·5 to 17 per cent., and for the actual manufacture 14·5 to 15 per cent. is usual, whilst the humidity decreases gradually to 12 per cent. in the packing-room.

Temperature, humidity, and air velocity considerably affect the power of moisture absorption of tobacco, and research has shown that the minimum temperature at which manufacture can take place is 50° F., but at this temperature the leaf is very brittle. For the better Turkish tobaccos the maximum temperature to which the leaf can be subjected is 85° to 109° F., for some dark tobaccos 140° F., and for others 212° F. As far as humidity is concerned the maximum is usually 85 to 90 per cent., whilst the maximum air velocity at any stage in manufacture should not exceed 195 ft. per minute. These conditions, of course, concern only the actual tobacco leaf and take no account of the comfort of the operative.

As one example of the conditions maintained in the rooms at various stages of manufacture for a dark tobacco, it is found that the moisture content should be 30 per cent. for all stages up to the cutting stage, this corresponding to an air condition of 85 per cent. During the actual manufacture of the cigarette, 17 per cent. moisture content is required, the corresponding humidity of the air being 70 per cent. In the packing department, 13 per cent. humidity is required, necessitating 60 per cent. relative humidity of the air. It is interesting to consider how these humidity conditions are compatible with comfort in the factory.

With the usual conditions of air flow it is usual to fix temperatures so that the atmosphere provided is in the 'comfort zone', which for relative humidity between 60 and 85 per cent. requires temperatures of 64° to 74° F. in the first case and 64° to 71° F. in the second case.

For all tobacco manufacture, particularly for cigarettes of various types, the air-conditioning equipment must be capable of control over wide ranges of temperature and humidity, whilst automatic control should be able to maintain pre-determined conditions within narrow limits. Where thermostatic control is provided the thermostats should be fixed at about 3 ft. 6 in. above the floor, as this is the level at which the manufacturing processes take place.

Keeping Goods in Condition.

The keeping of cigars, or any other manufacture of tobacco, in condition is one which cannot be governed by rules, but is based more upon common sense and the circumstances. The basic idea, however, is to keep the goods in an atmosphere as nearly uniform as possible, with the thermometer in the neighbourhood of 60° to 65° F. and the R.H. over 60–75 per cent. As uniformity of temperature and humidity is the principal object, the doors of the case, cabinet, or vault should be kept closed as constantly as practicable. In extremely damp sections care must be taken against the goods absorbing too much moisture, and in extremely dry climates air conditioning must be employed. The chief consideration is to maintain as little fluctuation as possible to retain the natural humidity of the tobacco.

It goes without saying that cigars should be unpacked from the cases and placed on tables or shelves in the store as soon as they are received, assuming that the goods are in proper condition when they arrive from the factory. The first consideration should be to see that they are not allowed to dry out. Direct currents of air should be carefully avoided. It is well to provide stores with ventilation, as fresh air is always advantageous, but this ventilation can be on a very moderate scale, and should be so arranged as to prevent currents of air from falling directly upon the goods. Stoves, radiators, and all kinds of artificial heat should be kept at a distance. With proper care the natural moisture of the goods can thus be conserved.

If, in unavoidable circumstances, the cigars become dry, the dealer must take care that they are not brought back to their proper condition too quickly. It is this over-anxiety to recondition the cigars that causes the greatest trouble. In remoistening cigars hastily the outside stretches and expands, because it absorbs moisture more readily than the centre. The consequence is that the wrapper has a tendency to loosen, and when smoked the cigar leaks air. The process of bringing dry cigars back to their normal degree of moisture should occupy as nearly as practicable the same length of time as was required to dry them out.

The humidity of the container should be raised slowly so that as the goods absorb moisture the inside of the cigar will increase in moisture with the outer leaves.

Air Conditioning in the Tea Industry.

Like many other industries, the tea industry has ample scope for the use of air-conditioning and ventilating equipment, from the commencement of manufacture until the leaf is sent as the finished product to bulk store.

It will be interesting to consider the processes through which the tea-leaves pass, as we shall be better able to appreciate the advantages of air conditioning.

The processes in tea manufacture as generally practised in India to-day are:

(1) Plucking.
(2) Withering.
(3) Sorting green leaf and separation of Pekoe tips.
(4) Fermenting.
(5) Drying or firing.
(6) Sorting.
(7) Final heating before packing.

The third process is often omitted, depending upon the quantity of leaf handled.

One of the important processes of tea making is the withering. It is extremely important that this process should be under close control so that it may take place at a convenient hour.

It is inconvenient to have all the machinery waiting for the withering, and certainly not convenient that a planter should get up at one or two in the morning to call his coolies to work to suit the wither.

In any but continuously wet weather no artificial means are necessary. The leaf is spread thinly on meshed-bottom trays and exposed to the action of the air, and consequently withers perfectly. In continuously wet weather artificial means are sometimes required.

One of the objects of withering is that the leaf should be prepared for the subsequent processes of manufacture. If rolling were attempted before withering the result would soon be a mass of torn fragments of leaf, and the juice would nearly all be spilt and lost. About 75 per cent. of the fresh leaf consists of water even on a dry day, and nearly half of this has to be got rid of before the fibre of the leaf and stalk will stand the strain of rolling without breaking up.

It must be noted that without special apparatus for circulating the air it is unwise to place the racks too close. The withering is then only hindered instead of assisted.

In hot climates it is necessary to have cool withering-houses separate from the main factory. There can be no question that in any circumstances the cooler the withering process the better, provided that the process of withering is actually proceeding all the time.

There are occasional times when a series of continuously wet days and nights render withering in open sheds an impossibility, and leaf kept lying about in such circumstances suffers much more than might readily be thought.

The main factor is to be able to control the admission of air to the withering-rooms, so that the correct quantity of slightly heated or dry air may be admitted as necessary.

Dry air is required, and if some simple process could be employed whereby the whole of the air passing through the withering-room could first be divested of its moisture, heated air could be discarded with great advantage and a distinct advance made in tea manufacture. Under the usual conditions, with a moisture-laden atmosphere consequent on the rains some heat must be used, but the wise tea maker ensures that the heat used is never more than just sufficient for his purpose. Wet leaf must not be unduly forced, but it can bear a higher temperature than dry leaf, because rapid evaporation of the moisture from its surface reduces the actual temperature of the leaf itself, as will be readily understood from our knowledge of psychrometry, and 100° F. is not too great a temperature under such circumstances, provided there is a low humidity.

Withering-fans are of the greatest value, provided they are fixed in the proper places and used in a sensible manner. It is common to find powerful fans fixed in the outer wall of a factory, operating upon quite a small space, and discharging directly into the outer air. It is little wonder that in such circumstances the fans are practically a failure, so far as actual withering is concerned, especially on any garden where all the extra heat generated in the factory is required to help withering. When the fans are placed as mentioned above they draw the whole of the heated air out of the factory in a few minutes, and so hinder instead of helping the wither. In this position they are, of course, useful for ventilating the factory, but for little else. They could assist the withering, however, if they were run once an hour for a few minutes only.

In order to obtain the best results from withering-fans, each fan should be made to operate upon a large area, and must be so placed that the air from the discharge side may be either sent directly out of the windows into the open air, or may be diverted into another room, where it will pass through a series of other racks before eventually passing out through open windows at the opposite end. In very wet weather and a cold climate the fans may be made to circulate the air throughout the withering-rooms rather than to discharge it outside. At such times a certain proportion of the moisture-laden air will make its way out here and there to be replaced with new air from the lower story, and so the air will be kept in reasonable condition while withering proceeds.

Before leaf is brought in all fans should be run for half an hour in order to thoroughly ventilate the withering-rooms. Immediately the leaf is spread, the fans may be set going again and kept going for the greater part of the night, or perhaps all night, the air admitted to the leaf being brought direct from outside or from the lower floors of the factory, according to circumstances.

With artificial withering the wet leaf should be exposed to a temperature of between 105° and 100° F. for the first 20 minutes or half an hour to carry off the wetness, then the temperature of the hot current of air should be reduced to about 95° F., and in the case of fine leaf the temperature may be further reduced to 90° F. when nearly done.

The moist warm-air system of withering is claimed to obviate the objection which has been made to the use of air specially heated for the purpose in a furnace or stove, as described in referring to the dry warm-air system, as the air used for withering will be charged with moisture by a repeated circulation over the leaf. In this way the evils said to be attendant upon the use of specially heated air are avoided in this arrangement by using the same air over and over again. The moist warm-air system of withering, which is the invention of S. C. Davidson, is set to enable the proper withered condition of the leaf to be produced with practically no evaporation of the naturally contained moisture, and the leaf when thus withered is claimed to produce a better quality of tea than when withered by the system hitherto in ordinary use.

The process consists, briefly, of subjecting the freshly picked or green leaf to the warm but, at the same time, non-drying influence of moisture-laden or nearly saturated air, at a temperature of 90° to 100° F., the leaf being gradually limped or withered, without appreciably evaporating its natural moisture content, and as the cells of the leaf when thus withered will still remain well distended with their juices, and also retain a very delicate texture, they will be much more easily turned during the subsequent rolling process than if they had been withered in the customary way, by the evaporation of from 20 to 50 per cent. of their moisture, which would have the effect of causing their cellular tissue and the substances lining the cell-walls to become shrunken and leathery in texture.

One method of producing this desirable form of wither is to expose the leaf to the action of a current or circulation of moisture-laden, warm air on trays, webs, or racks, suitably arranged for this purpose in a chamber or compartment, the sides, top, and floor of which are so constructed as to be as air-tight as possible, one or more inlets being provided for the moisture-laden air, which after circulating amongst the leaf in the withering-chamber can pass out of it through one or more outlets, or through a heating apparatus, and be reheated and returned into circulation in the withering-chamber.

In order to keep the warm air in the withering-chamber, or the withering-drum, moisture laden like a vapour bath and at a constant temperature of from 90° to 100° F., conduit pipes connected with the inlet and outlet ports are provided for conveying the exhaust air of the withering-chamber to and from an intermediately situated air-heating apparatus fitted with a fan, which draws the exhaust air from the withering-chamber or withering-drum, and causes it to pass through the heating apparatus and back again.

It will be seen that the same air circulated over and over again through the withering-chamber in this manner would rapidly get saturated with the vapour rising from the leaf, so that it would soon become incapable of absorbing any further moisture, and as this mixture of air and vapour is an effective conveyor of heat, it would act satisfactorily in carrying the heat from the air heater into the withering-chamber, and keep it up to the required temperature of 90° to 100° F. Any arrangement of air-heating apparatus, such as steam, or hot-water pipes, or air-heating stoves, can be used for heating the circulating air current.

The rolling-room should be situated in a cool part of the factory, away from the direct rays of the sun, and arranged so that the windows and doors may be thrown open to let in cool fresh air from verandahs or shade of some kind.

If the rolling-room forms a part of the main floor of the factory in common with the drying-room, it is very desirable to have the two departments separated by means of partitions of single brick wall or of lath and plaster. Similarly, it is necessary to separate this room from the engine-room or any other place where the heat is generated.

The time necessary for fermentation varies with circumstances, so that no absolute rule can be laid down. It may be anything from 2 to 5 hours (exclusive of the time occupied in rolling). Sometimes on gardens at a high elevation and in a cold climate very high-class tea has been made from a fermentation of 13 hours. When the climate is hot the time taken is less, but in any case, every effort should be made to keep the fermenting-room from getting hotter than 85° F.

It is recommended that no attempt should be made to wither in driers unless it is done at a maximum of 100° F. If, however, it is done at a high temperature a light liquor and a tea of the ordinary flavour will be the result. It is better to wait patiently until the leaf withers than to force it, and spoil the tea. Perhaps a fan would prove one of the best auxiliaries in tea manufacture to expedite withering.

Designing Air-Conditioning Equipment for Factories.

We will now consider the possible methods of dealing with the air-conditioning requirements of the factory illustrated in Fig. 60, which may be taken to represent a cotton-spinning mill. It will be assumed that the class of cotton being worked calls for a condition of the mill atmosphere corresponding to 72° F. dry bulb and 55 per cent. R.H. This condition must be maintained in summer and winter, irrespective of external weather conditions, and it will therefore be assumed that the extreme summer conditions of the external atmosphere are a dry bulb temperature of 90° F. and 70 per cent.

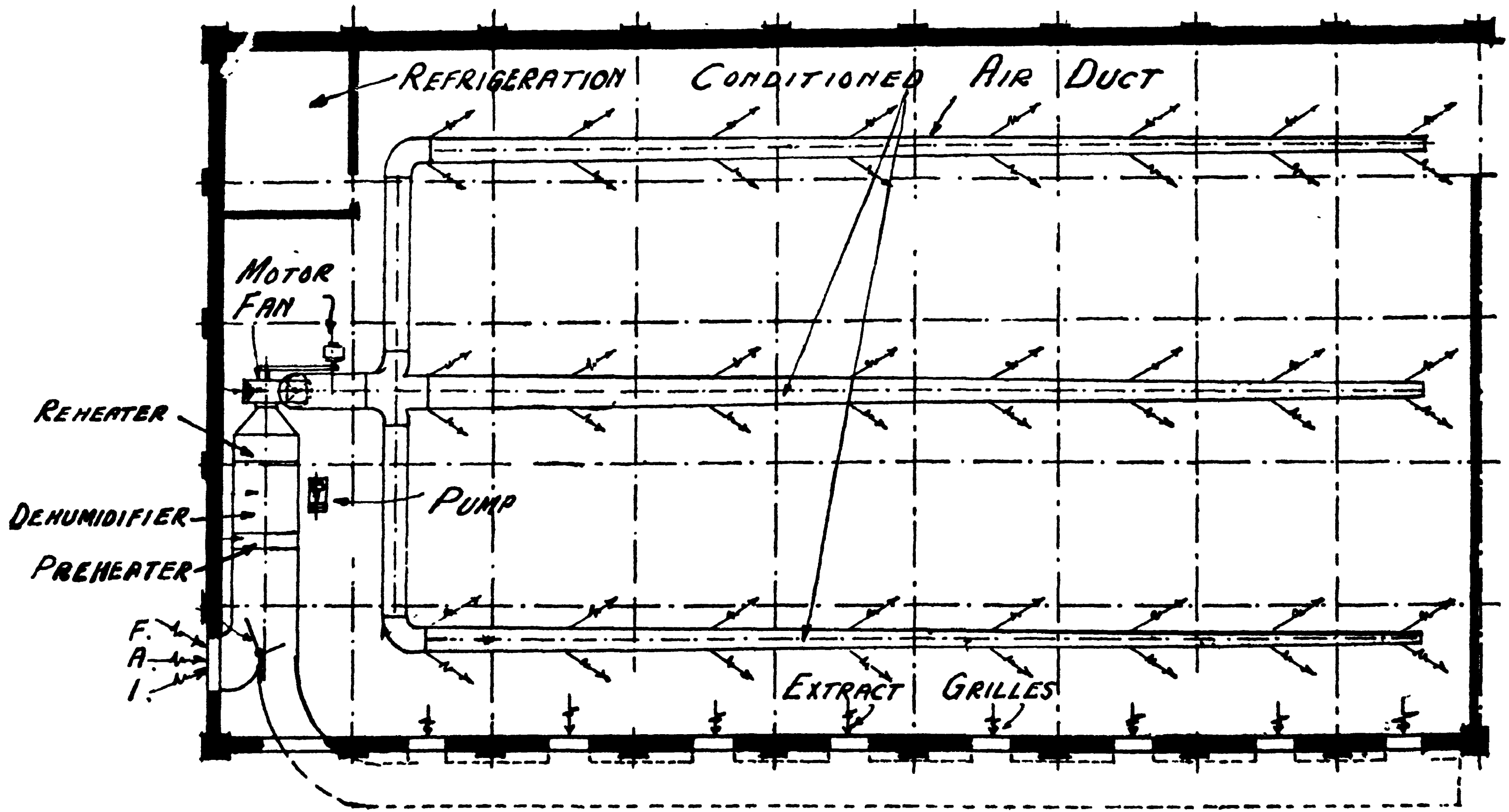


FIG. 60. Factory air-conditioning scheme.

R.H., whilst in the winter the basis for calculation is 32° F. and 50 per cent. R.H.

Firstly we will consider the air-conditioning problem of the summer months. The external basis condition may, from Fig. 59, be seen to correspond to a dew-point of 78° F., a moisture content of 154 grains per lb., and a total heat content of 43 B.T.U. per lb., the wet bulb temperature being 82° F.

The desired internal conditions correspond to 55° F. dew-point, a moisture content of 66 grains per lb., and a total heat of 27·5 B.T.U. per lb. at a wet bulb temperature of 61° F.

It is immediately apparent from these figures that any air introduced to the mill from an external source must have both heat and moisture removed before it is in the correct condition. If the humidity had been of secondary importance it would have been possible to make use of the apparent cooling effect obtained by evaporative cooling, in passing through an air washer, when if the apparatus were correctly designed the air could be assumed to be cooled through 70 per cent. of the wet bulb depression, that is, 70 per cent. of the difference between the wet bulb temperature of the external air and the dry bulb temperature, in our case

$$0{\cdot}7 \times (90-82) = 6° \text{ F. approximately.}$$

Before proceeding any further with this point it should be mentioned that the various sources of heat tending to a rise of temperature in the mill without any air being introduced have been calculated by well-known methods to be as summarized below:

	B.T.U. per hour
Transmission through structure	90,000
Heat due to radiation from sun	70,000
Heat equivalent of motors in mill	1,100,000
Heat from operatives	340,000
Total	1,600,000

The entering air can only be cooled 6° F., that is, to 84° F., which is not low enough to give any useful cooling effect in this case, during extreme conditions. There will, however, be certain times when evaporative cooling could do all that is required.

Air taken into the conditioning plant is cooled by contact with refrigerated spray water or any other direct cooling method to the dew-point of 55° F., and must be supplied in sufficient quantity that in absorbing all the heat in the room it reaches the desired internal dry bulb temperature, the R.H. being thereby adjusted to the correct percentage. There are instances where other factors dictate the quantity of air to be used, but these will not be considered at this stage.

The approximate quantity of air required will, therefore, be

$$\frac{1{,}600{,}000}{(72-55)\times 0\cdot 02} = 4{,}720{,}000 \text{ cu. ft. per hour.}$$

Let us assume that the plant is designed for this volume, which actually represents an air change of approximately 23 times per hour, which is a very high rate of ventilation. It would be possible with such a high volume to obtain the greatest benefit from spray or evaporative cooling, but for the correct conditions to be maintained, the wet bulb temperature must be no higher than approximately 40° F., which is equivalent to almost negligible R.H. at the internal dry bulb temperature of 72° F.

It is now possible to determine the total amount of heat to be removed from the air, and the capacity required for the refrigerating equipment. To ensure accuracy in the calculations it is desirable to keep air quantities in pounds. The total quantity required is

$$\frac{1{,}600{,}000}{(72-55)\times 0\cdot 24} = 392{,}000 \text{ lb. per hour.}$$

It will be assumed that in the interest of economy 75 per cent. of the air is recirculated from the mill, for otherwise the refrigerating equipment would be of such dimensions that the initial cost would be prohibitive.

Any air taken from the mill, as we have seen, has a total heat content of 27·5 B.T.U. per lb., and this before re-use must be cooled to saturation at 55° F., corresponding to a heat content of 23·5 B.T.U. per lb., so that 27·5—23·5 = 4 B.T.U. per lb. must be removed.

Air taken from an external source must be cooled from a total heat of 43 B.T.U. to 23·5, requiring the removal of 43—23·5 = 19·5 B.T.U. per lb.

The total amount of heat to be removed may be found as follows:

75 per cent. recirculated × 392,000 × 4	=	1,535,000 B.T.U. per hr.
25 per cent. fresh air × 392,000 × 19·5	=	1,920,000 B.T.U. per hr.
		3,455,000
Margin for losses		545,000
Total		4,000,000 B.T.U. per hr.

The term 'ton of refrigeration' is employed as a unit of refrigerating effect and is derived by multiplying the latent heat of ice, 144 B.T.U., by the number of lb. in 1 ton, becoming 312,000 B.T.U. This is assumed to be absorbed in 24 hours, so that $\frac{312,000}{24} = 13,000$ B.T.U. per hour is the equivalent of 1 ton of refrigeration.

It should be noted that this term was originated in America where 1 ton = 2,000 lb., so that the U.S.A. ton of refrigeration is 12,000 B.T.U. per hour.

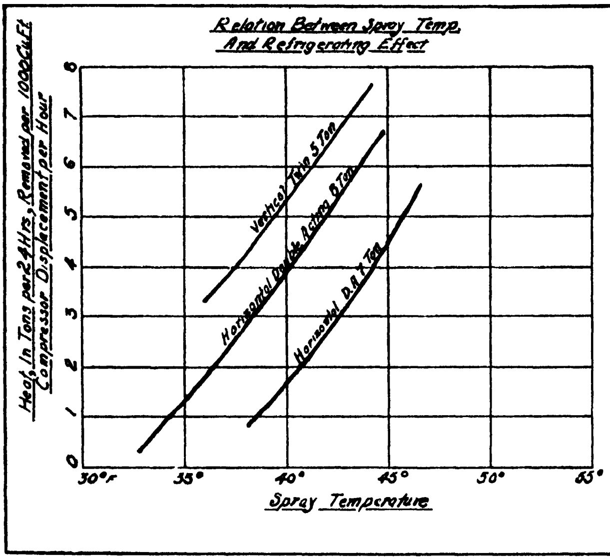


FIG. 61. Relative refrigeration performance.

The refrigeration effect required for the conditions which we have been considering is thus:

$$\frac{4,000,000}{13,000} = 307 \text{ tons.}$$

This figure is not the rated capacity of the plant required. A. Lewis shows† that the actual capacity of the refrigerating plant may be from three to six times the rated capacity, according to the temperature of spray water. The higher efficiency is obtained with the higher temperature of spray water, this temperature being in each case 48° F. and 51° F. respectively.

Fig. 61 shows clearly the relative performance of refrigerating plants at various spray-water, that is, condenser, temperatures. In no case should the spray-water temperature be less than 40° F., as below this point the cost of refrigerating plant becomes excessive.

In this example we will take the spray-water temperature as 45° F. We may then take the capacity of the plant to be 4·5 tons per

† 'Refrigeration applied to air conditioning', *The Heat. and Vent. Mag.* Jan. 1927.

24 hours per 1,000 cubic feet per hour compressor displacement, so that a displacement of 68,000 cu. ft. per hour would be needed, that is, 1,134 cu. ft. per minute, the compressor units being chosen accordingly.

The Baudelot cooler would require a boiling-point of, say, 31° F., which corresponds to a gauge pressure of 46 lb.

The difference of temperature between the ammonia and spray water would average 20° F., so that taking K for the Baudelot coils as 60 B.T.U. per sq. ft. per degree difference, the transmission would be 1,200 B.T.U. per sq. ft.

The cooler surface required for 4,000,000 B.T.U. per hour would be:

$$\frac{4{,}000{,}000}{1{,}200} \text{ or } 3{,}330 \text{ sq. ft. approx.}$$

The Baudelot cooler, compressor, etc., are arranged in separate units adjacent to the air conditioner.

Carrier has shown that for cooling by contact with cooled spray water it is rarely possible to cool the air to a lower temperature than 3° F. above the final spray-water temperature, which means a final temperature of 52° F., so that the spray water rises from 45° to 52° F. the quantity required being:

$$\frac{4{,}000{,}000}{(52-45)} = 571{,}400 \text{ lb. per hr., equivalent to } 970 \text{ gallons per minute.}$$

Alternative Scheme for Ten Air Changes.

As mentioned, there are occasions where air changes are specified and the other factors must be varied to suit. We will consider the same building with a scheme designed for ten air changes per hour.

The volume of the building is 204,000 cu. ft., so that $204{,}000 \times 10 = 2{,}040{,}000$ cu. ft. per hour are required, and taking an average of 12 cu. ft. to 1 lb. we find $\frac{2{,}040{,}000}{12} = 170{,}000$ lb. per hour are required.

This quantity must be introduced at such a temperature that in rising to the room temperature 1,600,000 B.T.U. per hour are absorbed.

The entering air must be lower than the required room temperature by $\frac{1{,}600{,}000}{170{,}000 \times 0{\cdot}24} = 39°$ F., and must enter at $72-39 = 33°$ F.

We will ignore for the moment the fact that it could not be introduced at so low a temperature without causing conditions of draught, but apart from this there is another problem. If the air leaves the conditioner at 33° F. fully saturated it still contains too little moisture, for the room condition is represented by saturation at 55° F. We must therefore add moisture by some means in the necessary quantities, remembering that the evaporation of moisture necessitates heat and that the air must not be increased in temperature by the addition of evaporated moisture.

There are several air-conditioning systems whereby moisture is finely atomized and introduced into a room in that condition. If, therefore, this is done, as the moisture is evaporated the heat necessary for evaporation can only come from the surrounding atmosphere which must then be cooled. It is possible to allow for evaporating as much as 20–25 grains of moisture per lb. of air in this manner.

Taking an average value of 1,000 B.T.U. to be required to evaporate 1 lb. of water, we see that the evaporation of 25 grains per lb. for the example taken would result in absorbing $\dfrac{170{,}000 \times 25 \times 1{,}000}{7{,}000}$ $= 610{,}000$ B.T.U. per hour.

As this amount of heat is absorbed in moisture evaporation the air need not be cooled to so low a temperature. The amount of heat to be absorbed by the entering air is now $1{,}600{,}000 - 610{,}000 = 990{,}000$ B.T.U. per hour, and the air must be lower than the room temperature by $\dfrac{990{,}000}{170{,}000 \times 0{\cdot}24} = 24°$ F., and must enter at $72 - 24 = 48°$ F.

It will be recalled that the air in the room requires to contain 66 grains of moisture per lb., whilst from Fig. 59 we see that at 48° F. it has 50 grains, which with the addition of 25 grains per lb. evaporated in the room gives 75 grains as the total figure, actually in excess of that required.

There must be some combination of entering air temperature and weight of moisture evaporated in the room which exactly gives the necessary condition, a problem which is helped by plotting conditions graphically, as may be seen from Fig. 62.

The cooling effect of various entering air temperatures and moisture evaporations are given and in conjunction with the psychrometric nomograph it is found that with air entering at 45° F. and an evaporation in the room of 21 grains per lb. of air, the correct amount of heat is absorbed.

It is interesting to consider the refrigeration required for this arrangement. We have now to cool, it will be assumed, a similar quantity of air as with the greater air changes, namely 25 per cent. of $392{,}000 = 98{,}000$ lb. per hour, to 50° F., the balance of

$$170{,}000 - 98{,}000 = 82{,}000 \text{ lb. per hour}$$

being recirculated.

The total heat of air at 50° F. fully saturated is found to be 20·5 B.T.U. per lb., so that 43—20·5 = 22·5 B.T.U. per lb. must be removed. The heat to be removed is:

82,000×4	=	328,000	B.T.U.	per hour
98,000×22·5	=	2,200,000	,,	,,
Total		2,528,000	,,	,,

This represents a considerable saving in refrigeration by the use of the combined systems, and may be explained by the fact that a small drop in the temperature of air results in a very large drop in its capacity for absorbing moisture, so that to supply a smaller quantity of air at a slightly lower temperature allows a large amount of the cooling to be done subsequently by evaporation.

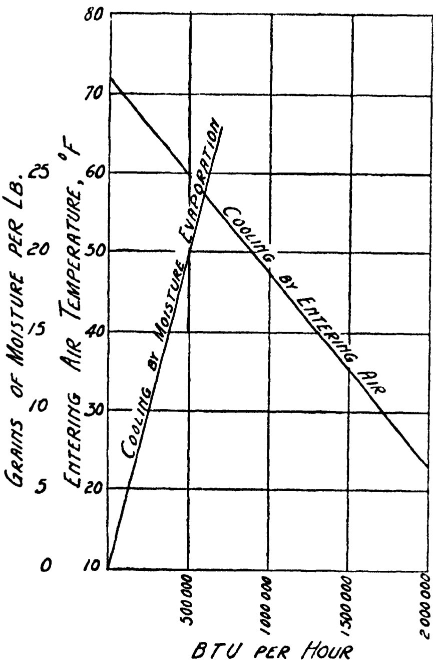


FIG. 62. Evaporative cooling graph.

Conservation of Refrigeration.

In considering the various items which form the total amount of heat to be removed it must be apparent that the whole are not exercising a constant influence. For instance, the heat of the sun is not felt during the early part of the day, and in any building it is generally in the early afternoon that the temperature tends to reach its highest point. In the early morning the external atmosphere, even in the summer months, is considerably cooler than during the later part of the day, so that the heat to be removed from the incoming air is a variable factor.

In industrial buildings, the heat from power units is perhaps the most constant item, but a study of production methods enables the rate at which power is dissipated to be determined.

It is possible therefore to calculate the likely load at any particular part of the day.

With a variable load it is uneconomical to provide refrigerating equipment for any type of building of sufficient power to deal with the maximum load, which may only be reached on a few occasions during the year. With this in view, the author conceived a few years ago a system whereby refrigeration could be conserved during non-operative periods of the air-conditioning plant, bearing in mind that such periods are often several times as long as the periods during which refrigeration is required. The system consists briefly in the storage of large quantities of refrigerated water or brine which is cooled during non-peak load periods, preferably during the local electrical off-load periods in order to benefit by the reduced tariffs then obtainable for electrical energy. The water thus cooled is used during the period when the air-conditioning plant is working, mixed in such proportions as circumstances may dictate with water from the air-washer tank which will have been increased in temperature by contact with the air passing through the spray chamber. By this system it is possible to have a refrigerating machine of comparatively small dimensions, for it has only to deal with an average load.

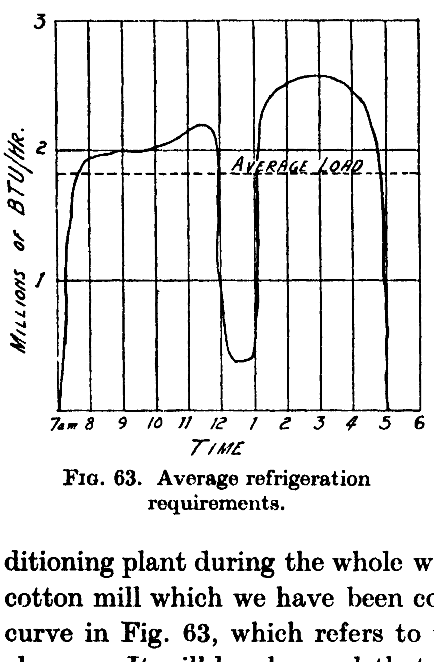


FIG. 63. Average refrigeration requirements.

The first step in the design of such a system is to estimate the load variations of the air-conditioning plant during the whole working period, which for the same cotton mill which we have been considering would be shown by the curve in Fig. 63, which refers to the atomizer scheme with ten air changes. It will be observed that at 7 a.m. the plant is started up and within half an hour all machinery is in operation. From 7 a.m. until 3 p.m. the load is increasing with the heat of the day with a sudden drop at midday, when machinery is not in use and the operatives are away for meals.

The average refrigerating load over a period of 10 hours is about 1,800,000 B.T.U. per hour, that is, a total of 18,000,000 B.T.U. per working day. The remaining 14 hours of the day are available for supplying and conserving this amount of refrigeration, which must be done at the rate of $\frac{18,000,000}{14} = 1,300,000$ B.T.U. per hour, approximately, which is half the maximum load and calls for a refrigeration plant of $\frac{1,300,000}{312,000} = 43$ tons capacity.

There must be some limit in temperature to which the storage medium, either brine or water, may be reduced, or otherwise the efficiency of the refrigerating equipment will become so low that the benefit of reduced requirements will be lost, and to conserve the whole of the day's requirements is often inefficient.

It is therefore in most instances wise to keep the refrigerating plant running throughout the refrigeration requirement period to provide for part of the total load. For the example which we are considering, the refrigerating machine must be of sufficient capacity therefore to provide 18,000,000 B.T.U. partly by direct use and the remainder from storage accumulating during the non-load periods.

Let x = capacity of machine per hour.

$$\text{Then} \qquad \underset{[day]}{10x} + \underset{[night]}{14x} = 18{,}000{,}000$$

$$x = \frac{18{,}000{,}000}{24} = 750{,}000 \text{ B.T.U. per hour.}$$

This is equivalent to a refrigeration rate of 25 tons, and is a comparatively small plant. The arrangement of the plant would now be such that the refrigerator would be run continuously only if external conditions of the atmosphere are at the maximum for which the plant is designed.

During the off-load period of 14 hours the plant would produce $14 \times 750{,}000 = 10{,}500{,}000$ B.T.U., and it may be taken that this could be conserved as water at 35° F., so that in rising to the basic initial spray temperature of 45° F. each lb. of conserved water would give up 10 B.T.U. The storage required is therefore 105,000 gallons.

The cheapest form of construction for a refrigeration accumulator of this capacity is likely to be concrete sunk in the ground with cork insulation combined in the structure. The losses to the earth would be very low and if the cover is properly insulated and the tank can be situated in the shelter of the north walls of the building only small losses would take place.

Summarizing the differences which are possible by conservation of refrigeration, we have reduced the machine from 81 to 25 tons actual capacity with a probable saving in capital expenditure of £2,000, against which must be set the additional cost of conservation equipment, so that at least £1,400 would be saved. Moreover, assuming 25 H.P. to be required by the refrigerating machine, the cost per 24 hours would be perhaps 16 hours at 1*d.* and 8 hours at ¼*d.* per unit, totalling 37*s.* 6*d.*, against say 9 hours at 1*d.* for the larger machine, costing £3 per day. Assuming average conditions throughout the year, this represents an annual saving of £250 which could be capitalized as £5,000.

The Problem of Humidifying.

We have dealt with the cooling and dehumidifying equipment of the mill during the summer months when external temperature and humidity are high, but during the winter months another problem is presented. In extreme conditions the amount of moisture in the external air is insufficient for internal requirements. On the other hand, the amount of heat produced by operatives and machinery is far in excess of that required to keep the temperature at the right level when cold outside.

The heat losses from the building will have been calculated as 800,000 B.T.U. per hour, compared with heat available of 1,440,000 B.T.U. per hour. Reviewing the psychrometric conditions, we see that the external condition of 32° F. and 50 per cent. R.H. from the psychrometric nomograph corresponds to a wet bulb temperature of 29° F., with a heat content of 10·6 B.T.U. per lb., and a dew-point of 19° F. with a moisture content of 13·5 grains per lb.

The entering air must therefore have added to it $66-13{\cdot}5 = 52{\cdot}5$ grains per lb. If this is done by evaporation in the room at the same time, the excess of heat produced above that required to balance heat losses can be absorbed. It is necessary to absorb $1{,}440{,}000-800{,}000 = 640{,}000$ B.T.U. per hour, corresponding to an evaporation of about 640 lb. of moisture or $640\times 7{,}000 = 4{,}480{,}000$ grains per hour.

The amount of air absorbing 52·5 grains per lb. which must be introduced is thus

$$\frac{4{,}480{,}000}{52{\cdot}5} = 85{,}500 \text{ lb. per hour.}$$

The air-conditioning equipment has been designed for summer duty to deliver 170,000 lb. of air per hour, so that we have now the alternative of reducing the amount of air and introducing it to the mill at the internal temperature or supplying the same amount of air and adding the moisture not supplied by evaporation externally by heaters combined with the air-conditioning plant.

For ventilation purposes during the winter a lower air volume would be satisfactory, so that the fan may be reduced in speed to give only 85,500 lb. per hour, that is about 5 air changes per hour, the air being heated from 32° F. to 72° F. before being introduced to the mill. The heat required for raising the air is thus

$$85{,}500\times(72-32)\times 0{\cdot}24 = 820{,}000 \text{ B.T.U. per hour.}$$

With the alternative arrangement, in order to hold the correct amount of moisture the air would need to enter at the internal dew-point of 55° F., and every lb. in rising in temperature to 72° F. must absorb $(72-55)\times0{\cdot}24 = 4{\cdot}1$ B.T.U., so that only $4{\cdot}1\times85{,}500 = 350{,}000$ B.T.U. per hour would be absorbed by the entry of cool air, with a rate of ventilation of 5 air changes per hour, whereas 640,000 B.T.U. needs to be removed. The amount of air at 55° F. required to do this is therefore

$$\frac{640{,}000}{350{,}000}\times 85{,}500 = 156{,}000 \text{ lb. per hour.}$$

The heat to be added to this in raising the temperature from 32° to 55° F. and evaporating moisture is $156{,}000\times(23{\cdot}5-10{\cdot}6) = 2{,}010{,}000$. It is therefore apparent that it is more economical to employ local atomizers in the mill for adding moisture than to use a central plant, owing to the large amount of heat absorbed in moisture evaporation.

Chapter Five

DRYING SYSTEMS

INDUSTRIAL drying systems are employed for drying many materials of diverse natures and each material differs in its requirements in connexion with the drying process. Briefly, drying systems may be classified under the following general groups:

(1) Dryers of all types, where drying takes place without any addition of heat.
(2) Chamber dryers.
(3) Tunnel dryers with automatic conveyor systems.
(4) Rotary dryers, for coal, sand, chemicals, etc.
(5) Vacuum dryers.
(6) Spray drying systems for eggs, milk, soap, etc.

The general principles of design of all drying systems remain similar, and it is only in the detailed application that variations take place, but the various examples which are to follow will illustrate the problems which must be solved, whilst any systems for other materials than those mentioned will be capable of design in a similar manner.

Drying by Air Movement without Addition of Heat.

Drying any material involves the removal of a determined amount of moisture from that material by some convenient means. Moisture must be evaporated from the material and hence some means of conveying away that moisture is essential, so that air must pass over the material in such quantities and in such conditions of temperature and humidity that the precise amount of moisture is carried away. Every material has peculiarities in connexion with moisture evaporation in relation to the temperature, humidity, and rate of air movement which determine the rate at which drying may take place without affecting the material adversely, these matters only being determined by experimental observations. It must not be thought that the presence of heat or high temperature is sufficient to ensure drying taking place, for if the humidity of the surrounding air is high it obviously has but a small capacity for absorbing moisture. On the other hand, air can be too dry for satisfactory drying, resulting in a rapid removal of surface moisture from the material in the early stages so that when subsequently the internal moisture is removed cracking or other effects take place on the surface which was prematurely dried.

Drying may, subject to climatical conditions being suitable, take place without the use of any heat, by the circulation of large volumes of air over the material. In some instances, bricks, timber, and tiles are dried by the air-circulation system with general success. Although we have stated that there is no addition of heat during drying there must still be heat present to ensure evaporation of the moisture. Conversely, if heat is required to evaporate moisture there must be some resultant cooling taking place somewhere in giving up this heat. Actually air at a given dry bulb temperature and R.H. is passing over a given material of which the vapour pressure of its moisture content is greater than that of the surrounding air, resulting in a tendency for moisture to leave the material, to be absorbed by the air until such time as the vapour pressures of the material and air moisture are similar, when no further transfer of moisture will take place.

Therefore, where climatical conditions are generally such that the dry bulb temperature of the air is high and the R.H. low there are far greater possibilities of effecting drying by air circulation than in those parts where cold humid conditions are likely to prevail. To illustrate this remark, referring to the psychrometric nomograph, Fig. 58, we see that air at 100° F. and 40 per cent. R.H. has a maximum moisture content at 100° F. of 300 grains per pound and an actual content of $\frac{40}{100} \times 300 = 120$ grains. If this air is used for drying, the total heat content will remain unchanged during the drying process, but as moisture is absorbed the R.H. will increase to a maximum theoretical value of 100 per cent., the dry bulb temperature at the same time being lowered to a level which will have a similar total heat to that of the initial conditions. The total heat content of the initial condition, from the nomograph, is 40 B.T.U. per lb., and by the procedure explained in Chapter IV the dry bulb and wet bulb temperature for 100 per cent. R.H. would both become 78° F. and the air would contain 145 grains of moisture per lb., so that $145 - 120 = 25$ grains per lb. of air could be removed from the material. If, however, the entering air had been at only 30 per cent. R.H., conditions would be different and by a similar method the air could leave at 100 per cent. R.H. and 75° F. with 130 grains of moisture per lb., the initial moisture being $\frac{30}{100} \times 300 = 90$ grains per lb., and the amount removed per lb. of air being $130 - 90 = 40$ grains of moisture. The relation between the amount of air required in the two cases is thus 40 : 25, that is 1·6 : 1. This, for so small an alteration in initial R.H., is a very large change and illustrates how the drying capacity of a given plant will vary with weather conditions.

Similarly, if the entering air is at 80° F. and 40 per cent. R.H., 20 grains per lb. of air are removed, or if at 80° F. and 30 per cent. R.H. 28 grains per lb.

In the design of air-movement drying systems it is useful to have a graph such as Fig. 64, correlating the dry bulb temperature and R.H. of the air with the quantity of air required to remove 1 lb. of moisture. In practice, the air will rarely leave at a R.H. higher than 80 per cent., but the curves are designed to take account of the variation in the value of the R.H. leaving the dryer which occurs in practice by natural means alone.

It is well known that for every substance the moisture which it contains depends entirely upon the temperature and R.H. of the surrounding air, so that although air at a certain temperature may appear to be capable of carrying away sufficient moisture the hygrometrical condition may be such that no moisture will leave the material and drying is not possible.

We will take as an example an air-movement drying system to deal with the rough drying of a small laundry.

It will be taken that the drying room contains 10 drying horses each 7 ft. long $\times$ 6 ft. high. It is apparent that various classes of goods of widely differing moisture content will need to be dried, and we must therefore determine which is likely to provide the largest amount of moisture.

(a) Blankets. Each horse having load of 4 blankets ($3\frac{1}{2}$ sq. yds.), wet 4 lb., dry $3\frac{3}{8}$ lb.
Moisture to evaporate $= 10 \times 4(4-3{\cdot}375) = 25$ lb.

(b) Sheets. Each horse having 6 sheets ($5\frac{1}{3}$ sq. yds.), wet 3 lb., dry $1\frac{1}{2}$ lb.
Moisture to evaporate $= 10 \times 6(3-1\frac{1}{2}) = 90$ lb.

(c) Flannel shirts. Each horse having 25 shirts (2 sq. yds.), wet $2\frac{1}{8}$ lb., dry $1\frac{1}{2}$ lb.
Moisture to evaporate $= 10 \times 25(2{\cdot}125-1{\cdot}5) = 155$ lb.

The flannel shirt loading is therefore the heaviest loading, and if the drying is to be done in 30 minutes the moisture to be removed is 310 lb. per hour.

A drying system for a laundry must be capable of drying clothes in the winter and summer months, so that a very liberal basis of design must be taken, and therefore 2,000 lb. of air per lb. of water to be evaporated should be supplied, which as may be seen from Fig. 64 would allow drying to take place in very humid conditions.

The total air required is $2,000 \times 310 = 620,000$ lb. per hour, equivalent under average conditions to a volume of 124,000 cu. ft. per minute, which could be provided by two 60-in. propeller fans.

As another example of air-movement drying we will consider a problem as it would be in a warm climate where the lowest temperature is 70° F. with R.H. of 40 per cent. maximum, under which conditions we see that only 430 lb. of air per lb. of water is required. In this case it is desired to dry timber under forced draught conditions, the amount of moisture to be removed being 120 lb. per hour. The volume of air required would be:

$$\frac{430 \times 120 \times 12}{60} = 10,320 \text{ cu. ft. per minute.}$$

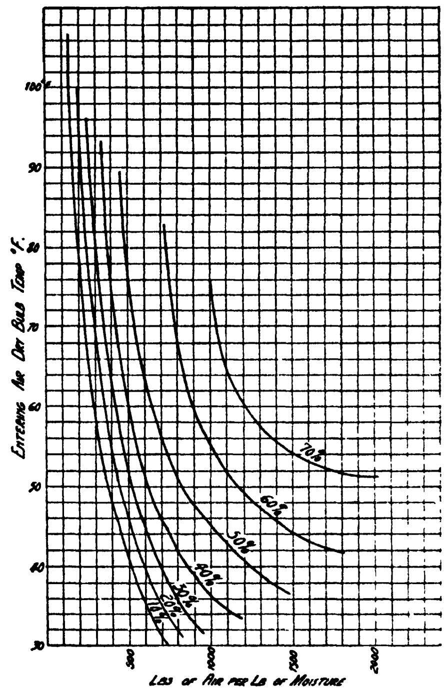


Fig. 64. Air movement drying graph.

Design of Chamber Dryers.

The so-called chamber dryer exists in several forms, either comprising a room in which the material is stacked, with heating pipes or radiators in the room, or with warm air introduced for drying purposes. The general principle of design is the same for either type with obvious exceptions.

We will consider a plant to be designed to dry leather, the number of hides to be dried being 400 each day, the minimum drying period being 50 hours. The weight of a wet hide is to be taken as 8 lb. and in a dried condition 5 lb.

The amount of moisture to be removed in 24 hours is therefore $400(8-5) = 1,200$ lb.

In the case of some materials it is known that a certain maximum temperature of the air must not be exceeded to avoid spoiling the material, and for this problem we will assume that the maximum is to be 90° F. and that the air is to leave at 70 per cent. R.H.

It will also be taken that air is taken from some source at 50° F. and 40 per cent. R.H. before being heated to 90° F. and introduced into the drying chambers.

From the psychrometric nomograph we see that the entering air corresponds to a wet bulb temperature of 41° F. with a total heat of 16 B.T.U. per lb. At 100 per cent. R.H. it could contain 54 grains of moisture per lb., but actually has $\frac{54 \times 40}{100} = 21{\cdot}6$ grains per lb., corresponding to a dew-point of 28° F.

In being heated to 90° F., which we will assume allows 5° for balancing heat losses from the chambers, there is no change in moisture content, but the wet bulb temperature and total heat must increase. Air at 90° F. and 100 per cent. R.H. could contain 220 grains of moisture per lb., so that with only 21·6 its R.H. is

$$\frac{21{\cdot}6 \times 100}{220} = 10 \text{ per cent. approx.}$$

The wet bulb temperature corresponding with this condition is 58° F. and total heat 25 B.T.U. per lb.

At this condition the air is supplied to the drying chambers. It will be taken that it leaves at 70 per cent. R.H., and it must also be recalled that 5° F. cooling is to take place in balancing heat losses. This 5° represents about $0{\cdot}24 \times 5 = 1{\cdot}2$ B.T.U. per lb., so that only $25 - 1{\cdot}2 = 23{\cdot}8$ B.T.U. per lb. is available for the drying process. With 23·8 B.T.U. total heat and 70 per cent. R.H., the temperature is found as 59° F., at which the maximum moisture content is 75 grains per lb. and the actual $\frac{70}{100} \times 75 = 52{\cdot}5$ grains per lb.

The amount of moisture which could be absorbed is thus

$$52{\cdot}5 - 21{\cdot}6 = 30{\cdot}9 \text{ grains per lb. of air.}$$

The moisture to be removed per hour is $\frac{1{,}200}{24} = 50$ lb., so that the amount of air required is $\frac{50 \times 7{,}000}{30{\cdot}9} = 11{,}350$ lb. per hour, or approximately at 90° F., 2,500 cu. ft. per minute.

The amount of heat to be added to the entering air is $25 - 16 = 9$ B.T.U. per lb., or a total of $11{,}350 \times 9 = 102{,}150$ B.T.U. per hour.

The arrangement of drying chambers requires some consideration, for two days is required to dry any particular batch of hides. For this reason it would be considered good practice to have three drying chambers, so that each day one could be unloaded whilst another was filled with wet hides.

Design of Brick-drying Plants.

There are many methods in use for the drying of all descriptions of bricks, tiles, and earthenware products. The ordinary building brick exists in so many types formed of clays of various natures that there are many plants designed for drying them. In a modern brick-field drying is an important stage of manufacture because faulty drying results in the bricks being scummed, that is, having a deposit due to steam from the kiln chambers in which the bricks are being burnt condensing upon bricks in adjoining chambers which have not advanced so far in the burning, whilst the cracking of bricks in the kiln is also in many cases related to the drying process. The scumming of bricks in the kiln can generally be stopped by reducing the moisture content before the bricks enter the kiln by passing them through a properly designed tunnel dryer, in which they are reduced to the minimum possible moisture content.

The time required for drying various classes of clay varies over a wide range, but generally speaking if a brick contains 1 lb. of water approximately 35 hours is required, but if, on the other hand, the same brick is made of a greasy clay as much as 60 to 70 hours is necessary. The drying of bricks follows the general principles of most drying systems in that the material should firstly be allowed to remain for a period in a warm humid atmosphere so that the whole mass can be properly warmed and the risk of surface drying be prevented.

For brick-drying systems the necessary heat for the plant can be obtained either by the use of live or exhaust steam or waste heat from the kilns, and it is the latter arrangement that generally proves to give the most efficient plant and reduces the cost of drying.

In other cases a very cheap and efficient arrangement consists of the use of a direct coke furnace, the smoke and hot air being drawn over this furnace to be discharged into the drying tunnels. This particular system has the advantage of giving very rapid and convenient means of adjusting the temperature of the air in the tunnels, as a fresh-air inlet can be arranged on the suction side of the fan which draws air from the furnace so that a mixture of furnace and fresh air may be delivered to the tunnels.

The drying tunnels for bricks, like other tunnel dryers, vary between 70 and 100 ft. in length, the height being about 5 ft. 6 in. and the width 3 ft. 6 in.; generally several tunnels are constructed side by side with light division-walls.

We will consider the design of a tunnel drying system for bricks to dry 30,000 bricks per week at the rate of 5,000 per day, each to have 3 lb. of moisture removed, the time required being 72 hours.

It will be taken that air will be introduced into the tunnels at 150° F. and will leave with 80 per cent. R.H.

The amount of moisture to be removed will be found as

$$\frac{5,000\times 3}{24} = 624 \text{ lb. per hour.}$$

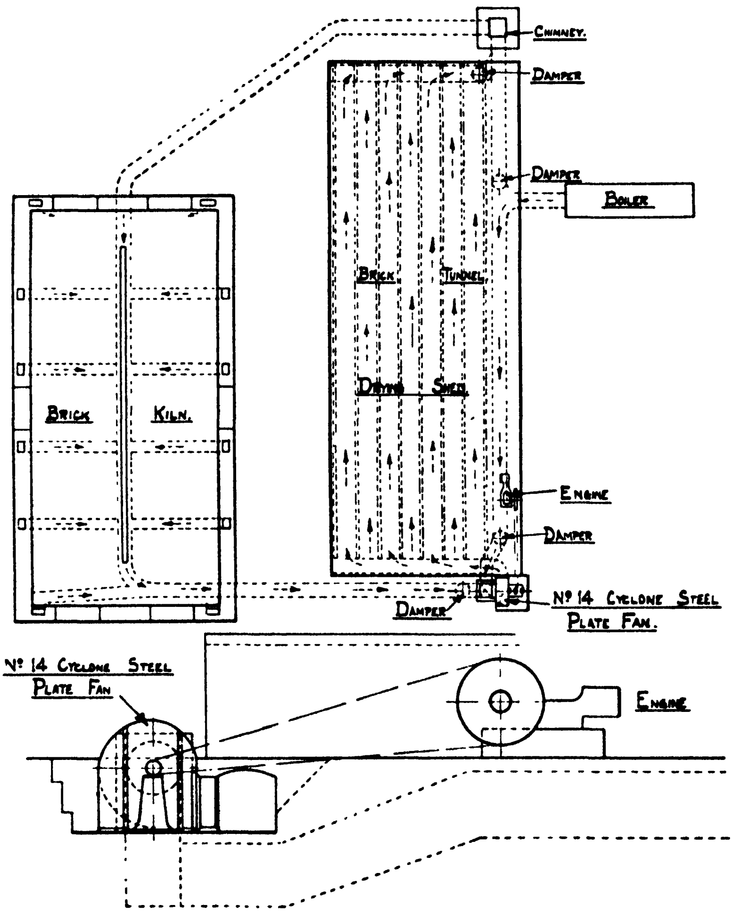


Fig. 65. Brick-drying system.

By a similar procedure to before, air at 200° F. having been raised from 50° F. and 40 per cent. R.H. will have a maximum moisture content at 200° F. of 1,300 grains; at 50° F. and 40 per cent. R.H. it would have 21·6, so that the R.H. entering the tunnels would be $\frac{21\cdot 6\times 100}{1,300} = 2$ per cent. and the air would leave at 86° F. with 80 per cent. R.H., actually having a moisture content of 152 grains per lb.

The amount of air required is therefore $\frac{624 \times 7{,}000}{152-21 \cdot 6} = 33{,}700$ lb. per hour, which at 150° F. is equivalent to 8,750 cu. ft. per minute.

The arrangement of a typical tunnel drying system is shown in Fig. 65.

It will be assumed that in this instance the heat necessary for drying will be obtained from a coke furnace. It may be taken that with this arrangement 90 per cent. of the calorific value of the coke is available as heat, or approximately 10,000 B.T.U. per lb.

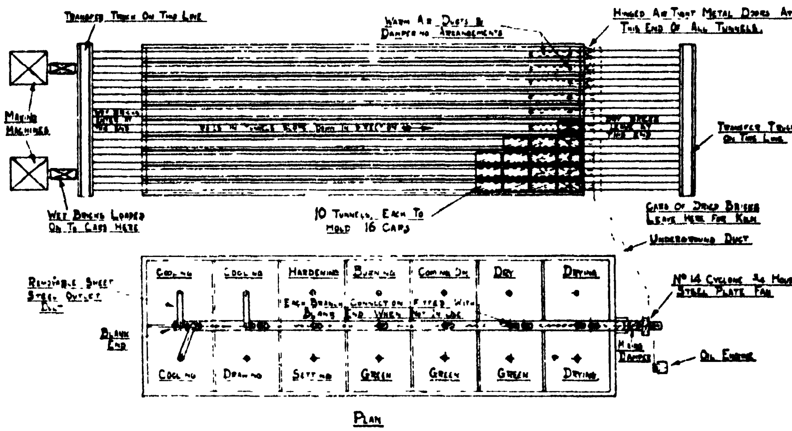


FIG. 66. Waste heat brick-drying system.

The total heat of the entering air is 16 B.T.U. per lb., and we find that on heating to 150° F. it has a total heat of 41 B.T.U. per lb., so that $41-16 = 25$ B.T.U. per lb. must be added, and a total amount of $33{,}700 \times 25 = 842{,}500$ B.T.U. per hour is required.

The amount of coke to be burnt is therefore $\frac{842{,}500}{10{,}000} = 84$ lb. per hour, and as up to 7 lb. of fuel per hour may be burnt per sq. ft. of grate surface, the grate area required for the furnace would be $\frac{84}{7} = 12$ approximately, or 4 ft $\times$ 3 ft. dimensions.

The Use of Kiln Heat for Drying.

As has been stated previously, it is possible to employ the residual heat of the bricks in the kiln for heating purposes, this being extracted through a ducting system from the top of the kiln as illustrated in Fig. 66, and thence introduced to the drying tunnels. When the bricks do not contain more than 1 lb. of moisture each on entering the drying tunnel the heat of the bricks in the black state in the kiln is sufficient to dry the wet bricks. The bricks are usually fired in the kilns at a temperature of 1,600°–2,000° F. for 24 hours, and afterwards allowed to cool to 600°–1,000° F. for a further period of 24 hours, so that the amount of heat available may readily be calculated; assuming that the bricks weigh approximately 5 lb. each and have a specific heat of 0·20, for in cooling 1,500° F. the heat given up is $1{,}500 \times 5 \times 0{\cdot}2 = 1{,}500$ B.T.U., which is above that required to evaporate 1 lb. of moisture.

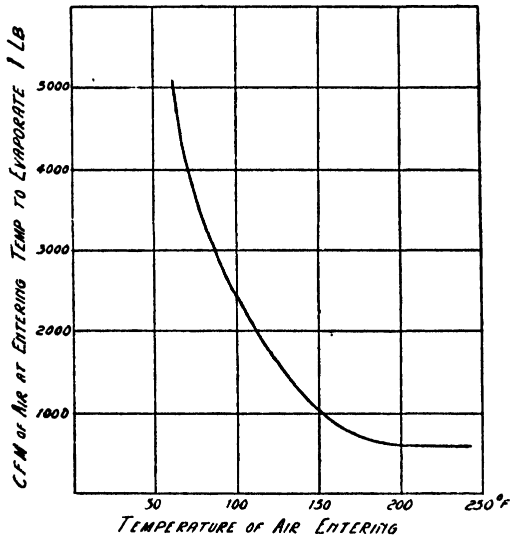


Fig. 67. Air required for drying.

Simplification of Drying Problems.

In this country the basic external conditions of the atmosphere upon which design should be based is about 50° F. and 40 per cent. R.H., and as 80 per cent. R.H. is a constant figure for the humidity of air leaving the dryer, it is possible to calculate the approximate amount of air in cu. ft. required per lb. of moisture to be evaporated, allowing for well-constructed and insulated drying chambers or tunnels. The curve in Fig. 67 gives data for this purpose. As the temperatures for various materials are governed by the nature of the material, it follows that each type of material will conform to a definite volume of air per lb. of water evaporated, whilst for a chamber or tunnel dryer where all the heat is added externally the temperature of the air leaving the dryer is also constant. The following table gives data for many materials dried in the tunnel or chamber dryer.

Material	Entering air temperature	Volume of air required per lb. of evaporated water in cu. ft.	Approximate leaving air temperature
	° F.		° F.
Artificial silk	110–130	1,500–2,000	72–4
Bedding	150–190	600–1,000	79–87
Blankets and flannels	120	1,750	73
Blacklead	200	600	93
Bricks	90–200	600–2,850	63–93
Clothes	100–180	700–2,400	70–85
Cardboard	140	1,250	77
Glue	70–90	2,800–4,000	54–63
Leather	85–100	2,400–3,100	62–73
Timber	70–150	1,050–4,000	54–79
Paper	150–300	400–1,050	79–130
Pottery	120	1,700	73
Rubber	80–90	2,850–3,300	59–63
Textiles	140–180	700–1,250	77–85
Yarns	120–160	850–1,700	73–81

If therefore it is desired to dry pottery goods containing 25 per cent. moisture to the extent of 120,000 lb. per week, the drying period necessary being 30 hours and the total period for which the plant is in operation 144 hours, we may design as follows:

$$\text{Total moisture} = \frac{120{,}000 \times 25}{100} = 30{,}000 \text{ lb.}$$

$$\text{Moisture removed} = \frac{30{,}000}{144 \times 60} = 3{\cdot}47 \text{ lb. per minute.}$$

From the table we see that at 120° F. pottery goods require 1,700 cu. ft. per lb. of moisture evaporated, so that

$$3{\cdot}47 \times 1{,}700 = 5{,}900 \text{ cu. ft. per minute}$$

are required for the plant in question.

The air would leave the drying room at an approximate temperature of 73° F.

Drying Systems with Intermediate Heating.

The use of drying systems with intermediate heating, in which heat is added in the chamber or tunnel in addition to that contained in the air which is introduced, is gaining in popularity owing to the increased temperature at which it is then possible to have the air leaving the dryer, and consequently the decreased amount of air which need be introduced to carry away a definite quantity of moisture from the material. When it is considered that air entering at 150° F. into a drying chamber usually cools to about 79° F., it will be seen that there is not much power for absorbing moisture although the initial temperature of the air is high. If, however, the air could leave at

120° F. it would be able to absorb four times as much moisture and one-quarter of the amount of air would be required for the same duty.

The additional or intermediate heating may be effected by the use of radiators or coils in the drying room, or by reheating the air in batteries, as is done in the multi-cell drying plant.

The calculations of the drying system vary slightly when intermediate heating is employed, and in these cases it is usual, depending largely upon the material handled of course, to introduce the air at a temperature equal to that at which it leaves the tunnel. If, therefore, 120° F. is chosen as the leaving temperature and 80 per cent. R.H., the air could contain 464 grains of moisture per lb. The amount of moisture which could be carried away per lb. of air would be $464 - 21{\cdot}6 = 442{\cdot}4$ grains. The entering air at 50° F. and 40 per cent. R.H. is raised to 120° F., at which temperature the total heat is found to be 32 B.T.U. per lb. At 120° F. and 80 per cent. R.H. the total heat is 85 B.T.U. per lb., so that $85 - 32 = 53$ B.T.U. per lb. can be added by intermediate heating.

So that, a plant which must evaporate from the material 300 lb. of moisture per hour must now have $\frac{300 \times 7{,}000}{442{\cdot}4} = 4{,}750$ lb. of air per hour, which at 120° F. is approximately 1,250 cu. ft. per minute.

Heat wasted in Air leaving Dryer.

Having decreased the amount of air required for drying we have at the same time greatly increased the moisture content and total heat at the leaving condition, and this is normally wasted. If the air before being discharged is passed through a thermo-recuperator consisting of a heat transfer unit in which the humid air is reduced to a low temperature, and the moisture condensed out, a large proportion of the heat otherwise wasted will be conserved.

For instance, with the dryer which we have considered, the air leaving at 120° F. and 80 per cent. R.H. has 85 B.T.U. per lb. total heat. The incoming air has to be raised from a heat content of 16 to 32 B.T.U. per lb., in having the dry bulb temperature increased from 50° to 120° F. The problem now is to determine how far the leaving air may be cooled and the incoming air thereby raised in temperature.

If the heat given up by the air leaving the dryer is determined for various saturation temperatures after passing through the thermo-recuperator, these figures may be plotted to give a straight line as in Fig. 68. Similarly, the gain in total heat of entering air at various increased temperatures may be plotted on the same diagram.

It will be obvious that there must be a difference of temperature for heat transfer to take place, and the smaller this difference the more heat is taken up by the incoming air. The intersection of the heat gain and loss lines gives the point at which both entering and leaving temperatures are similar, and the change of heat content for both is also the same.

Generally it is not practicable to have less than 15° F. difference between entering and leaving air, and by plotting a dotted horizontal line of length representing 15° the two temperatures are found

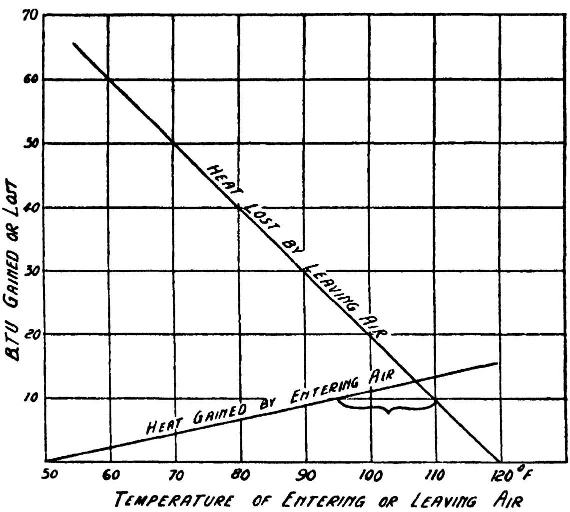


FIG. 68. Thermo-recuperator graph.

as 110° F. for leaving air and 95° F. for entering air, a change of heat content of 10 B.T.U. per lb. taking place. In a simple tunnel dryer of the type mentioned the saving by the use of a thermo-recuperator does not seem great, but in the multi-cell dryer where progressive increase in temperature takes place, a larger drop in temperature of leaving air is possible and the preliminary stages of drying then take place without added heat. A typical arrangement of a multi-cell dryer with thermo-recuperator is shown in Fig. 69.

Calculations for Multi-cell Dryer.

Considering now the dryer shown in Fig. 69, let it be taken that the plant is required to evaporate 800 lb. per hour of moisture from artificial silk hanks which are carried on poles by a chain conveyor through the drying tunnel.

With this drying system air enters the tunnel at both ends and increases in temperature as it passes through the successive stages to the centre of the tunnel from whence the air is exhausted to be

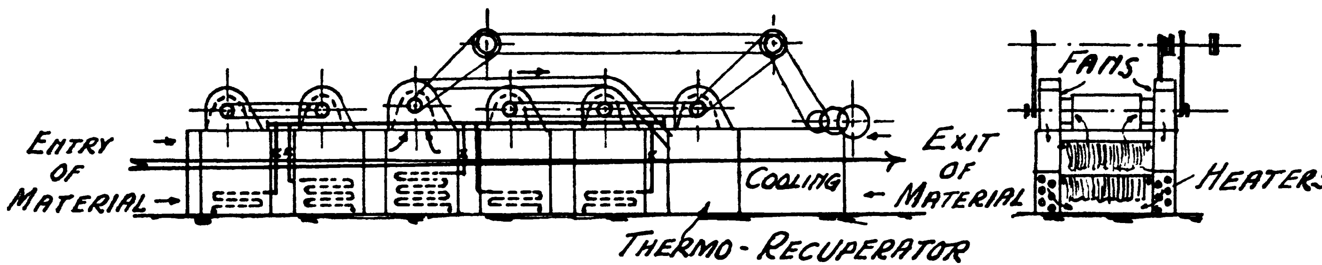


Fig. 69. Multi-cell dryer.

delivered to the thermo-recuperator in one of the final stages. It will be taken that the maximum temperature in stage 3 is 190° F. with 80 per cent. R.H., whilst the amount of moisture to be evaporated in each stage will be assumed as follows:

Stage . . .	1	2	3	4	5	6
Moisture (lb.) . .	75	200	300	100	75	50

The amount of air passing through the various stages will with the exception of stage 3 be the same. Stage 3 will have double the amount of any other stage.

In stage 3 the air will be introduced at its maximum temperature. Entering the tunnel at 50° F., and 40 per cent. R.H., it is heated to a final leaving temperature of 140° F. and 80 per cent. R.H. with a moisture content of about 860 grains per lb., and as it has 21·6 grains per lb. in the entering condition can remove 860—21·6 = 838·4 grains per lb., the total amount of air required being

$$\frac{800 \times 7{,}000}{838\cdot 4} = 6{,}700 \text{ lb. per hour.}$$

Of this quantity 3,350 lb. per hour will be introduced at each end of the tunnel, but each of the circulating fans will circulate 6,700 lb. per hour.

Considering now the first stage, air is taken in at 50° F. and 40 per cent. R.H. and is heated and delivered into the tunnel. In this stage the 3,350 lb. per hour must pick up $\frac{75 \times 7{,}000}{3{,}350} = 157$ grains per lb., giving a total of 179 approximately, represented by a dew-point temperature of 83° F., but it will be taken that it only leaves at 50 per cent. saturation and 104° F.

In the second stage the air must absorb $\frac{200 \times 7{,}000}{3{,}350} = 418$ grains per lb., the total grains becoming 597 representing 120° F. dew-point, and with 65 per cent. R.H. a leaving temperature of 138° F.

In the third stage, the amount of air is 6,700 lb. per hour and the moisture to be absorbed is $\frac{300 \times 7{,}000}{6{,}700} = 313$ grains per lb.

Considering now stage 6, the moisture to be added is

$$\frac{50 \times 7{,}000}{3{,}350} = 104 \text{ grains per lb.}$$

or a total of 126 representing 74° F. saturation temperature, or at 50 per cent. R.H., 90° F.

In stage 5 the moisture added is $\frac{75 \times 7{,}000}{3{,}350} = 157$ grains per lb., giving the total as 283, or a saturation temperature of 94° F. which for 65 per cent. R.H. is a leaving temperature of 109° F. In stage 4 the moisture added is $\frac{100 \times 7,000}{3,350} = 209$ grains per lb., making a total of 492, for a saturation temperature of 100° F. which at 80 per cent. R.H. is a leaving temperature of 120° F.

The moisture absorbed in stages 1 and 2

$$= \frac{(597-22) \times 3,350}{7,000} = 275 \text{ lb.}$$

That in stages 4, 5, and 6 $= \frac{(492-22) \times 3,350}{7,000} = 225$,,

Stage 3 evaporates the remainder $= 300$,,

800 lb.

These figures give a check on the total amount of moisture evaporated. We will now determine the amount of heat to be added in the various stages:

Stage 1. The total heat at 104° F. and 50 per cent. R.H. is 49 B.T.U. per lb., so that the heat added is $(49-16) \times 3,350$ = 110,000 B.T.U. per hour.

Stage 2. The total heat at 138° F. and 65 per cent. R.H. is 101 B.T.U. per lb., so that the heat added is $(101-49) \times 3,350$ = 174,000 ,, ,,

Stage 6. The total heat at 90° F. and 50 per cent. R.H. is 38 B.T.U. per lb., so that the heat added is $(38-16) \times 3,350$ = 74,000 ,, ,,

Stage 5. The total heat at 94° F. and 65 per cent. R.H. is 45 B.T.U. per lb., so that the heat added is $(45-38) \times 3,350$ = 23,450 ,, ,,

Stage 4. The total heat at 120° F. and 80 per cent. R.H. is 85 B.T.U. per lb., so that the added heat is $(85-45) \times 3,350$ = 134,000 ,, ,,

Stage 3. The total heat at 140° F. and 80 per cent. R.H. is 150 B.T.U. per lb., so that the added heat is $(150-85) \times 3,350$ = 217,000 ,, ,,
and $(150-101) \times 3,350$ = 164,000 ,, ,,.

Total heat = 896,450 B.T.U. per hour.

These figures are of course theoretical, and in practice the amount of heat required, taking into account that required to raise the temperature of material and water, and that lost by conduction through the walls of the drying tunnel, is likely to be about double the theoretical figures. These items may be calculated approximately as follows:

To raise 800 lb. of water per hour from 50° to 140° F. requires $(140-50)800 = 72{,}000$ B.T.U. per hour.

The loss from walls, assuming the tunnel to be 60 ft. long × 8 ft. wide × 7 ft. high, having an exposed area of 1,800 sq. ft., is, taking an average temperature of 120° F. and a coefficient of 0·5 B.T.U. per sq. ft., $1{,}800 \times 0{\cdot}5(120-50) = 63{,}000$ B.T.U. per hour.

Other heat is of course required to heat up the structure, but for continuous operation this need not be calculated.

The material must be raised, however, from 50 to 90° F., and assuming the water to be 30 per cent. of the total weight and the specific heat of the material to be 0·50, we have here

$$800 \times \frac{100}{30}(90-50) \times 0{\cdot}5 = 53{,}500 \text{ B.T.U.}$$

which will probably be required during the first hour. The summary of heat requirements now becomes:

	B.T.U. per hour.
Calculated requirements	896,400
Heat for raising water	72,000
Loss from walls	63,000
Heating up material	53,500
Heating trucks, etc. (assumed)	40,100
Total	1,125,000

It must be remarked in passing that practical working observations show that the amount of heat required to evaporate 1 lb. of moisture is rarely less than 1,800–2,200 B.T.U., whereas the calculated figures show $\frac{1{,}125{,}000}{800} = 1{,}530$ B.T.U. per lb. It is therefore recommended that an addition of 25 per cent. should be made to calculated figures in deciding upon the dimensions of all air heaters.

Air Recirculation in Drying Systems.

There are many materials which require a high humidity of the air required for drying in the initial stages of the process, and it is therefore customary to arrange them for recirculating the air from the end at which it leaves the dryer and to deliver it back to the entering end (where air and material move in the same direction or in cabinet dryers). With a contra-flow tunnel dryer with air and material moving in opposite directions this is automatically accomplished, for the humid leaving air meets the wet incoming material, and it is not until the material is reaching the final stages of drying that it comes into contact with the hottest and driest air. Even so, some dryers are worked for the first few hours with a closed recirculating system. The proportion of recirculated to fresh air is gradually reduced as the drying proceeds and the material is better able, having been heated to the innermost parts, to be amenable to rapid drying.

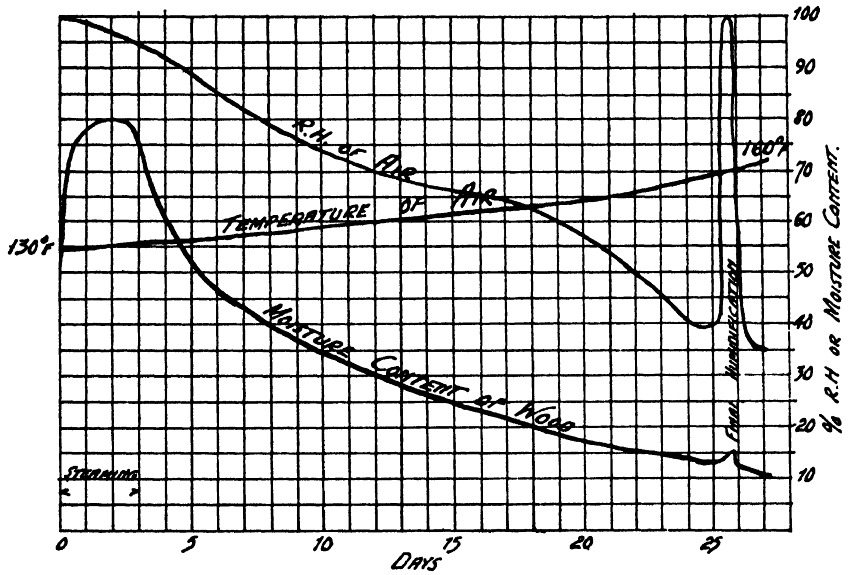


FIG. 70. Ideal drying curves.

Indeed with timber, steaming is resorted to for several days before actual drying takes place. A. Ihne shows† an interesting graph illustrating the variations in temperature and humidity during the stages of drying 2-in. oak boards.

Fig. 70 gives ideal curves based on his test data for a complete drying process, which could, by means of temperature and humidity controls operated by clock mechanisms, be closely adhered to in practice.

Drying of Food Crops, etc.

After passing successively through the processes of withering, rolling, and fermentation, tea is dried. The original Chinese method of drying was over a chulas or charcoal fire, but this method has been supplanted by the use of mechanical dryers, which are far more efficient.

† *Le Séchage des bois*, by A. Ihne.

Since the introduction of the first mechanical dryer considerable progress has been made in this both as regards its capacity and mechanical design. The original 'Sirocco' machine was termed an up-draft dryer and was self-acting, that is to say, the drying chamber, containing a number of trays for holding the leaf, was placed immediately above the direct-fired air heater, the hot air from the latter passing by convection upwards through the drying chamber and through the trays of leaf contained therein. The first dryer of this type had only four trays, and its capacity was consequently small, but larger machines with more trays of greater superficial area were

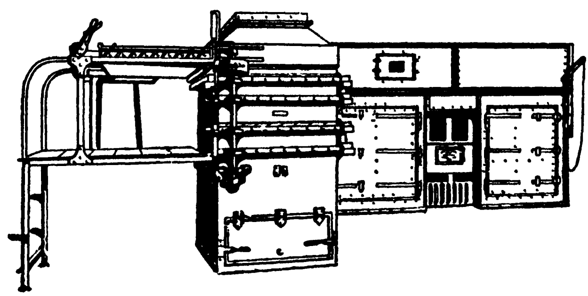


FIG. 71. Down-draft dryer.

evolved, becoming very popular among the planters. It will be observed that these up-draft dryers do not require power for their operation, a necessity in those early days of tea planting when an engine in the factory was the exception rather than the rule.

This type of dryer is now made with 16 and 20 trays, which, with working temperatures of 200°–220° F., have evaporative capacities of 90–100 lb. and 110–130 lb. of water per hour respectively. The corresponding outputs of dried tea depend upon the extent of the wither, the quality of the leaf, and upon the climatic conditions.

The next development in tea-drying machinery was represented by the down-draft Sirocco tea dryer, shown in Fig. 71, in which mechanical power was required for the operation of the fan that draws the heated air from the stove into the drying chamber. By the aid of a fan a larger volume of hot air could be passed through the trays of leaf, with the result that the capacity of these dryers was, generally speaking, larger than that of the up-drafts. They, like the up-draft machines, are still used considerably for the final firing.

The method of working the machines is as follows: the handle, which is placed at the side of the drying chamber and controls both the fan valve and the tray levers, is raised, and each tray of wet roll pushed into the chambers through the bottom tray port. The handle is then pressed down, which causes the tray to move up the first set of pawls, where it is retained. Simultaneously the fan valve opens, which establishes a strong draught of hot air through the trays, and the bottom tray port opening is closed so as to prevent an in-flow of cold air. Each tray in its passage through the drying chamber reaches the intermediate port, where it is withdrawn, the tea turned, and the tray reinserted. The tray then passes up to the top port, where it is finally removed, the drying operation being completed. This tray is then respread with fresh wet roll and inserted at the bottom port as before.

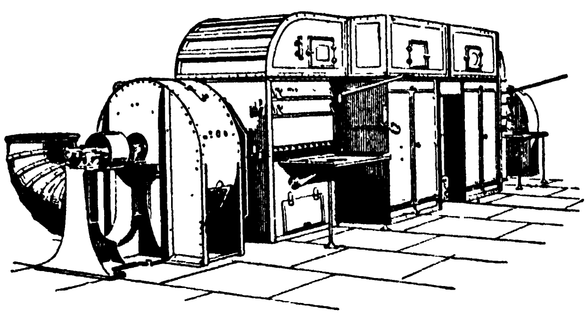


FIG. 72. Enclosed tilting-tray dryer.

The down-draft dryers working at a temperature of 200°–220° F., and with good quality fuel, have the following evaporative capacities:

Large down-draft with vertical flue heater, 180–220 lb. per hour.

Large down-draft with multi-tubular heater, 180–220 lb. pei hour.

Double down-draft with multi-tubular heater, 260–300 lb. per hour.

The Sirocco enclosed type tilting-tray pressure dryer shown in Fig. 72 may be termed a semi-automatic type of dryer. This dryer is fitted with trays having a tilting movement whereby the leaf, during the process of drying, is transferred, and at the same time effectively turned over, from one line of trays to the next lower, and the finished tea finally discharged from the bottom line of trays into a relatively large collecting chamber in the base of the dryer. The hot air is driven under pressure up through the trays and the leaf thereon in the same way as the 'endless chain pressure dryer', to be described later. The drying is thus finished at the highest temperature, so that even if the tea in the collecting chamber still contains a small percentage of moisture, the heat retained by it will be sufficient to complete the evaporation.

The drying chamber is fitted with five superimposed lines of trays. The first line, or feed drawer, is fitted in a frame which slides into and out of the drying chamber. Briefly, the operation of the machines is as follows: the attendant pulls out the feed drawer, and having spread a charge of leaf thereon, pushes it back into the drying chamber. He then turns the crank handle, which tilts the four lower rows of trays in succession, leaving the row immediately below the feed drawer empty. By releasing the control lever on the feed drawer the trays of the latter are tilted, and the leaf dropped on the row below. On returning the lever to its original position the trays are brought back to the horizontal, and the drawer can be pulled out and recharged. A special device is provided which prevents the trays of the feed drawer being dropped until the cycle of tilting of the four lower rows is completed, thus leaving the row of trays immediately below the feed drawer ready for a further charge of leaf. Any leaf that falls through the trays of the feed drawer during spreading is caught by a canvas screen, which is automatically extended below the trays when the drawer is pulled out. On pushing the drawer into the drying chamber the screen rolls up and delivers into a hopper any leaf that has fallen through.

The four lower rows of trays are operated by an arrangement of gear wheels and cams which, by the turning of a crank handle, causes each line of trays to tilt in successive order from horizontal to vertical position, beginning with the lowest line, the contents of which are discharged into the collecting chamber in the base of the dryer. The latter acts, at the same time, as air-pressure chamber. A further movement of the crank handle raises these trays and retains them in their normal horizontal position during the remainder of one complete cycle of the tilting of the trays.

The row of trays immediately above the lowest line is then tilted and discharges the leaf on the lowest line of trays. The further turning of the handle brings these trays back to the horizontal, whereupon the contents of the third line are similarly deposited on the second, and so on to the fourth line, which is thus left empty and ready to receive a fresh layer of leaf from the feed drawer.

It is, however, important that the leaf, in dropping from one row of trays to another, should do so in still air, otherwise the strength of the air blast would, to a certain extent, hinder the leaf from falling, and cause it to jam the trays. Accordingly, a fifth cam is provided on the camshaft which, immediately a fresh cycle of tilting starts, releases the trigger of the air-supply valve, which closes slightly in advance of the trigger release of the lowest row of trays. The air valve then remains closed until the fourth row of trays has tilted and is coming up again to the horizontal, whereupon it reopens and the drying again starts. All the gear turning the trays is readily accessible, whilst the complete cycle of tilting occupies only from 6 to 8 seconds.

This dryer, which is slightly larger in capacity than the earlier machines, is particularly suitable for gardens, the output of which is not sufficiently large to warrant the installation of an automatic dryer.

The Sirocco endless-chain pressure dryer shown in Fig. 73 is not only one of the most modern but also one of the largest tea dryers. It is entirely automatic in operation, and has an automatic feeder and spreader. The coolies empty the wet leaf into the feed hopper

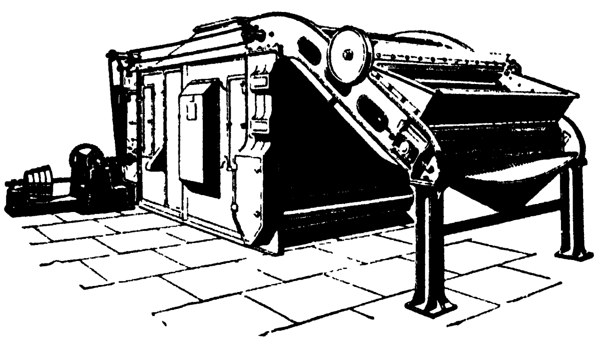


FIG. 73. Sirocco endless-chain dryer.

of the feeder and spreader, the hopper being at a convenient height for this purpose.

The automatic feeder and spreader consists of two cast-iron side plates attached at the upper ends to the cheek plates at the feed end of the drying chamber, and resting at the lower end on two pillars. Between the side plates is fitted an endless band of trays, which are blank and not perforated, carried by endless chains travelling along slides cast on the inner surface of the side plate. The tray chain pass over two sets of sprockets, each set being mounted on a through shaft. The upper set is driven by a chain from a sprocket operated from the driving gear of the drying chamber. The lower set is mounted on a shaft which runs in adjustable bearings, so that the correct tension can be kept on the tray chains.

The spreading plate, which is placed near the hopper, has two serrated edges. It rotates at a fairly low speed and is mounted on a spindle driven by a pulley from the dryer countershaft. By means of a handle at one side of the machine, operating a single worm and worm-wheel underneath each bearing of the spreader plate spindle, the distance between the edges of the spreader plate and the trays, and consequently the depth of the spread, can be instantly adjusted while the spreader is either standing or in operation. Not only is the depth of the material regulated by the spreading plate, but it is also kept uniform over the whole surface of the trays. As long as there is an ample supply of wet leaf in the hopper no attention need be paid to its distribution, as this is perfectly accomplished by the action of the spreading plate. On reaching the upper end of the automatic feeder and spreader the leaf falls lightly and uniformly on the top row of trays of the dryer, which convey it into the drying chamber. A tendency of the wet and sticky leaf to form a more or less solid mass is entirely overcome, firstly by the rotary action of the spreading plate, and subsequently by the leaf falling from the spreader on to the trays of the drying chamber. This ensures the leaf being shaken and opened up, so that all parts are exposed to the heated air, thereby causing uniform drying throughout.

The tilting action of the trays at the upper end of the spreader where the leaf is delivered into the dryer is such that any leaf adhering to them is shaken off, precluding any damp material being carried round. The returning trays, however, are accessible for inspection. To obviate any possibility of droppings from the spreader trays falling among the dried tea as it is discharged from the dryer, an undershield is fitted on the underside of the spreading trays, terminating in a hopper and shoot located between the spreader pillars. The trays of the automatic feeder do not enter the drying chamber proper, and are consequently not exposed to the hot air, and a tendency of the leaf to stew is thus eliminated. The automatic feeder forms an arch over the pit in which the dried tea is collected, but does not interfere with the latter, owing to the distance of the two pillars from the drying chamber, so that there is ample room for removing the dried tea and to give access to the inspection doors at the feed end of the latter. The drying chamber is fitted with three endless bands of trays, which travel from one end of the machine to the other, forming the rows of drying surface. The trays consist of narrow perforated metal sheets driven on endless travelling chains. The leaf is deposited by the automatic feeder and spreader upon the top row of trays and travels to the further end of the drying chamber, where it is discharged on to the lower row of trays which are travelling in the opposite direction, that is to say, towards the front of the machine. On reaching the end of the second row the leaf falls on to the third row, and so forth until the sixth and last row delivers it into the receiving hopper, from which it is mechanically discharged.

In order to check the fermentation of the leaf after it enters the dryer, a hot air by-pass is fitted to each side of the drying chamber, so that a small proportion of the incoming hot air is conveyed from the base of the chamber direct to the underside of the top and second row of trays.

After leaving the sorting machine and before being packed for dispatch, it is the usual practice to subject the tea to a final firing, in order to dry it thoroughly and to evaporate any moisture it may have absorbed whilst waiting to be packed. The machines used for this purpose are generally of the up-draft or down-draft type, as described.

It may be taken that *4d.* per hundredweight is an average figure of the cost of drying tea.

The table on p. 221 gives the results of tests carried out on Sirocco Tea Dryers, of the endless-chain pressure type, in use on various estates in India and Ceylon. In regard to this, it must be mentioned that the moisture content of the wet leaf entering the dryer varies considerably according to the amount of withering, or natural drying, which the leaf has undergone. In the sixteen tests it will be observed that three classes of fuel were used, namely, coal, oil, and wood. The calorific value of Assam coal is approximately 10,000 B.T.U., whilst oil and wood are respectively 20,000 and 5,000 B.T.U. In tests 1 and 4 of the large dryer and in test 1 of the small dryer, the air heaters were fired by the use of mechanical stokers.

It will be interesting to mention the terms used by tea planters in describing the degree of dryness of tea. The currency of India is employed for this purpose. There are sixteen annas in a rupee, and hence a planter speaks of '12 anna' dry tea, when he means that it is 75 per cent. dry.

Drying of Beet-sugar Pulp.

It has been customary for many years to dry sugar-beet in various ways, but it is only in comparatively recent years that any large measure of success has attended these processes, as in the past it has been found that the beetroot would not keep after having been artificially dried.

The amount of moisture contained in beetroots varies between 77 and 90 per cent., so that it must be obvious that unless great care is taken during drying, the product will be damaged and decreased in value. One of the foremost troubles experienced in the drying of beetroot, and indeed in any drying process, was that due to the burning of the constituents of the root. It has been found that dry beetroot will commence to deteriorate at 212° F., whilst the sugar burns at any temperature above 230° F. When half dry deteriora-

TESTS OF ENDLESS-CHAIN PRESSURE DRYERS

LARGE MACHINE

Test no.	Country	Class of fuel	Amount of damp material per hour	Amount of dry material per hour	Moisture evaporated per hour	Fuel consumed	Lb. of moisture evaporated per lb. of fuel	Lb. of dry material per lb. of fuel	Percentage of dryness of dried material	Remarks
			lb.	lb.	lb.	lb.	lb.	lb.	per cent.	° F.
1	India	Assam coal	1,075	373	702	160	4·4	2·5	92	Dry bulb 90
2	,,	,, ,,	1,247	672	577	137	4·2	4·9	65	,, ,, 90
3	,,	,, ,,	1,744	587	1,157	220	5·2	2·7	86	,, ,, 104
4	,,	,, ,,	1,002	379	623	153	4	2·5	82	,, ,, 100
5	,,	Bengal ,,	1,330	595	735	167	4·4	3·6	83	,, ,, 120
6	Ceylon	Jungle wood (good)	1,013·6	400·24	613·36	432	1·41	0·92	100	,, ,, 83 Wet ,, 75
7	,,	Jungle wood (good)	949	409·5	539·5	359·5	1·5	1·14	100	Dry ,, 95 Wet ,, 83
8	,,	Oil Fuel	1,231	531·5	699·5	108	6·37	4·92	100	Dry ,, 88 Wet ,, 76

SMALL MACHINE

Test no.	Country	Class of fuel	Amount of damp material per hour	Amount of dry material per hour	Moisture evaporated per hour	Fuel consumed	Lb. of moisture evaporated per lb. of fuel	Lb. of dry material per lb. of fuel	Percentage of dryness of dried material	Remarks
1	India	Assam coal	817	436	380	81	4·7	5·3	76	Dry bulb 100
2	,,	,, ,,	762	253	509	120	4·2	2	88	,, ,, 116
3	,,	,, ,,	986	426	559	106	5	4	86	,, ,, 84
4	Ceylon	Jungle wood (good)	742	290	452	226	2	1·28	100	,, ,, 84 Wet ,, 73
5	,,	Jungle wood (green)	633	258	375	402	0·93	0·64	100	Dry ,, 84 Wet ,, 73
6	,,	Jungle wood (good)	651·2	248·6	402·6	398·6	1·01	0·62	100	Dry ,, 85
7	,,	Oil Fuel	800	350	450	81	5·55	4·32	100	,, ,, 80 Wet ,, 74
8	,,	,, ,,	644·83	308·33	336·5	54	6·23	5·7	100	Dry ,, 82 Wet ,, 78

tion commences at 230° F., and when wet at 260° F. It will be apparent therefore that drying must be carefully considered in relation to these temperatures.

Fig. 74 shows a triple-stage drying process† which has been specially designed to facilitate drying beetroots. Each of the three stages of this dryer is separated from the others by air locks, and the beetroots are passed through the three stages on an endless-belt

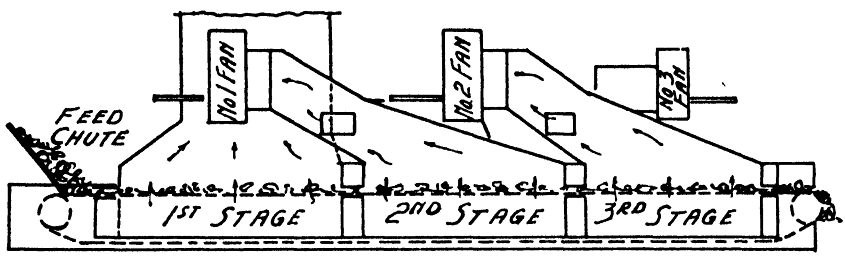


FIG. 74. Sugar-beet drying plant.

conveyor. The temperatures maintained in the various stages are as follows:

Stage 1. 260° F. In this stage about 65 per cent. of the moisture in the cossettes is removed in 15–20 minutes.

Stage 2. 230° F. In this stage about 25 per cent. of the moisture is removed in 20 minutes.

Stage 3. 170–190° F. In this stage the remaining 10 per cent. of moisture is removed in a further 20 minutes.

The hot air is provided by means of mixing atmospheric air with the products of combustion from a coal furnace. The furnace gases are drawn into a mixing chamber from which the first fan draws its air. By means of easily controlled dampers and shutters the exact temperature required can be obtained. This air is forced through the cossettes from below upwards at a velocity of about 175 ft. per minute. The last remaining moisture (it must be remembered that these cossettes are at the third or final stage) is removed and the air thereby increased in humidity to about 4 or 5 per cent., and its temperature reduced from 212° F. to between 170° and 190° F. It then passes to another mixing chamber, where it is reheated by the addition of more furnace gases to 230° F. and atmospheric air is added to increase the volume. Fan No. 2 blows this air from below upwards through the cossettes in the second or intermediate stage at a velocity of 200 to 220 ft. per minute. This air removes further

† See 'The drying of agricultural products', by B. J. Owen, M.A., D.Sc., *Proc. Ind. Chem. E.*, vol. vi, 1928.

moisture, and its humidity percentage on leaving may be as high as 20 per cent., while its temperature is lowered to about 150° F. For the first stage (wet cossettes) a similar system of reheating is employed and the air heated to between 250° and 260° F. Its volume is increased by the addition of atmospheric air and it is blown upwards through the cossettes by Fan No. 1 at a velocity of 250 ft. per minute. This time the air is completely saturated and is exhausted to the atmosphere at a temperature of 110° to 120° F.

The volume of air at each stage is so regulated that it is enough to remove the required quantity of moisture; it is greatest for the wet cossettes in the first stage. The length of conveyor exposed at each stage diminishes from the first stage to the third, and the velocity of the air likewise diminishes. In determining these three velocities consideration must be had, first, of the increasing dryness of the material—a velocity that would fail to move wet cossettes would blow nearly dry cossettes off the conveyor. Further, the velocities are limited by economic considerations.

Hay Drying.

Some interesting experiments conducted by the Department of Agricultural Engineering, of the University of Wisconsin, have indicated the possibility of artificially drying hay through the use of both heating and ventilating equipment. F. W. Dufee, who has had charge of these experiments, outlines what has already been accomplished and the difficulties yet to be overcome. They were able to dry and cure hay artificially with considerable success, although there were many limitations regarding the process which have not yet been determined, such as the minimum desirable temperature, and the minimum amount of air to most economically cure the hay. The maximum temperature which can be used to avoid fire hazard has not been determined nor to any great extent the specifications of the furnace and the fan.

This method involves the handling of 3 to 3½ times as many lb. as when the hay is allowed to dry on the ground. Therefore, the machinery for efficiently handling the hay is of importance, and this has yet to be gone into. One possibility is the use of buck rakes or sweep rakes, together with stackers.

The method briefly described is as follows.

The stack is built over an air cell into which warm air is blown, this finds its way out through the stack, thus drying it out. The general specifications of the equipment used so far are as follows. The air cell is 4 ft. wide at the base and 5 ft. high, made A-shaped.

The length is 8 ft. shorter than the length of the stack. A large pipe, similar to a furnace pipe, conveys the air from the fan into this air cell, over which the hay is stacked. The base of the stack is supposed to be 12 ft. wide, although it can probably be more. Thus there is a layer of hay 4 ft. all around the base of the air cell, the stack being 14 ft. to 16 ft. high.

Use was made of a 2½ Sirocco fan; this size, however, is probably a little too small for a stack of the size used, this being about 20 ft. long and containing 13 good-sized loads of alfalfa hay, which was put in the stack just a few hours after cutting, cut in the forenoon and stacked up in the afternoon. Before the stack was finished a heavy rain fell, thoroughly soaking the stack, but this did not interfere with the drying except that it took a little longer and a little more power.

Use was made of an ordinary warm-air furnace for heating the air and the furnace warm-air duct connected to the intake side of the fan. The smoke stack was turned down and run into the bottom of the hot-air jacket of the furnace, thus blowing all the smoke through the stack. This apparently did not depreciate the quality of the hay in the least. In fact, some claim that the smoked hay is even more palatable than in its natural state. This method greatly increases the thermal efficiency of the drying system and makes it possible to get up a very hot fire in a very short time, and the forced draught keeps it burning reasonably clean. The furnace used had a 36-in. fire-box and was just a common ordinary warm-air furnace which had been picked up second-hand.

It requires only 8 to 10 hours steady blowing to thoroughly cure out a stack of this size if it is stacked uniformly. Uniform stacking is important, otherwise the more compact sides will dry out slower than the other sides. At times the temperature in the stack, that is in the hay itself, is considerably above 200° F.

The air cell is built up of 2×4's or 2×6's and covered with wire fencing or anything that may be handy, to prevent the hay falling into the cell. Roughly speaking, about 100 lb. of coal are required to dry out 1 ton of hay.

They are very anxious to learn the result of any similar work that may have been done along this line. As mentioned, one of the serious problems in curing hay in this way is that of getting the green hay into the stack economically and efficiently, and then moving it a second time into the barn. There is, of course, the alternative of working out a plan to cure the hay in the barn, but so far they have felt that the fire hazard connected with this plan was too great to warrant its general use.

Oil Burner Makes Hay in Eight Hours.

An interesting adaptation of the oil burner has been developed at the Agricultural Experiment Station at Purdue.

The equipment is mounted on wheels for easy transportation, and consists essentially of a cylindrical combustion chamber, and a suction fan operated by the motor of the tractor that provides mobility for the equipment. The fan is designed to handle 7,000 cu. ft. per minute effectively against a rather high resistance. The combustion chamber is lined with refractory brick. Oil is fed to the combustion chamber by a gear pump, at pressures between 30 lb. and 80 lb.

A large volume of excess air is purposely supplied to the burner, ensuring a high degree of combustion and reducing the temperature of the products of combustion to about 200° F. This temperature has proved suitable for making hay or drying soya beans unless it is desired to germinate the seeds, when 140° F. must not be exceeded.

A crib, which may be built up of fence rails or composed of posts set in a circle with wire around and across the top, is heaped with the green hay, and the warm air is blown into the crib, from which it passes through the entire stack. A slight trace of soot may be found on the hay nearest the crib, but the cows of the famous Purdue herd appeared to prefer this to sun-dried hay when offered a choice.

A stack of green alfalfa cut early in the morning was cured in 8 hours, at a fuel cost of 25*s*. for both burner and tractor.

Drying of Agar.

Agar, a product obtained from dried seaweeds, is another material dried by means of hot air.† It has an analysis as follows:

Moisture	18·41	per cent.
Protein	1·12	,,
Nitrogen free extract	76·94	,,
Ether extract	0·19	,,
Crude fibre	0·14	,,
Ash	3·20	,,

The drying of this product, to produce a marketable and efficient agar of constant chemical and physical properties and of high water-absorbing power, involves a problem of considerable magnitude. Ordinary drying methods are clearly impracticable, because of the tendency to case-harden the material and prevent the release of moisture from the interior. Final plans involved drying in two stages, and the adoption of the principle of air flotation as a means of classi-

† *Chemical and Metallurgical Eng.*, May 1927.

fying dried from undried particles. The product of the dewaterer passes, by a short screw conveyor that constitutes a seal, into the lower part of the first humidity drying chamber, consisting of a vertical cylinder of aluminium, 35 ft. high and 3 ft. in diameter, supplied at the bottom with clean hot air by a fan and steam heater.

In this first dryer the agar is kept in suspension in a moist atmosphere until it is light enough to be carried to the top and to pass into a fabric down pipe, through the interstices of which the moist air escapes. The semi-dry agar is trapped at the bottom and delivered into a second screw conveyor, also acting as a seal, and connecting with a second vertical chamber the same size as the first, in which dehydration is concluded. Glass windows in these drying units disclose what appears to be a miniature snow-storm, the upward pressure of heated air being regulated so that the material passes forward only when it is dry enough and light enough to be carried upward the required height. From the final fabric down pipe the agar passes to sacks for packing.

Spray Drying Systems.

It is of interest to refer to the new spray drying system† developed by the Kestner Evaporator & Engineering Co. Ltd.

Such plant is required in the preparation of many special foodstuffs, essences, and extracts, in the preparation of various milk products, and in the fine chemical industry. The advantages arising from the conversion of materials normally obtained in the form of a solution or suspension to a dry powdered state are well recognized, but hitherto the number of materials that could be dried without affecting certain of their essential qualities has been distinctly limited. Many such products are marketed in powder form, but it is well known that the reconstituted solution or suspension prepared from the powder is not always of the same quality as the original material. This deterioration in quality is usually due to the exposure of the product to an excessively high temperature during the drying process, or to prolonged exposure at a temperature which for short periods has no harmful effect.

If the drying operation is carried out in such a way that the material during treatment is maintained at a low temperature, and at the same time it is dried without long exposure, the deterioration in the qualities of the product will be reduced to a minimum. When the question of drying time is considered it is obvious that if the drying temperatures are to be low the time required for the drying

† See *The Industrial Chemist*, March 1927.

operation will increase, other conditions remaining constant. The rate of drying is dependent, firstly, upon the difference in temperature between the direct source of heat and the material under treatment, and secondly, upon the area of the surface of contact between the source of heat and the material under treatment. Other factors have a minor influence upon the rate of drying, but for the moment only the above need be considered.

If, now, the temperature is reduced in order to ensure that the product being dried cannot be overheated, it becomes necessary to increase the surface of contact between the two materials in order to avoid an excessively long drying time.

The Kestner patent spray dryer has been designed in order to provide a suitable plant for the dehydration of the most delicate materials with due regard to simplicity of operation, accessibility for cleaning, ease of control, and general efficiency.

Briefly, the plant consists of a cylindrical drying chamber, in the upper part of which is mounted an atomizer to which the liquid to be treated is fed. The atomizer consists of a special funnel-shaped disk, which is rotated at a high speed. The liquor is fed to the disk by gravity, and is discharged in the form of a finely atomized spray, travelling radially from the disk in a horizontal plane. Warm air enters at the top of the drying chamber and travels downwards, evaporating the moisture and carrying with it the powder.

In order to assist in the atomization and prevent drying before the atomized particles are sufficiently dispersed, a current of cold air is admitted through an air duct which surrounds the atomizer itself, the cold air entering the drying chamber in such a way that it encloses the atomized liquid until the latter has travelled a suitable distance from the disk. The warm air, which is admitted at the top of the drying chamber, passes from an air heater to the air distributor, which forms the upper part of the drying chamber itself. By means of a system of air deflectors, the warm air is caused to enter the drying chamber with a rotary motion, and as it moves downwards it impinges upon the horizontal zone of atomized liquid. The greater part of the moisture from the sprayed particles is evaporated almost instantly, and by virtue of the rotary motion of the warm air the dried powder separates towards the sides of the drying chamber and falls to the conical base, from which it is discharged.

After the warm air has travelled to the base of the drying chamber, it passes upwards with a low velocity through a special form of air outlet arranged in such a way that the air deposits practically the whole of the powder, which would otherwise be carried out of the drying chamber. Finally, the air is filtered through bag dust collectors, with or without cyclone separators.

Considering the drying of solutions, the size of the sprayed particles depends mainly upon the viscosity of the liquor fed to the atomizer and the speed at which the atomizer disk is revolved. The size of the dried particle depends upon the size of the sprayed particle and upon the percentage of solid matter present in the liquor under treatment. In many cases the powder produced will pass through a 200-mesh sieve. Since the whole of the surface of the particle is exposed to the warm air, and, in addition, owing to the difference in specific gravity between the particle and the air in the drying chamber, the constant variation in velocity of the latter will cause considerable relative movement between the particle and the surrounding warm air, thus providing ideal conditions for rapid dehydration. When these conditions are compared with those obtaining in the drum-type dryers, the superiority of the spray dryer for the dehydration of materials adversely affected by heat will be obvious.

The special features of the plant described above and the method of operation may now be considered in greater detail. To obtain a reasonable quantity of available heat from the warm air, it is necessary to admit the latter at a temperature considerably above that to which the majority of delicate products can be heated without risk of damage. However, if the particles of atomized liquid are thoroughly and rapidly mixed with the incoming hot air, the temperature of the latter will fall rapidly, due to the heat absorbed by the evaporation which takes place.

Assuming that the particles and the air are intimately mixed instantaneously, there will be no tendency for the temperature of the particles to rise above that of the final air temperature. By means of the atomization obtained with the Kestner atomizer, and by virtue of the arrangement of the hot-air distributor, such a result is actually obtained in the Kestner spray dryer. The air on reaching the horizontal plane in which the atomized liquid is being distributed is moving with a high rotary velocity and with a relatively low downward velocity, thus the moment the hot air reaches the zone into which the atomized liquid is being distributed, mixing with the most intimate contact and consequent rapid evaporation takes place; thus in the fraction of a second the air is cooled to a temperature slightly above its final exit temperature, the remainder of the cooling taking place as the air travels from the upper to the lower part of the drying chamber. This final cooling is due to the removal of a small quantity of moisture from the almost dry powder which has been formed in the upper part of the drying chamber, and as the air has still a high velocity, by means of which good contact with the powder is ensured, this final stage of the evaporation is carried out with sufficient rapidity to enable a thoroughly dry powder to be removed from the bottom of the drying chamber, although in this region the air temperature is low. The powder never reaches the temperature of the exit air, and as in many cases this can be as low as 150° F. for normal working, it will be seen that the risk of deterioration of the product is very slight. If necessary, even lower temperatures can be used, but this necessitates an increase in the size of the drying plant.

The cold air that is admitted with the atomized liquid has an important influence upon the rapid evaporation. It enables the atomized spray to disperse thoroughly and to be evenly distributed over the top of the drying chamber before the incoming hot air reaches it; at the same time, the cold air serves to cool the atomizer itself and the liquor feed pipes.

With regard to the actual construction of the machine, the atomizer is generally carried from the top of the cylindrical drying chamber, and, for inspection or cleaning, it may be withdrawn without difficulty. When dealing with liquids which tend to leave a deposit in the feed pipes, the latter are arranged so that they may be withdrawn for cleaning without interrupting the working of the plant. The atomizer may be driven by means of a direct-coupled steam turbine or by electric motor and suitable gearing. The power required is low, and care has been taken in designing the atomizer to provide a piece of mechanism that will operate for long periods without attention, and be of high mechanical efficiency.

The control of the plant under normal conditions depends solely upon observations of the exit air temperature. When the most suitable conditions for the treatment of any particular product have been determined, it is only necessary to adjust the liquor feed to maintain a regular exit air temperature. If, during working, the exit air temperature falls, the rate of liquor fed to the plant is reduced until the correct drying conditions are obtained. By varying the speed of rotation of the atomizer, or by varying the type of atomizing disk, the size of the sprayed particles may be adjusted to suit the particular requirements. In general, an increased drying time and a slightly higher drying temperature is required if a powder consisting of relatively large particles is produced. The powder may be removed from the lower cone of the drying chamber continuously, or it may be discharged at frequent intervals to suitable containers.

The plant is designed to operate continuously, but if for any reason intermittent working is required, it is suitable, for owing to the construction the plant may be started up rapidly, and if it is necessary to change over from one product to another at frequent intervals, the ease with which the plant may be thoroughly cleaned renders this an easy matter.

The power requirements are small, as the atomizer requires little power, and, owing to the low air resistance in the plant, that required for the air blowers is reduced to a minimum. The amount of heated air required for a given evaporation depends upon the nature of the material under treatment. If it is possible to work with a high air inlet temperature and a low exit temperature, a process of high thermal efficiency will result. Steam-air heaters, or tubular heaters in which hot oil is circulated by means of the Kestner oil-circulation process (Merrill patent), are employed to heat the air for plants in which it is necessary to avoid any risk of contamination. In certain cases it is possible to operate the spray drying plant with waste hot gases or even with waste flue gases, this latter arrangement being ideal for treating heavy chemicals which must be dried at a low cost. Where waste gases are available but are not sufficiently clean for use in the spray drying plant, indirect air heaters may be operated with such gases. Thus, air heaters may be mounted in the boiler flues and arranged in such a manner that when the heated air is not required for the spray drying plant it is diverted to the boiler front.

It is difficult to enumerate all the classes of materials that can be dried successfully by means of the Kestner patent spray dryer. Perhaps the most important application of the process is in connexion with food products, such as full-cream milk, skimmed milk, whey, eggs, various infant and invalid foods, meat and vegetable extracts, sugar, syrups, etc. Many pharmaceutical preparations can be dehydrated and obtained in a form in which their properties are preserved, while at the same time the powdered product may be stored for long periods without deterioration.

The process is also of great value in the preparation of certain fine chemicals, and the fact that the drying is carried out in the presence of air alone assists considerably in the prevention of metallic contamination, etc.

It is not necessary that the liquid to be treated shall have a low viscosity. Liquors which are comparatively viscous can be atomized without difficulty by means of the most recent type of atomizer. The process is admirably suited to deal with materials which are liquid at elevated temperatures, and which solidify on cooling. Such materials can be produced in powder form in a similar type of plant to the above, cold air being circulated in place of hot air.

The arrangement and operation of the plant are clearly shown on the diagrammatic sectional drawing, Fig. 75.

Calculations for Spray Drying.

In spray drying systems the calculation of the amount of air and the temperature required for drying follows the normal course,

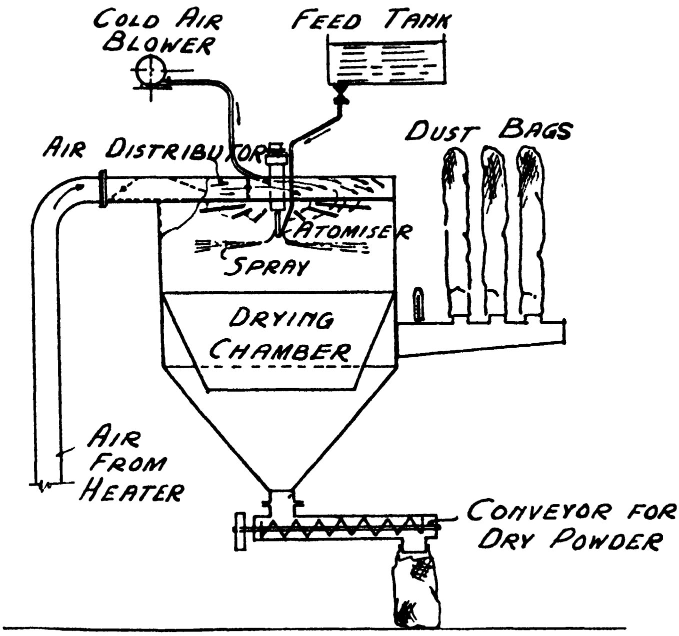


FIG. 75. Kestner spray drying system.

subject to minor allowances found necessary by practical tests. It is usual to heat the air where possible by utilizing waste heat from furnaces. It is often necessary to obtain control of the size of particle precipitated in spray drying, and it has been found that where rotating atomizers are employed the size of the particles decreases as the speed of the atomizers decreases. Similarly, with spray drying systems where atomization is obtained under pressure through nozzles the particle size decreases with an increase in the nozzle pressure.

Calculations for Rotary Dryers.

There are several factors affecting the calculations for rotary dryers which are not of such importance with other systems of drying. It must be remembered that the greater the velocity of the air across the material the more rapid the drying process will be. According to T. J. Horgan,† for materials of a dusty nature having to be dried to a low final moisture content the air speed will have to be kept as low as 2 to 3 ft. per second, whereas for coarse granular products for materials which in the dry condition may still contain a large percentage of moisture, velocities of 12 to 15 ft. per second may be employed.

It follows, therefore, that the diameter of the rotary dryer is controlled by the volume of air which is needed to evaporate and carry away the necessary moisture in the material, and at the same time by the permissible velocity. Similarly, the length of a rotary dryer must depend entirely upon the necessary time which the material must remain in the machine to become properly dried. This in turn depends upon the closeness of contact between the air and the material.

Each material in the case of a rotary dryer should be the subject of exhaustive tests in the laboratory to determine the best drying conditions. As a matter of interest the following table is given illustrating the various technical data in connexion with a typical rotary dryer.

Operating Results with Double-Shell Dryers

Materials	*Iron ore*	*Sand*	*Coal*
Temperature, external air, ° F.	60	40	80
Temperature, exhaust from fan, ° F.	123	98	110
Temperature, material entering dryer, ° F..	48	32	78
Temperature, material delivered from dryer, ° F.	188	210	219
Calorific value of fuel used, B.T.U.	13,540	12,100	14,100
Fuel consumed per hour, lb.	767	398	308
Amount of moisture in material fed, per cent.	17·6	5·93	14·2
Amount of moisture in dried material, per cent.	6·3	0·27	1·29
Water evaporated per hour, lb.	6,321	2,187	2,746
Water evaporated per lb. of fuel	8·25	5·49	8·91
Dried material per hour, lb.	46,144	36,460	18,300
Fuel per ton of dried material, lb.	37·2	24·4	37·7
Heat in exhaust air, per cent.	7·4	8·8	8·4
Heat lost by radiation and other causes, per cent.	8·0	11·6	6·8
Heat used to evaporate water, per cent.	69·0	52·3	69·9
Heat used warming material, per cent.	15·6	27·3	14·9

The Ejector Drying System.‡

The ejector drying system, originally patented in 1918 by W. H. Carrier, has many features of value for drying certain materials.

Many products are very easily injured in conditioning, or drying, heating, or cooling the same by a too rapid or non-uniform drying or treatment, in which the outer portion or certain exposed portions of

† 'Rotary Dryers', by T. J. Horgan, B.Eng. etc., *Proc. Inst. Chem. E.*, vol. vi, 1928.
‡ Patent No. 121080, protection period now expired. Extract from Spec.

the materials are dried, heated, or cooled to a much greater extent than the interior or less exposed portions. In drying macaroni, for instance, a too rapid drying of the outer surface causes the product to crack and break, thereby destroying its commercial value. Similarly, in the curing or drying of green tobacco, a too rapid drying in the first stages kills the living cells before they have undergone the period of starvation which is required to convert the starch into sugar, and later on in the process, for certain grades of tobacco at least, the drying requires to be modulated for the best results in order to oxidize the sugar formed in the previous step and to bring about the desired light brown shade which is required in the highest grades. In the final steps of drying these and other similar products, it is necessary to the attainment of the best results to dry the product to a uniform degree throughout, and yet retain a uniformly distributed final percentage of moisture, to give the products their natural elasticity and retain the other properties which are dependent in a large degree upon the hygroscopic condition of the products.

To accomplish this satisfactorily, three conditions are important. It is essential to maintain a moist atmosphere of regulated temperature and humidity about the material, which conditions can be varied in the process of drying at various stages, as required in the proper manipulation of each particular product.

There must be a thorough and uniform circulation of air over all portions of the material being treated, and any air being introduced into the treating chamber should not come into direct contact with the material except it first be thoroughly mixed with the air surrounding the material, so as to homogenize it with respect to temperature and humidity.

It is necessary that the temperature and moisture content of the air be uniform throughout the drying chamber, so that some of the material shall not dry too fast while other portions of the material dry too slowly. This requires that the air while passing over the material shall not be greatly changed in moisture content and temperature due to the absorption of moisture from the material, with the consequent lowering of the temperature. To ensure this result, the circulation through the material should be relatively rapid, and the rate at which the moisture is given off from the material relatively slow.

Uniformity of air conditions could be secured, as already proposed, by handling an excessively large volume of air and passing it through the material only once and then reconditioning it by a separate apparatus, but this would mean the conditioning of a very large volume of air, and would entail a great expense in the cost of apparatus and in the cost of operation.

The method of drying, conditioning, and regulating the moisture content of hygroscopic materials in a chamber according to this invention, consists of an induced circulation of air in a predetermined path through the chamber, such path having a mixing zone and an exhaust zone, the zones being so arranged that the greater portion of the air when it reaches the exhaust zone is again drawn into the mixing zone by the introduction into the mixing zone of a smaller quantity of conditioned air at a relatively high velocity, the air from the exhaust zone becoming mixed with the charge of conditioned air, and the mixture then circulating through the chamber in the aforesaid predetermined path, and the cycle then recurring.

Thus, with this method, a secondary current of air of relatively large volume is caused to circulate through the material by induction, which is produced by discharging a relatively smaller volume of conditioned air into the chamber containing the material through a series of nozzles or restricted openings. The conditioned air is discharged at a relatively high velocity, usually from two to four thousand feet per minute.

The moisture content, and preferably the temperature also, of the air introduced is normally quite different from the air in the chamber, and it is by a mixture of the air in the chamber with this air of abnormal condition introduced in relatively small volume that the proper temperature and humidity desired in the chamber is maintained. The quality of the air introduced is varied in such a way as to balance the moisture given off or absorbed by the material, with a corresponding drop or rise in the temperature produced relatively by such processes. Thus, in drying material, the air is introduced at a lower moisture content and preferably at a higher temperature than the air in the chamber, while in adding moisture to the material, as is usually required in the last process of drying and conditioning, the air is introduced at a higher moisture content and at a lower temperature than the condition maintained in the chamber.

Usually the material is placed in the chamber so as to leave a space in the chamber at opposite sides of and above the material, and the air is discharged through a row or series of nozzles or outlets near the ceiling at one side of the chamber, and blown horizontally directly beneath the ceiling toward the far side of the chamber, while the air is removed or escapes from the chamber through an exhaust duct or openings, preferably located at the same side of the chamber as the supply nozzle and underneath the same, either adjacent thereto or at the floor line, as preferred. The space about the material should be practically free from obstructions, and should have an area approximately equal to the free area for air passage through the material, and the material should be hung in rows, placed on trays, or otherwise disposed, with the air passages through the same parallel to the direction in which the air is blown. The effect of this arrangement is to induce a large volume of air, usually three to four times the volume of air discharged into the chamber, to circulate and thoroughly mix with the introduced air in the middle space between the material and the ceiling. It then passes downward through the free space

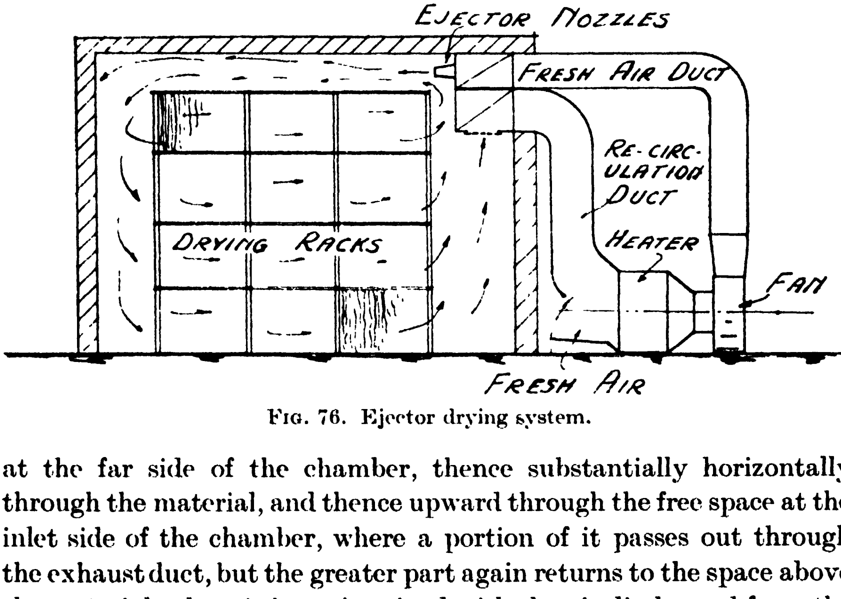


FIG. 76. Ejector drying system.

at the far side of the chamber, thence substantially horizontally through the material, and thence upward through the free space at the inlet side of the chamber, where a portion of it passes out through the exhaust duct, but the greater part again returns to the space above the material, where it is again mixed with the air discharged from the nozzles. This effects a very uniform distribution through all portions of the material, and secures an equally uniform distribution of temperature. As a result, the temperature and the hygroscopic condition of the material, whether in drying or in moistening, is effected uniformly throughout all parts of the material, which has always previously been very difficult, if not impossible, to secure.

The temperature and humidity of the air in the room are regulated under the control of a suitable thermostatic and hydrostatic arrangement, and Fig. 76 illustrates a typical plant.

Refrigeration in Connexion with Drying Systems.

F. Weisker refers in detail† to the use of refrigeration drying systems, where the air used for drying is firstly freed from moisture by cooling

† *Zeitschrift für die Gesamte Kälte Industrie.*

and condensing the moisture content, in particular for the drying of fish. Subsequently it may be reheated before introduction to the drying chambers, and in order to economize in the cycle of operations the hot gas from the compressor may be used for reheating the dry air.

In drying fish the moisture content is reduced from 75 to 15 per cent. in three stages, each removing 20 per cent. moisture. With a total quantity to be partially dried per day of 11,400 lb. there is, therefore, $11{,}400 \times 0{\cdot}2 = 2{,}280$ lb. of moisture to be removed. Fig. 77

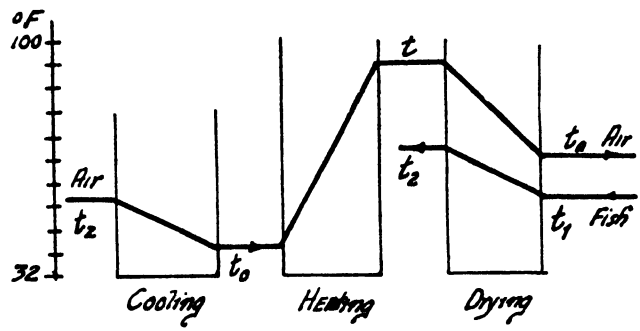


FIG. 77. Fish drying curve.

illustrates the phases of the drying process. The temperature t_z of the external air is assumed as 54° F., and the temperature t_1 of the fish the same. It is taken that the fish leaves at a temperature t_2 of 68° F. The specific heat of the fish is 0·8 and the heat to evaporate 1 lb. of water is 1,060 B.T.U., so that the following heat is required for drying:

Heating fish, $11{,}400 \times 0{\cdot}8(68-54)$ =	128,000	B.T.U.
Evaporating moisture, $2{,}280 \times 1{,}060$ =	2,420,000	,,
Sundry losses, say . . .	252,000	,,
Total Q_t . . .	2,800,000	,,

If the temperature of the entering air is reduced by refrigeration to $t_0 = 41°$ F., then the amount of moisture which 1 lb. of air may contain is reduced to 38·5 grains. With this entering moisture content the air leaves the drying-room, after absorbing moisture in the usual way, at $t_a = 65°$ F., and with 80 per cent. R.H., so that 1 lb. of leaving air contains, as may be seen from Fig. 59, 73·5 grains per lb., and $73{\cdot}5 - 38{\cdot}5 = 35{\cdot}0$ grains per lb. are absorbed.

The amount of air required is therefore:

$$\frac{2{,}280 \times 7{,}000}{35{\cdot}0} = 456{,}000 \text{ lb. per hour.}$$

This amount of air must be capable of giving up 2,800,000 B.T.U. for the various conditions to be satisfied, and must do this in cooling from t to $t_a = 65°$ F. Employing the well-known equation:†

$$Q_t = L \, . \, C_p(t - t_a)$$

$$2{,}800{,}000 = 456{,}000 \times 0{\cdot}24(t - 65), \text{ whence } t = 91° \text{ F.}$$

The cooling or refrigerating stage necessitates cooling the 456,000 lb. per hour of air from $t_z = 54°$ F. to $t_0 = 41°$ F. and condensing out surplus moisture.

Taking the R.H. of the external air as 80 per cent., the moisture content is 49·7 grains per lb., and as it can only hold 38·5 grains per lb. at 41° F., the amount to be condensed out will be 49·7 − 38·5 = 11·2 grains per lb., and a total of 456,000 × 11·2 = 5,100,000 grains.

Taking the heat given up in condensing 1 lb. of moisture as an average of 1,000 B.T.U., we see that the heat given up by moisture is

$$\frac{5{,}100{,}000}{7{,}000} \times 1{,}000 = 730{,}000 \text{ B.T.U.}$$

The heat required for cooling the air from 54° F. to 41° F. will be

$$456{,}000 \times (54 - 41) \times 0{\cdot}24 = 1{,}420{,}000 \text{ B.T.U.}$$

The total heat to be removed is therefore

$$730{,}000 + 1{,}420{,}000 = 2{,}150{,}000 \text{ B.T.U. per hour.}$$

After the cooling process the air is then raised in temperature to 91° F., requiring a heating effect of

$$456{,}000 \times 0{\cdot}24(91 - 41) = 5{,}500{,}000 \text{ B.T.U. per hour.}$$

If the refrigerating machine has an evaporator temperature of 28° F., a cooling temperature of 59° F., and a condensing temperature of 77° F., for the load considered a power consumption of 130 h.p. would be necessary, and the condensing effect is about 2,400,000 B.T.U. per hour. If this heat is now employed for the heating stage, and is taken to raise the temperature of the air from 41° F. to

$$t_x = 41 \; \frac{2{,}400{,}000}{456{,}000 \times 0{\cdot}24} = 63° \text{ F.,}$$

then the heat to be provided by other means becomes

$$5{,}500{,}000 - 2{,}400{,}000 = 3{,}100{,}000 \text{ B.T.U. per hour.}$$

As the total drying period is usually 18 hours, 6 hours being

† For derivation see *Heating, Ventilating, and Air Conditioning*, by Harding and Willard, U.S.A.

occupied in passing through each stage, the figures calculated may be divided by 6, giving the following summarized requirements:

Daily charge	= 11,400 lb.
Effect of drying-room	= 467,000 B.T.U. per hr.
Volume of dry air required . . .	= 17,000 c.f.m. approx.
Refrigeration required	= 360,000 B.T.U. per hr. = 30 tons of refrigeration.
H.P. required by compressor . .	= 22 h.p., equivalent to 56,000 B.T.U. per hr.
Heating stage requirement . . .	= 920,000 B.T.U. per hr.
Heating stage requirement if heat from condensing stage is used . . .	= 520,000 B.T.U. per hr.

Practical experience gives the following schedule for drying fish:

2 hrs. preliminary drying.
48 hrs. state of equilibrium.
4 hrs. drying.
144 hrs. further state of equilibrium.
14 hrs. final drying.

Chapter Six

DISTRICT HEATING AND HOT-WATER SUPPLY SCHEMES

Details of Principal Systems in the World.

ENGLAND is perhaps the one country where least development has so far taken place in connexion with large district heating and hot-water supply schemes. In America and Europe many systems exist, the following indicating the situation and numbers of systems:

United States, Canada, Mexico	165
Germany	17
Austria	3
Holland	2
Russia	2
Denmark	1
Switzerland	1
France	2

It is stated that in the United States of America alone the capital invested in district heating systems is £60,000,000 and the length of pipe-lines in service is nearly 3,000 miles. In Europe, £3,300,000 is invested, to provide 340 miles of pipe-lines.

These figures are sufficient indication of the important growth of district heating abroad.

The New York district heating scheme, the largest in the world, serves pipe-lines extending 255 miles, the farthest point from the central station being nearly 30 miles away. The amount of steam sold per hour reaches 2,500 tons, whilst more than 5 million tons are required per year. It is interesting to note that the amount of steam required is far greater than that available at many of the world's large electrical power stations. Fig. 78 illustrates the distribution of the various systems throughout the world.

In spite of early failures and many years of slow development, the last ten years have seen rapid progress, as Fig. 79, showing graphically the annual steam sales for New York and Hamburg, will show. The increased consumption has generally been at the rate of 20 per cent. per annum. The growth of these systems has, strangely enough, taken place during years of trade depression, and wherever the systems have branched, building development has tended to follow.

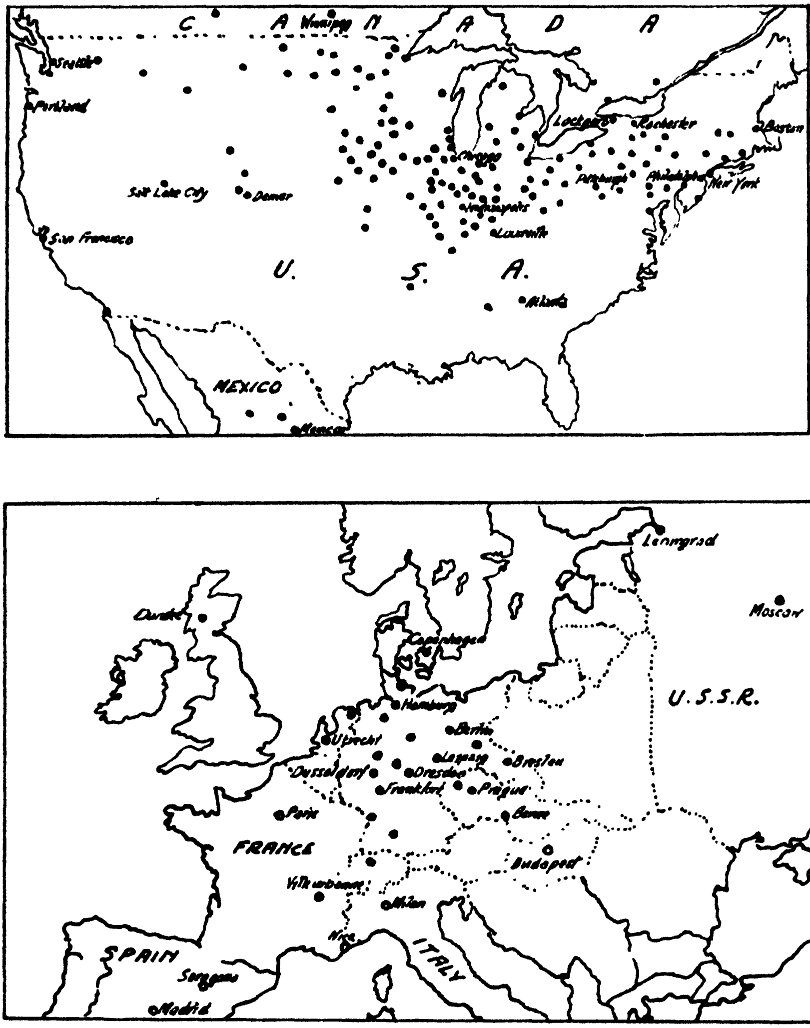


Fig. 78. District heating stations in the world.

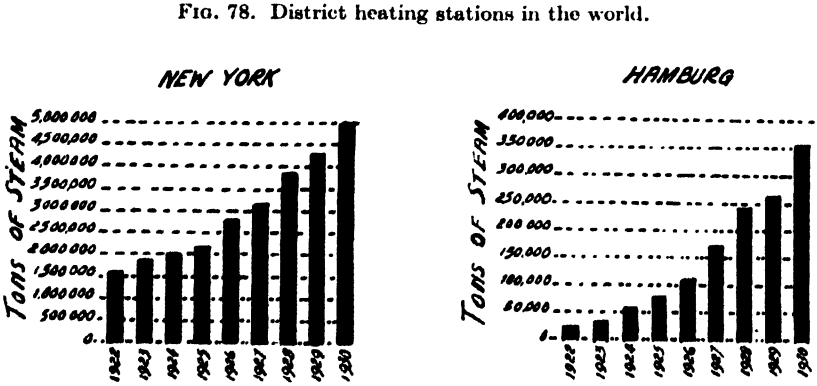


Fig. 79. Steam sales graph.

District Heating in Paris.

In deciding upon a general programme in connexion with their undertaking, the Compagnie Parisienne de Chauffage Urbain at Paris gave consideration to the points which follow.†

The principal district served was in the neighbourhood of the Gare de Lyon, near to Berry power-station, which at one time served the Metropolitan railway of Paris. It was necessary to decide upon the maximum consumption for every class of building, the relation between requirements and external temperature, the cost of installing distributing pipe-lines, management and maintenance charges.

As far as the production of heat was concerned, general study showed that it was cheaper to produce steam bled from turbines than by other means, the cost being in many cases halved by so doing. The alternatives were:

(1) Production at an old power-station, where the capital value had been written off;

(2) Production in a new modern boiler-house, providing for capital amortization;

(3) Taking steam from turbines without regard to the electrical requirements.

It was decided that the system should group power-stations more than 14 miles from the centre of the town by interconnexion of steam lines, thus providing extreme flexibility of the whole system, and enabling all stations to be run at the most efficient points for steam and electrical energy production.

In considering these matters a heat requirement curve was superimposed on the electrical energy curve. Whereas the peak load for electrical energy took place at 5 p.m., the maximum heating load was at 9 a.m.

With a combined electrical and heating station, however, when the electrical requirements are low there is a definite advantage that surplus boilers may be in use for supplying the higher heating load. The other advantage in combining electrical and heating stations is the economy in fuel consumption.

It is stated that in Paris a combined station would use 140,000 tons of coal per year, compared with 175,000 tons for separate stations.

The following figures represent 36 large undertakings in the United States:

	1928	*1929*
Total steam sold (tons)	11,230,000	13,300,000
Steam taken from turbines (tons)	1,620,000	2,282,000
Electrical energy produced (kWh.)	91,000,000	151,000,000

† 'Les relations du chauffage urbain et du chauffage central', by M. Ph. Schereschowsky. *Chauffage et Ventilation*, April 1934.

From these figures we may see that the annual increase in steam sales is about 18·5 per cent., that the amount of steam taken from turbines increased 30 per cent. in a year, and that electrical energy produced increased by 60 per cent. The rapid increase in electrical energy resulted at the same time in reducing the steam consumption from 41 to 33 lb. per kWh., an economy of 20 per cent.

The originally conceived system linking up outlying power-stations was not, however, financially practicable in the early stages, and a beginning was made with a smaller system, which was to provide data for the larger scheme. This smaller scheme, in the neighbourhood of the Gare de Lyon, has an hourly output of 80,000,000

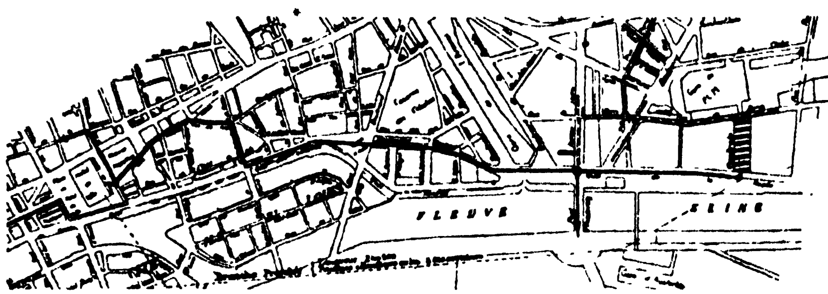


Fig. 80. Berry district heating layout.

B.T.U. The pipes leaving the central station are 10 in. in diameter, and are capable of providing for a heating demand much in excess of that mentioned. The system supplies heat to the following:

Dwelling-houses of various classes,
Office buildings,
Several hotels,
A garage,
Bath establishment,
Railway station, where trains also are heated whilst in the station.

The distributing system has a total length of 4½ miles, conveying steam at a pressure of 85 lb. per sq. in. Steam was chosen as the heat-carrying medium owing to the varied nature of the existing systems in the buildings, comprising, as they did, high- and low-pressure steam, hot water, and plenum plants. In the concession which the heating company has obtained, powers are provided for running either steam or hot-water distribution systems.

The system described proved so successful that at the time of writing large extensions will probably be in course of construction, extending, as shown on Fig. 80, for a further distance of nearly 6 miles, the steam main being 12 in. diameter and condense 7 in. diameter. Two distinct branches are provided, one running to the west with a pressure of 35 lb. per sq. in. and a maximum steam-main diameter of 10 in., with 4 in. condense. Expansion joints are provided every 200 ft.

The branch running to the Gare de Lyon runs at 60–70 lb. per sq. in., with 8-in. steam and 4-in. condense mains. The loss of pressure in the system, passing 15 tons of steam per hour, was found not to exceed 1·5 lb. per sq. in. in a length of 0·62 mile, that is approximately 2·5 lb. per mile. The usual steam-flow formulae were found to give losses of pressure far in excess of those found in practice.

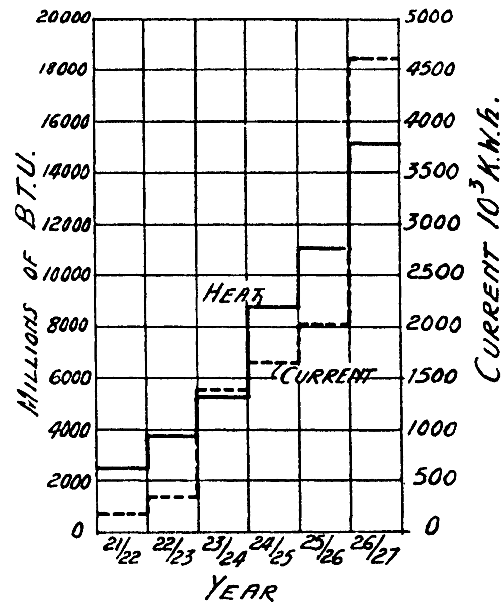


FIG. 81. Heat and current graph.

District Heating in Hamburg.

Steam for this system is derived from the Poststrasse power-station at a pressure of 170 lb. per sq. in. by a battery of 9 tubular boilers, each with 2,700 sq. ft. of heating surface. The original plant was installed as far back as 1894, and generating equipment is of a very old type. Steam is taken from the engines at 6 lb. per sq. in., run through grease separators to a distribution header, a live steam make-up line being provided.

Another power-station in the Carolinenstrasse has a battery of 16 boilers, 11 with superheaters. The total heating surface is 43,000 sq. ft. Steam at 170 lb. per sq. in. is superheated to 570° F., and taken to a turbine running at 3,000 r.p.m. driving a 2,000 kW. dynamo at 750 r.p.m. The electrical system serves a tramway system. The exhaust steam, at a pressure of 3–7 lb. per sq. in., is conveyed to the Poststrasse heating-station by a pipe-line 1·3 miles long.

The steam mains in this system are in places 32 in. diameter, the total length being 5 miles.

The heating system is in service for about 270 days each year, and Fig. 81 shows the quantities of heat and electrical current supplied during 1921–7. Fig. 82 is a typical day chart showing the fluctuation of the Hamburg system. It is stated that the maximum amount of live steam required to make up requirements has never exceeded 20 per cent. of the whole. The maximum heating output has been 220 million B.T.U. per hour, whereas the estimated total requirements of the whole town is 6,000 million B.T.U. per hour.

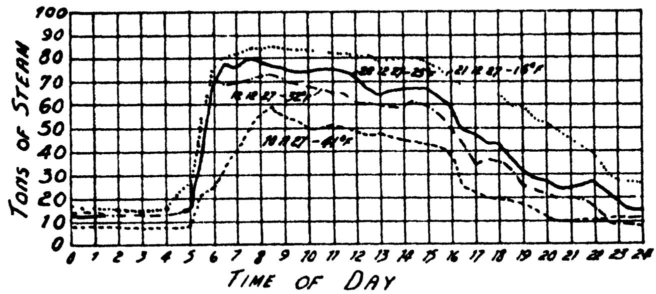


Fig. 82. Hamburg day chart.

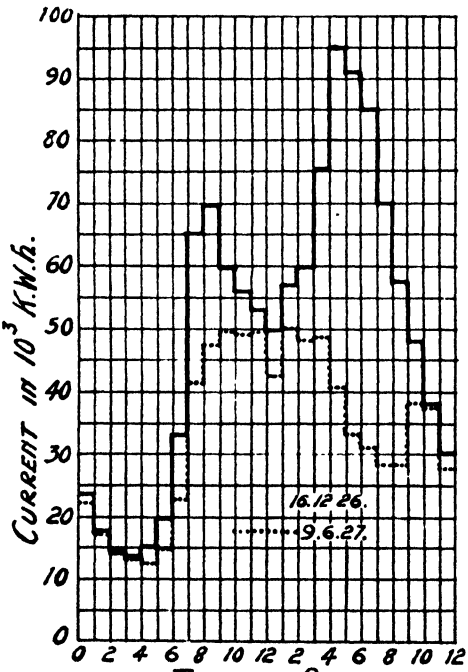


Fig. 83. Hamburg electricity load.

Future developments are expected to increase the output to 700 million B.T.U. per hour, but even then only a small part of the field of sales will have been covered.

Fig. 83 shows the electrical current produced in Hamburg in 1926–7 for a winter and a summer day. The peak load is 95,000 kWh. in winter and only 50,000 kWh. in summer.

The annual coal consumption in domestic fires is 600,000 tons, whilst the power-station consumes 220,000 tons. The extensions of the heating system are likely to deal with the steam necessary for 45,000 kWh., which should then lead to large reductions in the selling-prices of either heat or electrical energy.

Barmen, Prussia.

The boiler plant serving the district heating scheme in Barmen, Prussia, consists of 4 tubular boilers of 1,938 sq. ft. of heating surface and 43 sq. ft. grate area, and an evaporation of 11,900 lb. of steam per hour. High-pressure steam is taken to the various buildings which the system serves, and is either reduced to a lower pressure or serves a calorifier with a hot-water heating system in the building.

With this particular district heating scheme it is found that 10–25 per cent. saving is effected on the cost of heat compared with what was possible before centralization.

The Combined Power and Heating Station.

With all district heating schemes we are faced in the initial stages with the problem of deciding upon whether the heating plant shall be run as a separate and independent equipment with its own self-contained boiler units, or, alternatively, whether it shall be served by the boilers already installed for serving an electrical generating station, making use either of exhaust steam from the engines or turbines or the heat normally taken to waste by the condensers. A further point to be decided in the latter case is whether heat is to be considered as a by-product and sold as such, or, on the other hand, if it is to be debited with some proportion of the capital cost of boilers, subsidiary plant, and buildings.

In this country we have many important electrical generating stations, and the present policy tends towards the construction of power-stations, which in the case of the Battersea works of the London Electric Power Company will have an ultimate output of as much as 500,000 kWh.

The general policy of providing super power-stations concentrates enormous boiler power at one place, which for several reasons is usually remote from the district in which the heating demand would exist, this providing initial difficulties in combining the heating-station with the power-station. These difficulties may, however, be easily overcome.

The amount of heat which normally is wasted is sufficient to deal with the heating demands of a vast area. It must be remembered that under 10 per cent. of the heat value of the coal burnt at the power-station is converted to the equivalent of electrical energy.

With the centralization of the generation of electrical energy, many existing small generating stations are being employed only as sub-stations, with the result that the boiler plant of the smaller stations is being scrapped. This is no doubt an extremely short-sighted policy, for the boiler plant could have returned large dividends had it been used for supplying heat to the surrounding district. Such stations are, moreover, situated where heat is required, so that every foot of distributing pipe-work would be paying a dividend.

A sound proposition would have been to link up the smaller stations with the large in such a way that steam could be carried from the super power-station to heat-distributing stations at the smaller old stations. When the whole of the steam output of the super power-station is required to provide for electrical output, the smaller boiler plants could then deal with their own heating load.

Similarly, a heating system would be served from the super power-station which, when necessary, could be assisted by steam obtained from the outlying district stations. A careful survey of the heating and electrical loads at various hours and seasons, compared with the steam generated at the various stations, would indicate the maximum possibilities of such a scheme. There is no doubt that the most efficient district heating scheme is that in which several steam-generating sources are linked together. This method involves of necessity a vast undertaking, but need not all be carried out in the initial stages provided the ultimate scheme is kept in mind. Capitalization of the whole scheme as one initial undertaking would provide a problem which it is not likely would be overcome, but to present the ultimate scheme, at the same time commencing upon such a scheme, or part of the whole, as would most rapidly produce dividends with the least expenditure of capital, could not fail to be successful.

Apart from such large schemes there are other opportunities for providing district heating to deal with new villages or townships, where domestic rubbish, which has always to be disposed of by local authorities, can be usefully employed in providing electrical energy and heat. If there is insufficient rubbish for this purpose, other fuel can also be employed. It would also be possible to take steam lines from some outlying power-stations to serve compact villages.

It is not possible to define one method of approach which would solve all problems of district heating, for each must be dealt with on its merits. It appears inevitable, however, that development will eventually take place from the existing power-stations. As the electrical supply undertakings already have parliamentary powers of great use in running distributing systems through roads and streets, they are in a strong position, if sufficiently interested, to develop district heating without the delays which might of necessity occur were an entirely new undertaking to attempt to obtain the necessary powers.

Heat Wasted at Power-Stations.

There are few power-stations which are able to convert more than 8 per cent. of the heat value of the fuel into useful electrical energy. The remaining 92 per cent. is wasted in several ways. Firstly, the boiler plant alone could not have an efficiency of more than 80 per cent., but the losses at this source could largely be recovered by waste-heat boilers operating in conjunction with the district heating scheme.

The greatest losses, however, take place in the generating plant and auxiliary equipment, for the whole of the steam used for driving turbines or engines is condensed and returned to the boilers, the heat removed from the steam in condensing being wasted by the use of cooling water, which is returned to a river, or by a cooling tower.

Additional losses are represented by the power required for circulating cooling water, and depreciation on condensers, pumps, or cooling towers and their subsidiary equipment.

For sound organization, electrical energy should be regarded as a by-product, and the plants assumed to be run for heating output, for the proportion of the heat value of the fuel represented in electrical energy is very small indeed.

There is without doubt ample demand for the heat which could be provided.

Modern development of thermal storage systems would help considerably in dealing with peak load demands, and enable boilers to be run at constant load for providing conserved steam or heat to deal with the peak demand.

The World's Principal Power-Stations.

It will be interesting to consider a number of the principal power-stations in the world with a view to determining the approximate amount of heat which is normally wasted at full electrical output.

The Elbing power-station of the Ostpreussenwerk A.G. is equipped with two turbo-generators of 8,000 and 10,000 kW. capacity. The boiler plant is operated at 385 lb. per sq. in., whilst 327 lb. per sq. in.

is provided at the turbines. The details of the four boilers are as follows:

Boiler heating surface, each	6,200 sq. ft.
Normal steam capacity	51,000 lb./hour.
Continuous maximum	76,000 ,,
Temporary ,,	94,000 ,,

The West power-station at Berlin will have an ultimate output of 228,000 kW., generated by six turbo-generators, steam being supplied at 455 lb. per sq. in.; eight boilers are provided, each having 26,000 sq. ft. of heating surface. Each of the boilers is capable of an output of 218,000–335,000 lb. of steam per hour.

The East River generating station of the New York Edison Company had in 1927 two 60,000 kW. generators. Each of the boilers can produce a normal commercial rating of 50,000 lb. of steam per hour, with a continuous overload up to 250,000 lb. per hour. The steam-pressure is 375 lb. per sq. in.

The Ferrybridge power-station has eight boilers, each capable of an output of 75,000 lb. of steam per hour, at a pressure of 315 lb. per sq. in., to serve two turbo-generators of 19,000 kW. each.

As an example of linking up power-stations to serve heating systems it is interesting to note that the Newcastle-upon-Tyne Electric Supply Company and its associated company supply an area of 1,600 sq. miles, with a maximum demand of 175,000 kWh. The North Tees 'B' power-station has two turbo-generators of 25,000 kW. maximum output and one of 20,000 kW., making a total output of 70,000 kW. The boilers consist of eight water-tube type, each of 12,028 sq. ft. of heating surface, and an evaporation each of 75,000 lb. per hour of steam at 475 lb. per sq. in.

The following will eventually be the outputs of various other stations:

Hell-Gate (New York)	650,000 kW.
Hudson Avenue (Brooklyn)	680,000 ,,
Crawford Avenue (Chicago)	750,000 ,,
East River (New York)	1,250,000 ,,
State Line (Chicago)	1,000,000 ,,
Golpa (Germany)	430,000 ,,
Klingenberg (Germany)	540,000 ,,
Saint Ouen (France)	400,000 ,,
Gennevilliers (France)	340,000 ,,
Vitry Sud (France)	500,000 ,,
Saint Denis (France)	400,000 ,,
Arrighi (France)	500,000 ,,
Battersea (London)	500,000 ,,

The Saint-Denis II station was originally installed to deal with

15,000 kW., with six boilers each capable of producing 20,000–27,000 lb. of steam per hour.

The Vitry-Sud generating station has commenced with an output of 220,000 kW., with six boilers each giving 302,000 lb. of steam per hour at 585 lb. per sq. in.

The brief reference which has been made to only a few of the world's power-stations is sufficient to give an idea of the enormous waste of fuel which takes place unless a district heating scheme is worked in conjunction with the plant.

The amount of steam required per kWh. at the power-station varies between 8 and 12 lb., according to the type of plant installed. With most modern power-stations 9 lb. is the usual figure at full output. If a district heating scheme is employed, perhaps 80 per cent. of the heat value of the steam can be usefully employed, and, taking an average pressure of 500 lb. per sq. in., the amount of heat available per kWh. is $9 \times 0{\cdot}80 \times 1{,}000$ (approx. heat available per lb. of steam), that is 7,200 B.T.U. Thus 1,000,000 B.T.U. per hour is the output which might be considered to be obtained from

$$\frac{1{,}000{,}000}{7{,}200} = 140 \text{ kWh.}$$

The possibilities of available heat for district heating at various power-stations will therefore be as follows:

Power-station	*Millions of B.T.U./hour*	
	Present	*Future*
Hell Gate	..	4,650
Hudson Avenue	..	4,850
Crawford Avenue	..	5,370
East River	860	9,000
State Line	..	7,150
Golpa	..	3,070
Klingenberg	..	3,850
Gennevilliers	..	2,430
Vitry-Sud	1,570	3,570
Saint-Denis II	..	2,850
Saint Ouen	..	2,850
Arrighi	..	3,570
Battersea	1,785	3,570
North Tees 'B'	535	..
Ferrybridge	273	..
West Berlin	..	1,620

Our own country is as far behind in power-station development as it is in district heating, which one might say has not yet been attempted on any large scale.

District Heating in Great Britain.

There is only one district heating scheme of any magnitude, and that is situated in Dundee, Scotland, and even so is ridiculously small compared with plants abroad.

The scheme provided for heating and hot-water supply services for 260 dwelling-houses, and the farthest house is only 740 yds. from the central heating-station, heat being charged for at 3*s.* 2*d.* per week for a two-room and 4*s.* per week for a three-room house. It was found that owing to the smallness of the whole scheme the financial return was not very great. Overhead costs had to be charged which would not have been increased had the scheme been many times larger.

Moreover, a boiler plant was installed for the one purpose of serving the heating system, without employing refuse as fuel.

Possibilities of Battersea Power-Station for District Heating.

We have seen that at Battersea power-station alone there is a possibility of having available as much as 3,570 million B.T.U. per hour in waste heat only. Unfortunately, as any one knowing the district surrounding the power-station would agree, a very small market exists for heat in the immediate neighbourhood. The most likely district, the great business district of London, lies on the opposite side of the Thames, and is at its nearest some 1½ miles from the power-station.

The heat available at the power-station when operating at electrical full load could be obtained as exhaust steam. It must be remembered that the modern power-station maintains a definite vacuum on the condensers, so that it would not be possible to run steam supplies from the power-station to where heat is required without due regard to this matter. It might perhaps be possible to replace or convert the existing condensers so that they become calorifiers to serve a hot-water heating system, which could then radiate to the necessary points of the district. Alternatively, it would perhaps be possible to bleed high-pressure steam from a stage of the turbine and to run this to distant points as required.

The use of the hot-water distribution system has some advantages, because it is possible to obtain the maximum heating load even at electrical peak load. If high-pressure steam distribution is employed, there is a limit to the amount of steam which can be bled from the turbines, so that it is not possible for the heating system to develop to the full capacity of the boiler plant.

A further alternative might exist in distributing high-pressure steam direct from the boiler plant to where it is required, having the use of boiler plants in other districts, where the power-stations are to become redundant, during peak-load periods. If such boiler plants were not available for any reason the necessity would arise for installing further boiler plant to provide for heating load during electrical peak load. The provision, on the other hand, of steam thermal storage in the form of a Ruths accumulator system, and limitation of the maximum heating load to such as will allow the boilers to serve the steam accumulator during light electrical load periods, is a further possibility.

To distribute the vast amount of heat which the boilers could provide by a centralized hot-water system is particularly difficult, due to the large pipe-lines required and the considerable power required for circulation, so that this system must be abandoned.

The most satisfactory solution would lie in distributing steam at an initial pressure of 100 lb. per sq. in. to sub-stations at convenient points, from whence pump accelerated hot-water systems could radiate. To provide for the coincidence of electrical and heating peak loads, a steam accumulator system would be installed.

In distributing steam at 100 lb. per sq. in. the size of steam and condense mains would be such as would be practicable, and the cost of the mains far less than for centralized hot water.

Examining a map of the inner business area of London, it may be seen that it is first necessary to cross the river to the north side before coming to any closely built areas. To do this it would be advisable to remain on the south side, following the river as far as Vauxhall Bridge, there crossing to pass to a sub-station at a convenient point in Pimlico, following the Embankment and up Whitehall and Charing Cross Road to another at a point near the British Museum, finally passing along New Oxford Street and thence to the City to a further sub-station.

From these three main sub-stations it may be expected to connect to the largest heat consumers, but it is always possible to provide for additional points on the south side of the river if a demand exists relatively close to the power-station. In the initial stages of carrying out the work it would be desirable to instal the whole of the main steam-distributing lines and sub-station plant, for a delayed and slow-growing scheme would be sure to fail, owing to the time in which capital invested at the steam-distributing station would be locked up and giving no return. As the capital cost of the main steam lines, sub-station, and power-station equipment and incidental works may be as much as £250,000, it will be appreciated how important this may be.

But in order to ensure rapid earning power it could no doubt be arranged to pick up all buildings adjoining the steam-distributing system, as work proceeds with the mains. As ultimately with full possible load connected, the annual income may be as much as £750,000, assuming a minimum gross return of 25 per cent. on capital would mean that £3,000,000 would be invested, and taking the average capital outlay to be £15,000 per mile of distributing system would mean that not more than 200 miles of pipe-lines could exist, and the connected load must be obtained with this amount, or less.

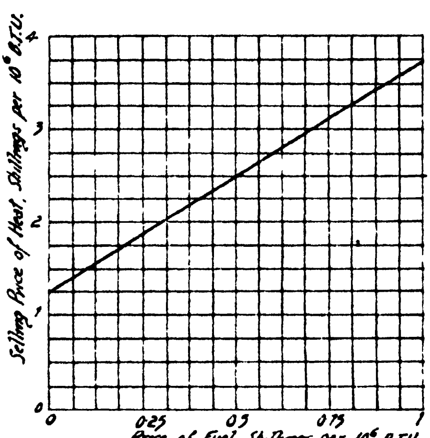


FIG. 84. Selling price of heat.

The Financial Problem of District Heating.

Apart from the provision for high capital expenditure for financing district heating schemes, a problem exists in deciding upon a scale of charges for heat. Such scales are usually incorporated in the powers given to the company and must therefore of necessity foresee variations in labour rates, the cost of fuel, and perhaps also the bank rate, for it is conceivable that at times recourse would need to be had to bank loans in connexion with the preliminary financing of extension schemes.

The return which could be expected on capital must remain problematical, for it depends entirely upon the proportion of connected to potential load. It would in many sections of the suggested system be possible to get a return as high as £10,000 per annum per mile, but this is unlikely for several years.

The sale price of heat in Germany has in places been adjusted to the curve in Fig. 84, which gives the selling price per 1,000 lb. of steam in relation to the cost per ton of the fuel used. In this connexion, it is interesting to note that selling prices as high as 4*s*. per 1,000 lb. of steam have been suggested, but this figure is far too high when it is remembered that the average heat output from 1 ton of fuel used in a privately owned heating system produces effective heat equal to 10,000,000 B.T.U., that is the equivalent of 10,000 lb. of steam, at a cost of perhaps 2*s*. 6*d*. per 1,000 lb. The selling price must be lower than 2*s*. 6*d*. per 1,000 lb. of steam to compete with private enterprise, and indeed is considerably lower with most of the large district heating companies.

Another authority† at one time suggested, in connexion with a scheme designed to have a boiler plant independent of electrical generation, an equivalent selling price of 6*s*. per 1,000 lb. of steam, but this system comprised 8·5 miles of main distributing pipes, the connected load being only 5,000,000 B.T.U. per mile and the cost per mile £8,000. Such a district was actually not capable of being economically served by a district heating scheme. It should be mentioned, however, that the suggested selling price was also to cover depreciation of the internal heating systems of all the buildings. Such systems are usually provided by building owners, only the cost of main distribution falling upon the district heating company.

The suggested basis of design for this system was, in the author's opinion, far too lavish to be economically possible.

The actual price of sale of heat must be determined for each system by taking into account the cost of fuel, the anticipated efficiency of the plant, the depreciation charges, overhead charges, and profit. It is not possible to fix a general price without due regard to all these items.

A District Heating System for 5,000-Inhabitant Village.

As an example of the problems which would need to receive consideration in designing a scheme for district heating to serve a whole village it is proposed to consider the district shown in Fig. 85, which was a design placed first in a competition organized by the *Builder*, entered by Mr. C. P. Meddick, to whom we are indebted for permission to reproduce this layout.

The layout was intended to deal with a population of 5,000 people, and was meant to provide a self-contained village of an ideal nature. It will be observed that the buildings provided for are:

Railway station and bus station.
Civic centre, shops and business premises.
Infants' school.
Elementary school.
High school.
Hospital.
Churches and chapels.
Swimming baths.
Light industrial centre.
Residential buildings.

† *A Treatise on Central Heating*, by F. Broadhurst Craig.

It is conceivable that eventually an estate of this nature may develop to 50 per cent. larger than the present size, so that due regard must be had to this possibility. The present site, being planned

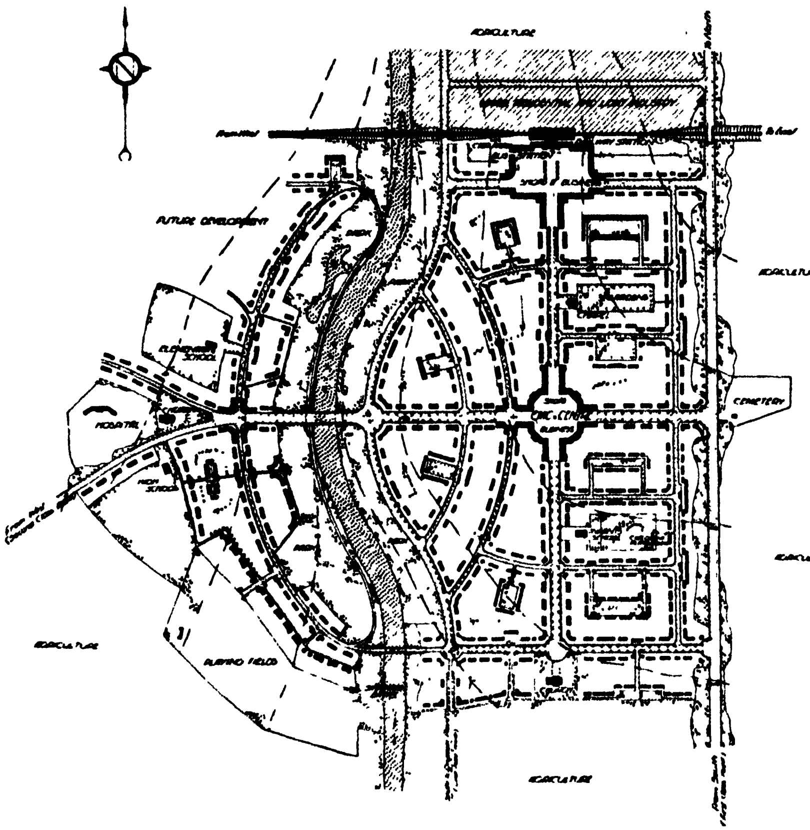


FIG. 85. Proposed village layout.

on the garden city principle, has extreme dimensions of 1,500 and 1,300 yards, occupying a total area of about 405 acres.

Total Heating Requirements of District.

Before commenting on the choice of system or any such matters we should firstly assess the probable total heating requirements of the district and that for possible future building schemes. If a scheme were being considered in practice, the heat losses for all buildings would usually be assessed according to the cubes, after typical buildings had been calculated in detail by the established methods, referred to in Chapter I. For our purpose the figures will be calculated as follows:

Buildings	*Total B.T.U./Hr.*
Railway station and bus station	300,000
Civic centre	2,500,000
Shops and business premises, near station	2,000,000
Factories, etc., north of station	4,000,000
Infants' school	300,000
High school	500,000
Elementary school	500,000
Hospital	1,500,000
Church west of river	700,000
Church in south	700,000
Chapel	500,000
Swimming baths	1,500,000
1,500 houses	75,000,000
Total maximum	90,000,000
Possible future load	45,000,000
	135,000,000

Total Hot-Water Supply Requirements of District.

The principal requirements for hot water would be for residential buildings, the actual load being estimated according to the principles outlined in Chapter II. These may be summarized as follows:

Buildings	*B.T.U./Hr.*
Civic centre	250,000
Shop, etc., near station	200,000
Factories, etc.	250,000
Infants' school	50,000
High school	50,000
Elementary school	50,000
Hospital	300,000
Houses (1,500)	30,000,000
Total maximum	31,150,000
Assumed future load	13,850,000
	45,000,000

Possibility of Combining Power and Heating Stations.

Although it does not fall within the scope of this work to give detailed consideration to the electrical power requirements of the district, the necessity for considering the combination of the power and heating stations must be apparent, to avoid duplicating boiler plant.

It is likely that the generating station for the district would need to have a total capacity of 6,000 kW. with provision for dealing in the future with a maximum load of 8,000 kW. We are now faced with the first problem in the fact that in a district where the whole of the buildings are connected to the heating system the minimum amount of steam required by a modern turbo-generator is too small to deal with the heating load. It is likely that the 8,000-kW. plant would be run on a condensing rate of 15 lb. of steam per kW., that is, a total of 120,000 lb. per hour. The available heat equivalent of this would be approximately 120,000,000 B.T.U. per hour, which is perhaps only 60 per cent. of the ultimate heating and hot-water load including distribution losses. Although it would be desirable in the case of a village of the type considered to have the minimum capital expenditure debited to the district heating scheme, it is evident that it must carry some part of the boiler and auxiliary plant.

Let us now calculate the necessary boiler power in view of heating and power demands. The district heating system requirement is:

Load	*Millions of B.T.U. per hour* Present	Future	Total
Heating	90	45	135
H.W.S.	31·15	13·85	45
Losses in distribution assumed as 5 per cent. of future total .	7	2	9
Grand totals	128·15	60·85	189

These figures represent approximately thousands of lb. of steam per hour.

On the basis of 15 lb. of steam per kW. the turbo-generators would need:

Thousands of lb. of steam per hour Present	Future	Total
90	30	120

The maximum heating demand is likely to exist at an hour when the power demand represents only approximately 66⅔ per cent. of the peak load. Assuming the heat in the steam taken from turbines to be used for heating, the amount available would be:

Millions of B.T.U. per hour Present	Future	Total
60	20	80

The additional heat required to be provided for heating peak load would therefore be:

Millions of B.T.U. per hour Present	Future	Total
68·15	40·85	109

It would no doubt be decided to put in four generators each of 2,000 kW., three being used in the initial scheme.

Reviewing the whole of these figures, it would appear desirable to use eight boilers, each capable of providing 25,000 lb. of steam per hour. Of these six would be installed in the initial stages. It would then be decided to charge the capital cost of half of the boilers and auxiliary plant against the district heating scheme.

Method of Heating Distribution.

We have now to decide upon the general method of heating distribution. For a scheme of the size of that being considered, with a reasonably level site there is but little doubt that hot-water distribution is likely to be most suitable and economical. It will, therefore, be decided to employ an accelerated hot-water system with a maximum flow temperature of 200° F. and return 150° F., that is, a temperature drop through the system of 50° F. In distribution by a hot-water system on so large a scale as this, it is permissible to size pipes in such a way that the maximum velocity in the largest pipe is 10 ft. per second, decreasing with the diameter of the pipes so that a constant pressure drop per foot run is obtained in the whole system.

It has been decided that the power and heating station shall be situated in the factory area at the north of the site, as near to the railway as possible, so that coal supplies are easily obtained on private sidings.

From the power-station, a main flow and return pipe-line would run across the railway down the main street, branching to the subsidiary streets and to cross to the areas on the west side of the river, of sufficient capacity to serve the future buildings which are likely to be erected there.

The pipes are then sized from the accelerated hot-water pipe-sizing calculator, Fig. 20, in this way:

From page 54, the velocity is given as

$$V = \frac{\text{B.T.U.}}{(T_f - T_r)1{,}200D^2}.$$

Substituting the known values for the maximum output of 189,000,000 B.T.U. per hour we have:

$$10 = \frac{189{,}000{,}000}{50 \times 1{,}200 \times D^2},$$

whence $D = 18$ in. approx.

Referring to the pipe-sizing calculator, we see that an 18-in. pipe carrying 189,000,000 B.T.U. per hour has an average loss of 0·08 in. per ft. run. The remaining pipes are now sized for the average 0·08 in. per ft. pressure loss.

Pumping Equipment.

From measurement, the maximum flow and return travel may ultimately be 14,000 ft., so that the total loss of pressure in the system against which the pumps must work will be:

$$\frac{0 \cdot 08 \times 14{,}000}{12} = 93 \cdot 4 \text{ ft.}$$

The maximum quantity of water to be delivered will need to be:

$$\frac{189{,}000{,}000}{50 \times 10 \times 60} = 6{,}300 \text{ gallons per minute.}$$

The power absorbed by the pumps can be obtained from the following formula:

$$\text{B.H.P.} = \frac{\text{G.P.M.} \times 1{,}000 \times H}{33{,}000 \times E},$$

where B.H.P. = brake horse-power required,
G.P.M. = gallons per minute,
H = head in feet,
E = percentage efficiency of pump.

In our case, assuming that the total duty is taken as 7,000 G.P.M. at 100 ft. head, to provide some margin, the power required is:

$$\frac{7{,}000 \times 1{,}000 \times 100}{33{,}000 \times 50} = 425 \text{ H.P.}$$

The total duty would be covered by eight pumps, six being installed in the initial scheme.

Calorifiers.

The cost of calorifiers would be debited against the heating scheme. The maximum duty would again be split up into eight units, six for the initial scheme, each having an output of about 33,000,000 B.T.U. per hour.

Some of these would of course be replacing condensers normally required by the generating plant, but as it is assumed that one company would operate power and heating services, this point will not be taken into account.

Capital Expenditure.

We will firstly estimate the capital expenditure of the system as it would be installed without extensions, for at this stage the capital invested in proportion to the loading would be high, and must, therefore, govern the selling price of heat.

We may summarize the various capital costs as follows:

Three boilers and auxiliary plant, buildings, etc.	£30,000
Distributing pipe-lines, with cost of excavation	£50,000
Calorifier and pumping equipment	£12,000
Total	£92,000

Running Costs.

With a combined power and heating plant on the lines discussed, obviously a large proportion of the heat is likely to be obtainable as a by-product of the power plant. This may, therefore, be charged at a low figure against the district heating system, any income derived by the electrical system in this way being in the nature of a subsidy, enabling lower charges to be made for electrical energy. We must, however, decide upon the proportion of heat to be obtained in this manner.

By comparison with records of heating and power systems it is likely that the figures would be as follows:

Month	*Total heating load millions B.T.U.*	*Millions B.T.U. from power-station*	*Balance to be provided by heating station boilers*
Sept.	1,890	700	1,190
Oct.	12,900	4,900	8,000
Nov.	43,800	16,700	27,100
Dec.	59,000	22,500	36,500
Jan.	56,600	21,500	35,100
Feb.	55,000	21,000	34,000
March	46,500	17,800	28,700
April	43,000	16,400	26,600
May	28,000	10,700	17,300
	346,690	132,200	214,490

Firstly, let us assess the fuel which would need to be burned to obtain 214,490 million B.T.U., in the heating station boilers.

With coal having a calorific value of 14,000 B.T.U. per lb. and a boiler efficiency of 75 per cent., there will be 10,500 B.T.U. obtained per lb. of fuel, or $10,500 \times 2,240 = 23{\cdot}6$ million B.T.U. per ton. The quantity of coal burned in the heating season would therefore be:

$$\frac{214,490}{23{\cdot}6} = 9,120 \text{ tons.}$$

Apart from fuel costs there are many fixed charges which would need to be apportioned to the maximum output, these being as follows:

Depreciation and repairs, 5 per cent. on capital outlay of £92,000	£4,600
Wages of stokers, six men for 30 weeks and two for 22 weeks at £2 10*s*. per week	£560
Salaries of administrative staff	£1,500
Power for pumps at preferential terms from electricity system	£1,500
Total	£8,160

Let us assume that an arrangement is made whereby steam is taken when available from the power-supply company at a price of 6*d*. per million B.T.U. The cost of heat from this source will be

$$\frac{132{,}200 \times 6}{240} = £3{,}300 \text{ per annum.}$$

The total fixed charges are therefore likely to be

Depreciation, power, wages, and salaries	£8,160
Purchase of steam from power company	£3,300
Total	£11,460

This represents against the total estimated sales of 346,690 million B.T.U. a charge per million B.T.U. of

$$\frac{11{,}460 \times 240}{346{,}690} = 7{\cdot}9d.$$

The price of coal will affect the addition to be made under such heading, but it should be possible to arrange for bulk delivery at 15*s*. per ton, costing:

$$9{,}120 \times \frac{15}{20} = £6{,}850$$

which represents a further

$$\frac{6{,}850 \times 240}{346{,}690} = 4{\cdot}75d. \text{ per million B.T.U.}$$

It would be reasonable to add as much as 25 per cent. to these figures to cover for interest on capital outlay and profits, so that the retail price of heat could be $1{\cdot}2(7{\cdot}9+4{\cdot}75) = 15{\cdot}25d.$, i.e. 1*s*. 3¼*d*. per million B.T.U.

The gross income derived would therefore be

$$\frac{346{,}690 \times 15{\cdot}25}{240} = £22{,}000.$$

It should be remembered, however, that it would be perhaps three years before a scheme of this nature was complete and earning at this rate, and it is well to ensure that the scheme is not likely to fail through initial under-capitalization. On the other hand, when contemplated extensions are made, the further capital to be expended, compared with the return to be expected, is low, for the initial scheme will have paid for the main distributing pipe system.

Notes on Distributing Systems.

Forgetting now the specific system with which we have been dealing, there are a number of points in connexion with district heating schemes which may well be dealt with.

It is usual to have all pipe-lines with welded joints, in view of the cost entailed either in inspecting pipes at various points after excavation, or of having tunnels of sufficient size for access. With steam distribution many of the United States systems do not return condense, but this procedure is certainly not advisable if efficient working is to be obtained. There has been much controversy regarding the use of iron or steel tube for condense pipes, but, although copper is certainly better able to resist corrosion, the additional cost is such that no steam district heating system of any importance employs anything but iron or steel. The Hamburg system, after being in use for seven years, showed no signs of corrosion of the condense return pipes.

The insulation of pipe-lines calls for some care, and many systems have been evolved for this purpose. In America it has been common practice to use the pipes in insulating material with metal or ceramic outer finish, burying them directly in the ground. This method has been found successful, but the insulation is expensive.

In Germany, it has been customary to employ common insulating materials, providing tunnels or subways to take the pipe-lines. The usual dimensions of these used by the firm of Rud. Otto Meyer are as follows:†

Key No.	*Diameter of steam main in inches*	*Dimensions of subway for steam main and condense*	
		Width (in.)	*Depth (in.)*
D1	2–2½	18	14
D2	3–4	20	16
D3	4½–6	24	18
D4	7–9	28	22
D5	9½–11	32	24
D6	12–14	36	28
D7	15–18	40	30
D8	19–20	44	34
D9	22	48	36
D10	24	52	40
D11	28	56	44
D12	30	60	48
D13	32	64	52
D14	36	68	52
D15	40	72	56

The subways are usually constructed in reinforced concrete, with inspection chambers at intervals. The roofs of subways are either flat or curved, according to the load to be carried, but it is advisable to assume a loading of 12 cwt. per sq. ft. as the minimum. The construction should be of a water-tight nature to prevent water

† *Chaleur et Industrie*, April 1930.

entering, and in cases where the surface water head is high, the subways must be absolutely water-tight and are necessarily costly.

In the case of the Barmen system, also in Germany, the pipes were insulated, cased in steel, and finally buried directly in the earth.

There is another system of insulation which is now being extensively adopted, and that is the cell concrete or alternatively the 'Reform' insulating system, both of which permit the pipes to be buried directly in the ground, with a considerable saving in cost.

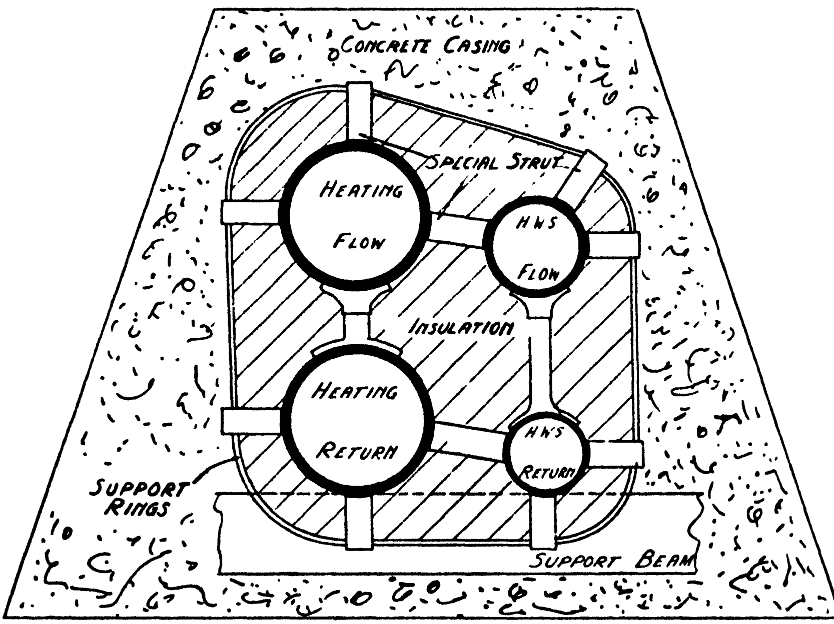


FIG. 86. Reform insulating system.

With the cell concrete system of insulation the pipes are first covered with oiled paper and are then surrounded with suitable shuttering so that the cell concrete may be poured around the pipes. When the material is set, the shuttering is removed and the outside of the cell concrete is then covered with a cement coating suitably reinforced. The cell concrete itself is a cellular cement product made from the same ingredients as ordinary cement mortar but with the addition of a foaming substance similar to soap suds. During the mixing of the material the foam bubbles become covered with a coat of cement mortar, so that when the mortar sets a material is formed containing numerous small air-filled cells spread uniformly through the material. The material has the advantage of absorbing only very minute quantities of water, so that it is particularly suitable for direct use when buried in the ground.

The 'Reform' insulation system is also designed for direct burial in the ground. But this system consists essentially of a concrete tube supported by means of radial brackets so that it surrounds the pipe leaving an inner space which is filled with insulating material packed to a suitable density. Supporting radial brackets are arranged at 2 ft. centres and provide tunnel support for the pipe allowing for the whole of its length. The outer protective casing of concrete is almost independent of the pipe, so that expansion or contraction of the pipe may take place without causing any cracks. As the concrete used is rendered water-proof, there is no possibility of water entering the inner casing. The concrete casing is also designed with a view to being sufficiently strong for supporting the pipe-lines and insulation and to provide protection against the weight of earth on top of the pipe. Fig. 86 illustrates the use of the 'Reform' insulating system for a group of four pipes, showing how they are arranged within one outer concrete casing.

Chapter Seven

STEAM DISTRIBUTION AND BOILER PLANTS

Design of Piping Systems for Various Uses.

THE flow of steam in pipes is a further instance in which the nomographic calculator may be applied with advantage, eliminating a large amount of tedious repetition in calculations.

The formula introduced by Meir has been found by the author to be satisfactory in practice, but his charts for various pressures have the disadvantage of requiring the application of correction factors according to the steam-pressure.

The formula used is

$$P_f = \frac{W}{144} 0{\cdot}0257 \frac{v^{1\cdot95}}{2g} \frac{l}{d^{1\cdot2}},$$

where l = length of pipe in feet;
d = diameter of pipe;
g = acceleration of gravity, ft./sec.^{2};
W = density of steam, lb./ft.^{3};
P_f = friction loss;
v = velocity, ft./sec.

The nomographic calculator in Fig. 87 is used by a similar procedure to others which have been described in more detail. The adjustable scale in this case is that indicating the loss of pressure, and is adjusted so that the indicating arrow points to the pressure which it is desired to work.

For example, it is required to find a suitable size of pipe to convey 2,000 lb. of steam per hour a distance of 100 ft., the initial steam-pressure being 100 lb. per sq. in. and the allowable loss 10 lb. per sq. in. The loss allowable per 10-ft. run therefore becomes $\frac{10\times 10}{100}$ $= 1$ lb. By setting the pressure-loss scale C to $\frac{100+(100-10)}{2} = 95$ lb. per sq. in. (the mean pressure) on the adjacent pressure scale we find that a 2-in. pipe would be required with a loss actually much less than that allowable, namely, 0·66 lb. per 10 ft.

The velocity is a factor which again varies with the steam-pressure or more correctly speaking the density of the steam. At higher pressure, the volume equivalent to a given weight of steam becomes less, so that the velocity is less than it would be for a lower pressure.

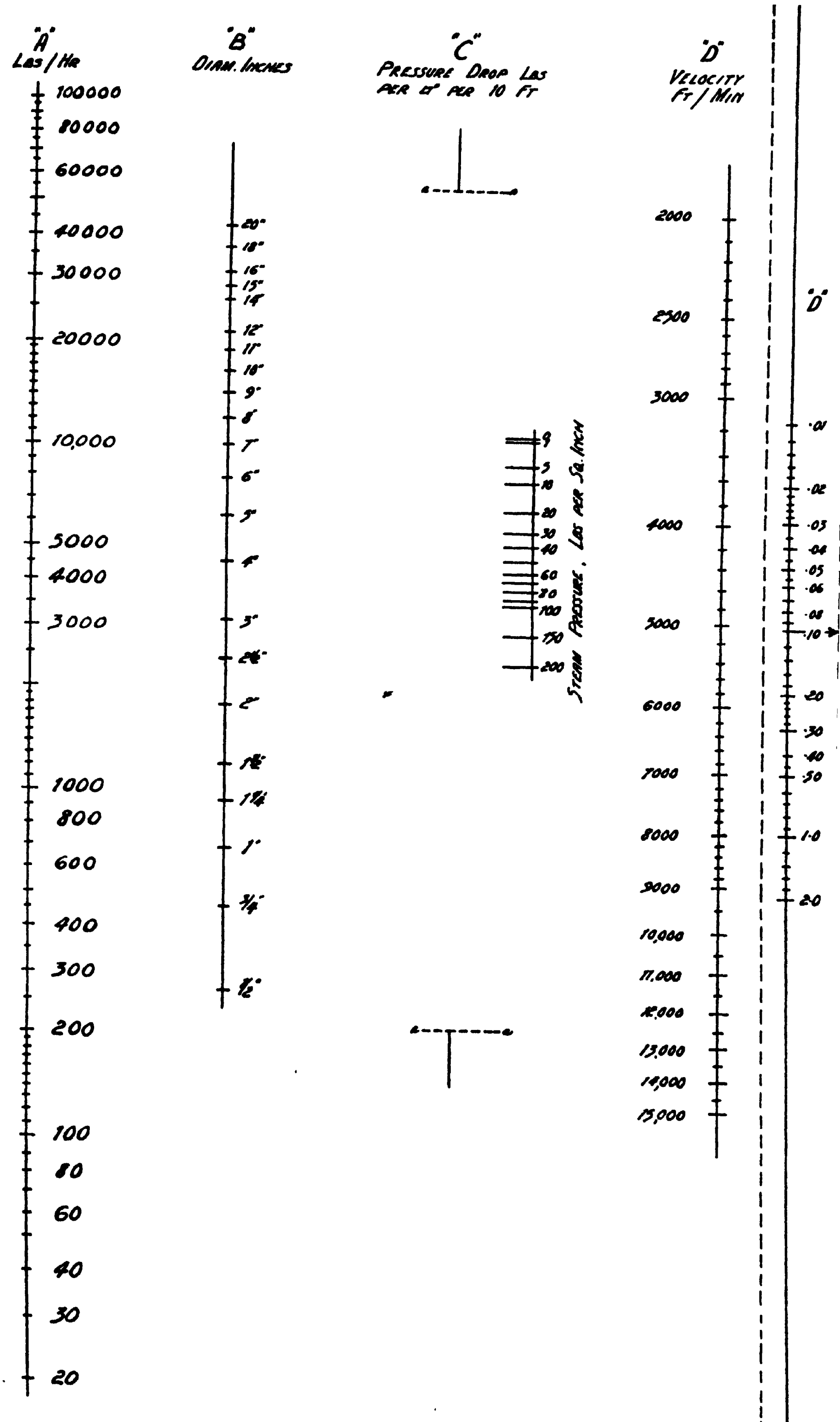


FIG. 87. NOMOGRAPHIC CALCULATOR FOR STEAM PIPES

Actually we may say that $v \propto \frac{1}{W}$. In order therefore to obtain the correct velocity for any desired conditions the correction must be made, taking W from steam tables.

To the inexperienced, the choice of suitable pressure losses or velocities is a matter of great difficulty, but generally it may be taken that the loss of pressure should not be greater than 20 per cent. of the initial steam-pressure, and the velocity not in excess of 9,000 ft. per minute.

There are, however, many instances where special circumstances dictate higher or lower values, but these cases are ones which experience and judgement alone can decide. The nomograph in Fig. 87 can be used for all cases where the steam-pressure exceeds atmospheric pressure.

The vacuum steam-heating system is one which calls for careful design of the piping system. With this system the maximum initial pressure is never likely to exceed 2 lb. per sq. in., and in some instances is practically negligible or less than atmospheric. The vacuum pump is capable of a suction effect of perhaps 10 in. of mercury, which is equivalent to a pressure differential of about 5 lb. per sq. in. It is usual to assume a maximum loss of pressure of 2 lb. per sq. in. in sizing pipes for the vacuum steam-heating system, even for large installations. The sizes of condense mains are not determined by formula, for they rarely if ever run full. Except with the vacuum system, a satisfactory procedure is to take condense mains two sizes less than for the steam. For the vacuum system, however, owing to the positive suction on the condense line it is possible to have condense mains four sizes less than the steam.

Expansion of Pipe-Lines.

Too little care is given to the problems which are produced by the expansion of steam or hot-water piping systems. With small installations at low pressure or temperature there is a very small chance of the expansion not being self-compensating by the nature and arrangement of the piping itself, but in large distributing systems the increased length of piping when high temperatures are reached and the occurrence of long straight lengths of piping makes careful consideration essential. Fig. 88 is a nomograph from which the increase in length of pipes at various temperatures according to the temperature at which they are fixed may be determined.

There is no doubt that the most satisfactory device for compensating expansion is the expansion loop, where there is sufficient room to accommodate it, but in other circumstances the copper-bellows type of expansion joint is reliable. Fig. 89 shows the proportions of expansion loops recommended by the author, and these should be arranged in the pipe-lines at not exceeding 200 ft. apart,

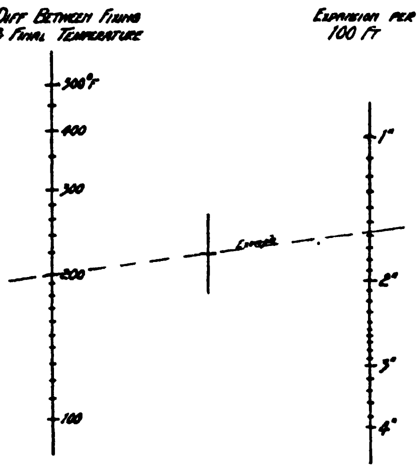
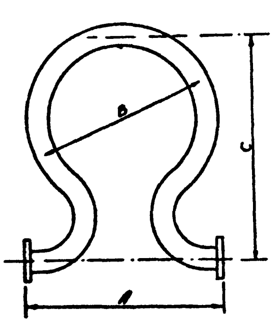


FIG. 88. Pipe expansion nomograph.



Diam.	*A*	*B*	*C*
2"	1' 8"	1' 6"	2' 1"
2½"	2' 3"	2' 0"	2' 9"
3"	2' 7"	2' 4"	3' 4"
4"	3' 1"	2' 9"	3' 9"
5"	3' 5"	3' 0"	4' 2"
6"	4' 0"	3' 6"	5' 0"
7"	4' 0"	3' 9"	5' 3"
8"	4' 0"	4' 0"	5' 6"
9"	4' 6"	4' 6"	6' 0"
10"	4' 6"	5' 0"	6' 3"
11"	5' 0"	5' 6"	7' 0"
12"	5' 6"	6' 0"	8' 0"

FIG. 89. Expansion loop.

with intermediate anchor-points at which the pipes must be immovably fixed.

Exhaust Steam and its Application to Modern Heating and Hot-Water Supply Systems.

Where steam is available at sufficient pressure to drive engines, pumps, or fans, particularly where turbines are employed, there is a definite economy in the use of steam, which may afterwards be employed for heating purposes in calorifiers or air-heaters. The modern large hospital plant offers many opportunities for this arrangement, especially where it is convenient to have a generating plant for serving all internal electrical requirements without recourse to the supply companies.

There are many alternatives by means of which the exhaust steam may be employed, but the wisest design is to arrange for the steam drives never to produce more exhaust steam than can usefully be employed for heating purposes. During the summer months when the only heating requirement is for domestic hot water, it is therefore necessary to resort to electrically driven plant, as otherwise exhaust steam would have to be discharged to the atmosphere and wasted. It should be noted, however, that the turbine- or engine-driven generator requires considerably less steam per kW. than would be used by smaller turbine-driven units which are nominally reputed to be doing the same work, so that in the summer it is economical to drive all units electrically.

At one time there was a definite tendency in design to have large institutions served by calorifier units in the individual blocks with steam taken from the boiler-house to each block. The heating system in each block would then be of the hot-water type. An arrangement of this nature, whilst in terms of heat units may perhaps be proved as efficient as any other, represents bad design for several reasons. In the first instance it leads to apparatus which should be under constant supervision being hidden in calorifier rooms at considerable distances from the boiler-house, where the engineering staff are available. Secondly, and what is of greater importance in connexion with the hot-water supply services, each block must be served by a unit capable of dealing with the maximum load of that block, whereas distribution of hot water from a centralized calorifier plant to all blocks allows boiler plant to be capable of the average load of the combined system, instead of the maximum load of the whole job.

There are, of course, instances where old installations need to serve remote building extensions, and in some of these cases there is no practical alternative to running steam to the extension to serve the block in question. Even so, if accelerated hot-water circulation is used, the pumps should be steam turbine driven, as also should any pump required for returning condense to the boiler-house, the exhaust from these being used in the calorifiers of the system they are serving.

Where constant supplies of exhaust steam are available and the heating load is steady there is much to be said for the use of a vacuum steam-heating system, in view of the ease with which the engine or turbine steam-consumption may be varied to suit heating requirements, and the gradation of temperature which is possible in the radiator itself. The principle of all vacuum steam-heating systems is similar and they differ only in the use of various special fittings, but with all systems the volume of steam admitted to the radiator may be varied in order to control the heat emission. For instance, by the use of a control valve on the steam inlet to the radiator the amount

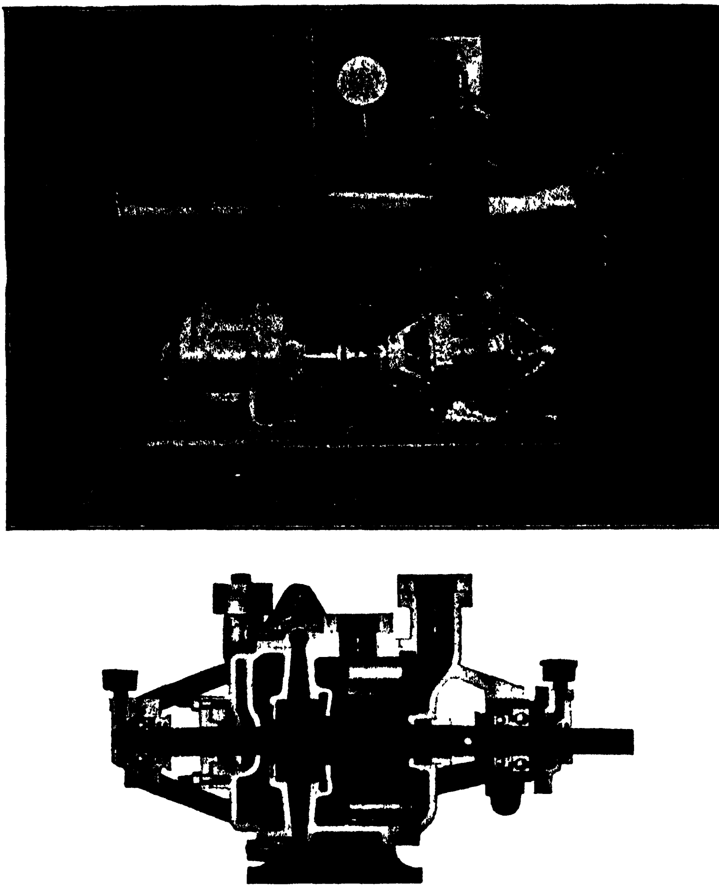


Fig. 90. Vacuum heating pump.

of steam may be varied, and at the same time if only a small amount of steam is admitted, the vacuum in the radiator will be greater, and consequently the radiator temperature lower. With the full 10-in. vacuum the temperature will be 192° F. and with 5-in. vacuum 203° F. Moreover, the radiator need only be partially filled with steam, which gives further control of the emission.

When exhaust steam is available, this is firstly taken through a grease or oil separator. Live steam is taken for make-up purposes through a reducing valve, and in the event of too much exhaust steam being available for requirements it passes through the back pressure valve to atmosphere.

The modern vacuum steam-heating system is served by a turbine vacuum pump, illustrated in Fig. 90, which combines the duty of vacuum producer, boiler feed pump, and air-separator, and is arranged for electric drive. Two independent turbine pump units are combined in one casing, on a common shaft, one exhausting air and vapour from the system, and the other taking the condensation from the system and returning it directly to the boiler or a hot well. The unit may either run continuously or be controlled by a vacuum regulating and float switch.

Where the older type of steam-driven vacuum pump is employed, this is controlled by a vacuum governor. When the vacuum reaches the desired value, the atmospheric pressure below the diaphragm acting against the vacuum and the pressure of the spring above it retains the valve sufficiently open to admit only the steam necessary to keep the pump moving and maintain the desired vacuum. If the vacuum drops, the pump is immediately increased in speed until the desired level is again reached.

The vacuum steam-heating system is particularly useful for application to low-pressure steam-heating systems which are inefficient in operation (forming actually a parallel to the use of a pump on a faulty hot-water system), for it needs only to have modulating radiator valves and thermostatic traps to each radiator and a complete vacuum pump unit to convert the system to efficient operation.

The Kiesselbach High-Pressure Thermal Storage System.

There are instances, more especially with works where large quantities of steam are required for process work, where the average requirements are at times, and for short periods only, greatly exceeded. It is well known that a boiler plant can only operate efficiently when run to capacity, so that if steam boilers are installed to do the maximum or peak load, the operation over the full working period is inefficient. The usual procedure with a steam boiler plant is to shut off the feed when big demands occur, and take advantage of the water stored in the boiler between high and low water-levels, for whilst this is done no heat is required for raising the temperature of feed water, and consequently a greater evaporation is possible, for a short period. Even so, the water-level must not be allowed to drop below the safety line. The Kiesselbach thermal storage system,†

† See *Iron and Coal Trades Review*, June 10, 1927.

illustrated in Fig. 91, as applied to a water-tube boiler makes use of this procedure in a practical manner.

With other types of boilers the general layout is on similar lines. A battery of boilers can be connected up to one storage reservoir, and it does not matter if several types of boilers are represented. The reservoir is an ordinary plain boiler drum. An existing boiler can be detached from the battery and converted into a storage reservoir, in order to lessen the initial capital outlay. The circulating pump is always in operation, its working capacity exceeding considerably the quantity of water absorbed for steam production at any time. The surplus water delivered into the boiler flows over and back

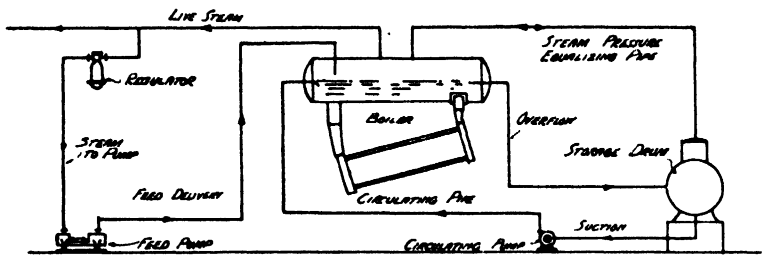


Fig. 91. Kiesselbach system.

into the storage reservoir. Both boiler and reservoir steam chambers are directly connected by equalizing pipes, thus equalizing the pressure throughout.

When the demand for steam is less than production, that is, on light load, the feed pump delivers more water into the boilers than is absorbed for steam production. The surplus water becomes heated and absorbs the excess heat units made available while the plant is working on light load. The surplus water overflows into the storage reservoir, which is heavily lagged with asbestos or other heat-insulating material. When the demand for steam exceeds average requirements, the feed pump slows down and may even become stationary, and during such periods hot water for steam production is taken from the storage reservoir. As this water from the storage is at high temperature, all the heat from the firing is made available for steam production. It follows, therefore, that the boilers are capable of yielding increased steam supply without suffering fall in pressure, notwithstanding the fact that firing continues at a constant rate.

As will be seen from the diagram in Fig. 92, a boiler plant equipped with the 'Kiesselbach' storage system requires no special control or regulation outside the apparatus, which automatically delivers water into the system proportionately to whatever quantities of steam are taken. The feed pump automatically responds to fluctuation in the steam pressure, and if the load falls below the average, the pressure will rise and the pump will be accelerated. On the other hand, when the load rises above the average the pump slows down or stops altogether. In the case of steam-driven feed pumps the regulator is inserted in the steam delivery pipe, but if the pump is driven electrically it is inserted in the water pipe on the pressure side. In both cases the regulation is entirely self-acting, no oil or water being required for it. The regulating apparatus is simple in design and

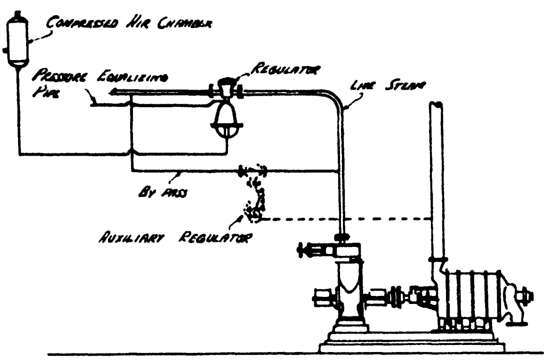


FIG. 92. Kiesselbach system.

construction; it will operate with variations in steam pressure from 1·5 to 3 lb. per sq. in. Boiler plant fitted with economizers or preheaters are in addition provided with a separate regulator, which is also self-acting in relation to the pressure obtaining with the water feed-pipe. As the water pressure decreases when the main regulator is out of action the quantities of water passing through the economizer are so regulated as to preclude any inadmissible increase in temperature, and it automatically provides against undue fall in pressure.

Ruths Accumulator Systems.

Of the many systems of thermal storage which have been evolved, the Ruths accumulator system, particularly for industrial plants, is extremely valuable.

In the majority of industrial plants the steam demand is characterized by short sharp peaks developing and falling off with great rapidity. In addition, major variations may occur lasting an hour or more, during which the average demand assumes a higher or a lower level than the general average taken throughout the day. The short sharp peaks are due to the intermittent nature of the steam demand of certain sections of the plant, while the major variations are due to the discontinuous operation of large individual steam consumers or groups of lesser steam consumers.

Generally, the steam consumers fall into two well-defined groups—high-pressure consumers such as power generating plant, pumps, compressors, process units requiring high temperature, etc., and low-

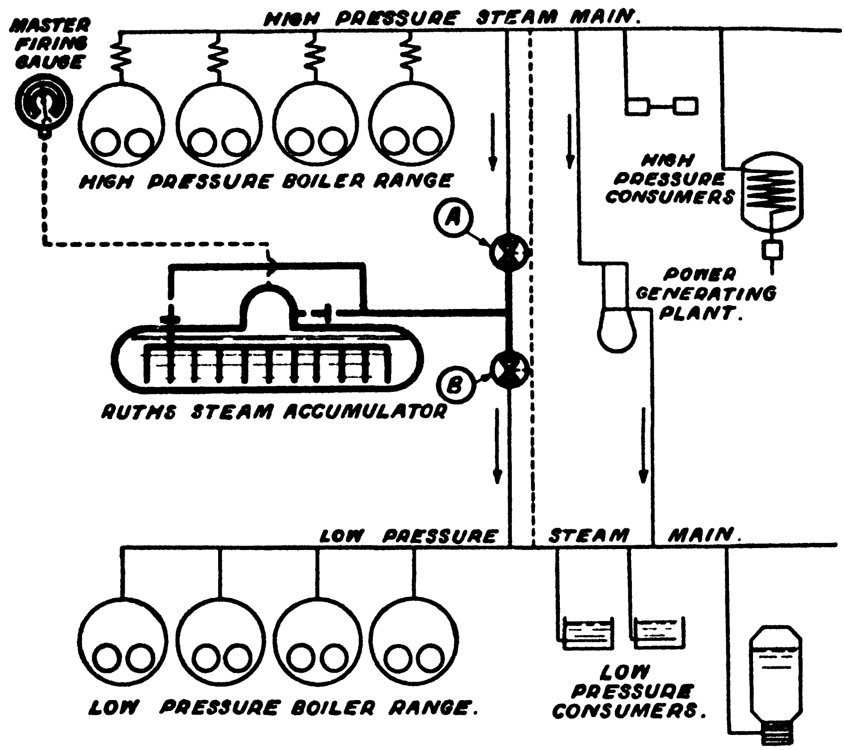


FIG. 93. Ruths accumulator system.

pressure consumers, such as dye kettles, soap-boiling pans, evaporators, etc. In addition, it is often the case that steam is generated partly at high pressure and partly at low pressure. Such conditions are highly favourable to the installation of the Ruths steam storage system. Fig. 93 shows a typical arrangement, A being the overflow valve and B the reducing valve, to which reference has already been made. A master firing gauge located in the boiler-house shows the amount of steam in storage and gives the boiler-operating staff timely warning of any necessary change in the rate of firing.

In such works the steam demand usually varies as much as 60 to 80 per cent. above and below the average demand, as shown in Fig. 94. Without a steam accumulator the conditions can only be met by constant change in the rate of firing, while the steam pressure necessarily fluctuates over a wide range, handicapping all steam consumers and lowering their rate of output. In addition, the working conditions are against the development of high thermal efficiency in the boiler-house, and lead to steam wastage by the consumers.

With the Ruths system installed, the boilers would be fired at a

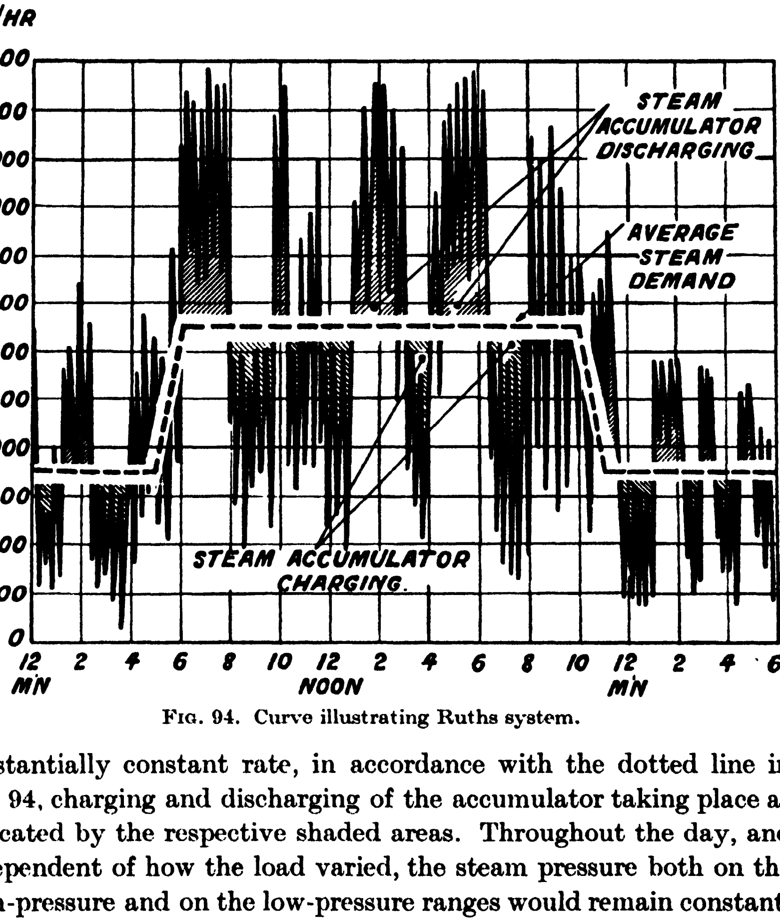


FIG. 94. Curve illustrating Ruths system.

substantially constant rate, in accordance with the dotted line in Fig. 94, charging and discharging of the accumulator taking place as indicated by the respective shaded areas. Throughout the day, and independent of how the load varied, the steam pressure both on the high-pressure and on the low-pressure ranges would remain constant.

In some works the main consumers operate at boiler pressure and conditions are unfavourable to the use of the steam storage system. Under these circumstances, the Ruths Feed-Water Accumulator may be employed with advantage. The underlying principle of this is very similar to the Keisselbach system, in that surplus steam is utilized for the building up of a reserve of pre-heated feed water, at or near saturation temperature; the peak load carrying capacity of the system is, therefore, dependent upon the permissible increase in feed temperature. In the majority of cases where the feed-water accumulator is applicable, peak loads 25 to 30 per cent. in excess of the average demand may be met without drop in steam pressure.

In general, the accumulator is arranged in open communication with the high-pressure steam main, as shown typically in Fig. 95. All feed water entering the system is first dealt with by the accumulator feed-pump. The rate of discharge of this is controlled by the valve C, which is opened and closed by a pressure impulse taken from the high-pressure steam main. The feed water passes through the

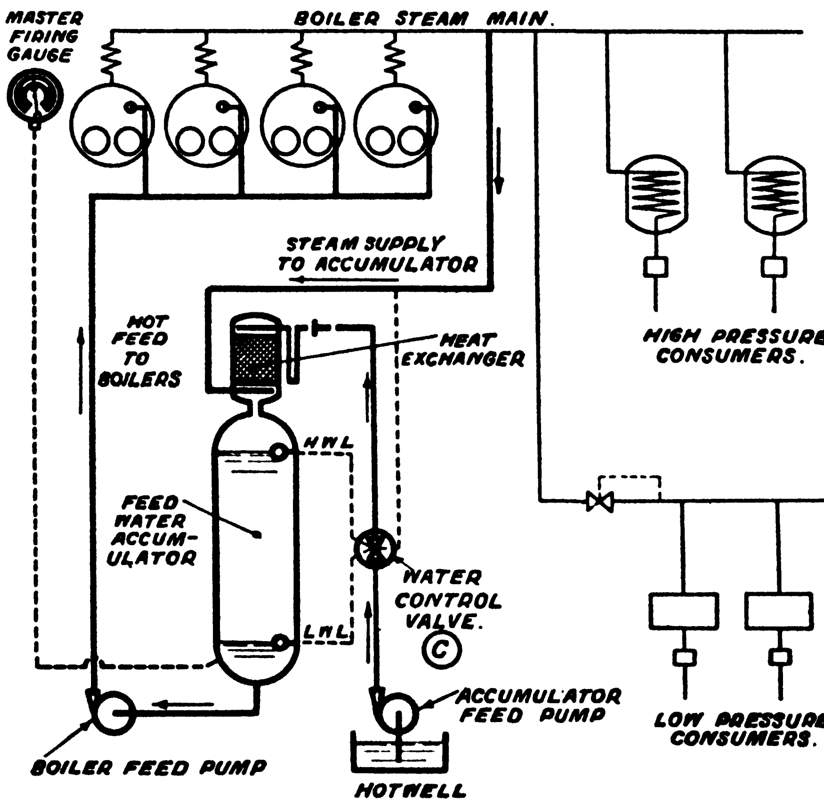


FIG. 95. Ruths accumulator system.

valve C into the direct-contact heat exchanger shown, where surplus steam is condensed. The hot water so formed passes into the storage vessel, from which it is drawn by the boiler feed-pump. It will be understood that the opening of the valve C, the rate at which steam is condensed in the heat exchanger, and the rise or fall of water in the storage vessel are dependent upon the general variation in the demand for steam in relation to the rate at which it is being generated. The feed-water accumulator is, in effect, a steam consumer, which, during peak-load periods, is automatically put off the line.

Fig. 96 shows typically how the Ruths Feed-Water Accumulator operates under the load conditions indicated. The boilers are fired at a constant rate, equivalent to the average steam demand, as shown by the lower line. With feed water at saturation temperature, the corresponding rate of steam production would be as indicated by the upper line. The shaded area represents periods when the accumulator charges by rise in water-level, while the intervening periods represent periods of discharge. As already indicated, the peak load

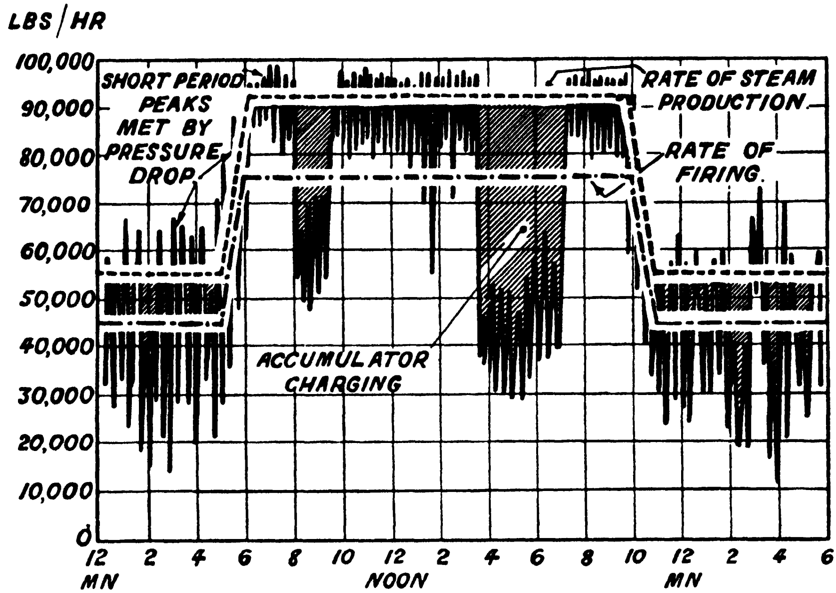


FIG. 96. Curve illustrating Ruths system.

carrying capacity of the feed-water accumulator is limited to the extent to which the feed-water temperature may be raised. Short sharp peak loads in excess of the capacity of the plant are met by ebullition under pressure-drop from the mass of water in the boilers and in the accumulator, as shown in Fig. 96. A master firing gauge in the boiler-house gives the firemen warning of any necessary change in the rate of firing, and the level of the water in the accumulator.

Hot Process Water Storage.

In certain industries, notably dye-works and laundries, hot-water storage at atmospheric pressure offers the most economical means of providing for peak demands. Fig. 97 shows a typical arrangement of plant, incorporating back-pressure power-generating equipment. Two surface heaters are arranged in series, one supplied with exhaust steam from the power-generating plant and the other with surplus live steam from the boiler main. Cold water enters the system through the circulating water pump shown, the rate of discharge of which is controlled by the thermostatic regulator E. This maintains a constant outlet water temperature by varying the rate of circulation of water through the heaters, dependent upon the amount of

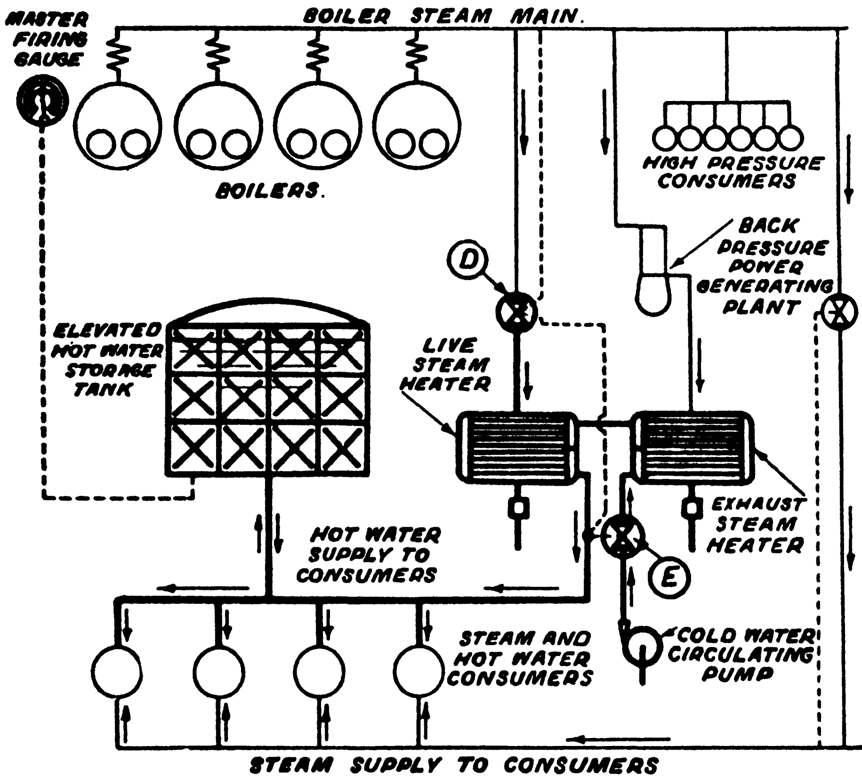


FIG. 97. Ruths accumulator system.

exhaust and surplus live steam available. Live steam enters the second heater through the overflow valve D, which is opened and closed by a pressure impulse taken from the boiler main. If at any moment the total steam demand is equal to the total amount of steam then being generated, the valve D remains closed, and only sufficient water is circulated through the heaters to condense the exhaust steam from the back-pressure power-generating plant. If, now, the general steam demand falls off, the boiler pressure tends to rise. This causes the opening of the valve D and, by temperature impulse, a corresponding opening of the valve E to admit sufficient cooling water to the heat-exchange system to condense all the live steam available. An elevated hot-water storage tank forms an essential part of the plant. The capacity of this is sufficient, by rise and fall of water-level, to cover all variations, both in the rate of demand for hot water and the rate at which it is generated.

Fig. 98 shows typically how the plant would operate with a total steam demand of the nature indicated. The boilers would be fired

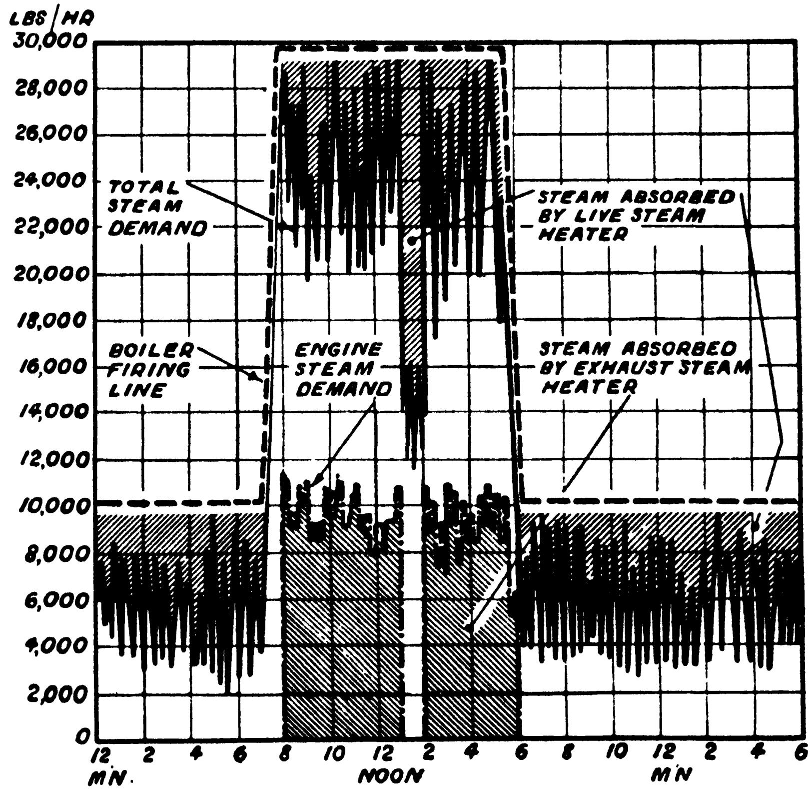


Fig. 98. Graph illustrating Ruths system.

in accordance with the dotted line, the amount of steam condensed in the exhaust and live-steam heaters being shown by the respective shaded areas.

In rolling mills large quantities of exhaust steam are usually discharged to the atmosphere, due to the fact that while this steam might be usefully employed for the generation of power, the time rate at which it is available varies over too wide a range. The exhaust-steam accumulator is similar in general principle to the live-steam accumulator except that it operates at nearly atmospheric pressure and over a very narrow pressure range, usually about 5 lb. per sq. in.

Fig. 99 shows typically how the accumulator is arranged. Exhaust steam leaving the rolling-mill engine is led to a header inside the shell and is distributed throughout the mass of water by means of nozzles and circulating pipes similar to those used in the live-steam accumulator except that the nozzles are designed with a view to keeping the back pressure on the engine as low as possible. Steam is

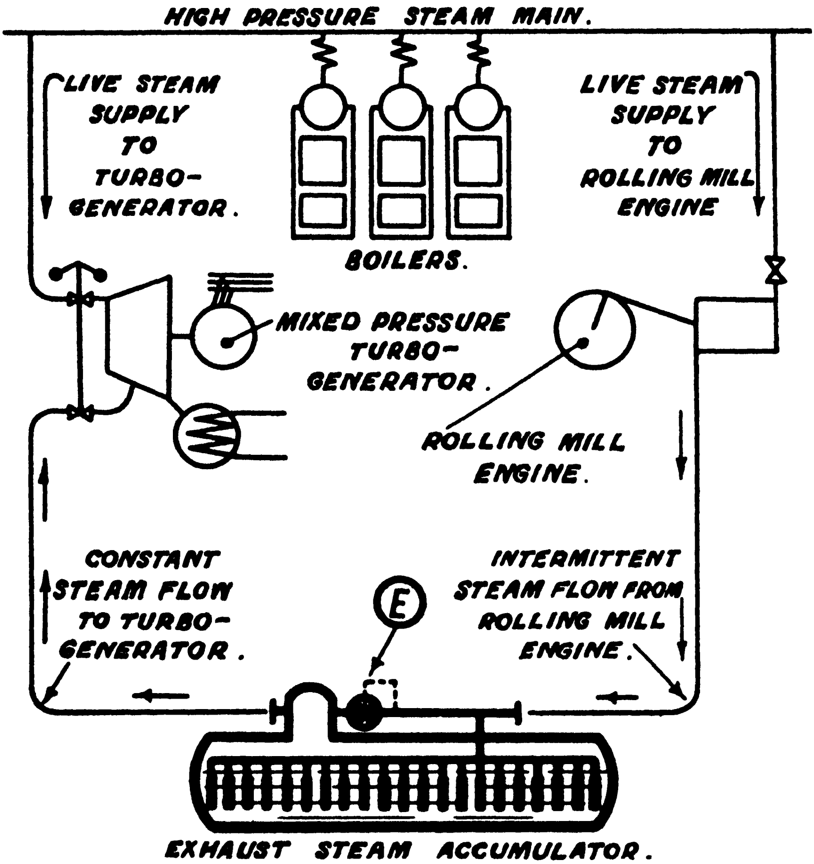


FIG. 99. Ruths accumulator system.

supplied at a constant rate to the mixed pressure turbo-generator from the accumulator, the valve and by-pass E being incorporated to bring the engine exhaust main into direct communication with the low-pressure steam-supply main when the accumulator is fully charged.

Fig. 100 shows typically how the accumulator is charged at a widely varying time rate and how it regenerates this steam at a constant time rate for the supply of the mixed-pressure turbine.

Fig. 101 shows how all three systems of thermal storage may be incorporated in a steel-works plant in order to obtain the maximum of productive and thermal economy. The live-steam accumulator fulfils the function already described, the overflow valve J and the reducing valve L maintaining a constant steam pressure on the high-pressure and low-pressure ranges respectively, while the exhaust-steam accumulator conserves steam which would otherwise be blown to the atmosphere. The boiler feed water is pre-heated by steam drawn from the live-steam accumulator through the thermostatic

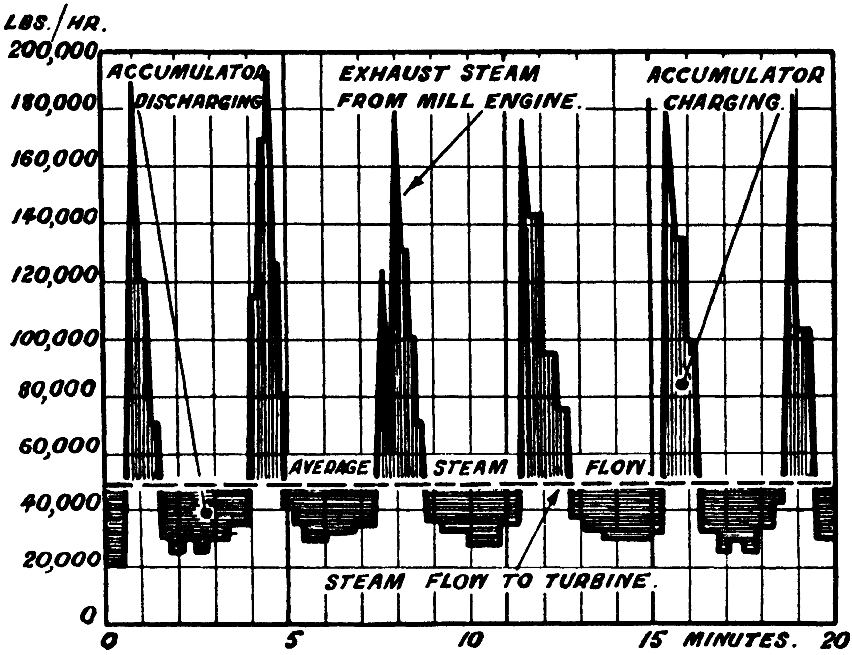


FIG. 100. Graph illustrating Ruths system.

control valve M, which maintains a constant temperature at the outlet from the feed-water heater to suit the rise in temperature through the economizer. The mixed-pressure turbine is supplied with live steam through the overflow valve K. If the steam pressure tends to drop, due, for example, to a temporary shortage of waste heat, this valve closes and steam is withdrawn from the accumulator through the non-return valve N. Below a certain pressure in the accumulator and to prevent slowing down of the turbine, the overflow valve K is automatically opened independent of the pressure on the boiler main.

The system described offers important possibilities where the bulk of the steam demand could be met by the use of waste heat boilers. Feeding such boilers with water substantially at saturation temperature greatly improves their steaming capacity for a given amount of waste heat available, while the use of the live-steam accumulator in

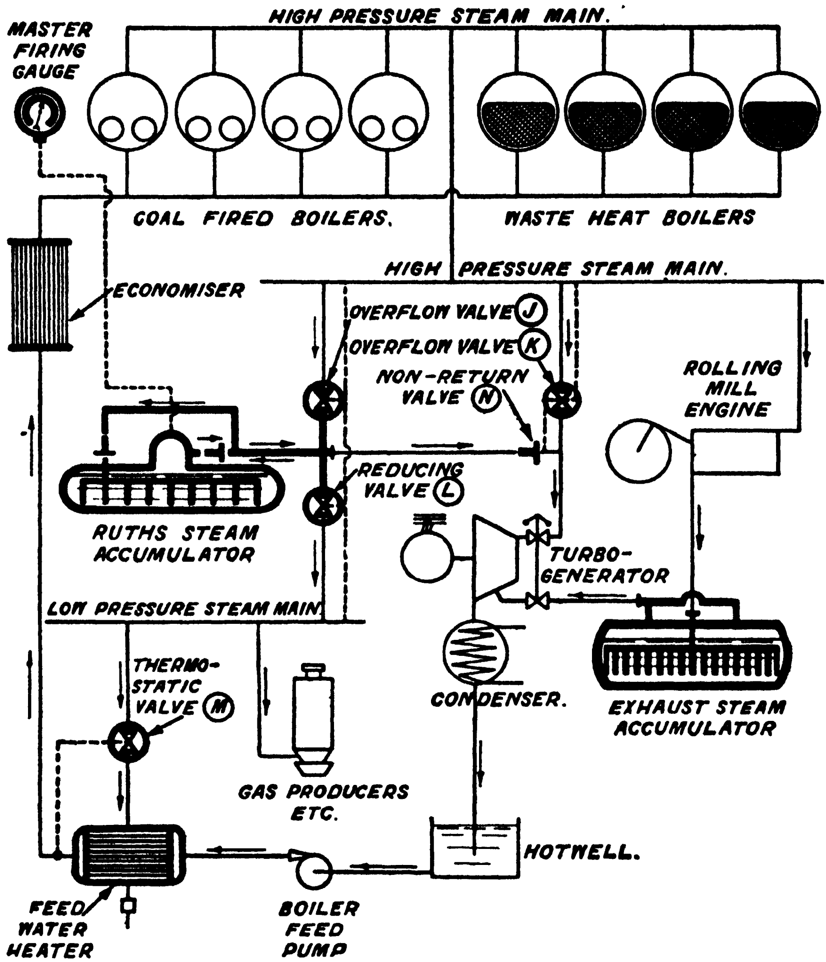


FIG. 101 A. Ruths accumulator system.

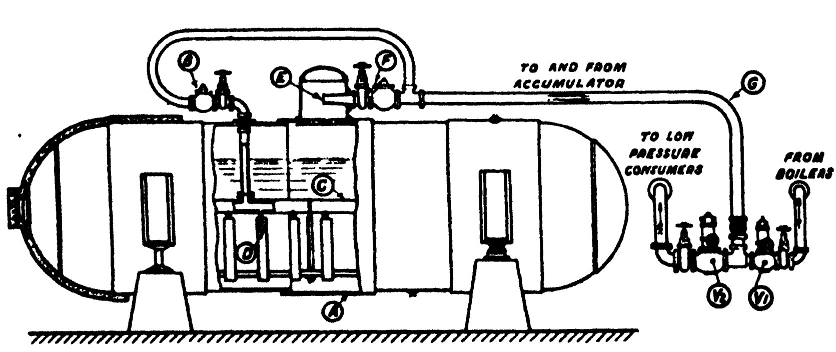


FIG. 101 B. Construction of Ruths accumulator.

the manner described, to supply steam to the turbine, relieves the boilers of a considerable amount of load during periods of shortage of waste heat. With the system described it should be possible in many steel-works to eliminate altogether the use of coal-fired boilers or to reduce considerably the amount of steam generated on solid fuel.

The Ruths Steam Accumulator consists of a riveted steel vessel A filled to about 90 per cent. of its total volume with water. Steam enters the accumulator during the charging process through the non-return valve B, the internal header C, and the nozzle group D. Steam leaves the accumulator during the discharging process through the dome and safety-nozzle E and the non-return valve F. Charging is effected by the condensation of steam in the accumulator and the consequent rise of pressure and temperature of the water contained in the shell. This stored thermal energy is recovered by ebullition from the mass of water and the regeneration of steam under a falling pressure.

Control of the process of charging and discharging is obtained by the overflow valve V_1 and the reducing valve V_2. If the boiler pressure tends to rise due to a falling off in the general demand for steam, the valve V_1 opens and all surplus steam then being generated is passed to the accumulator by way of the pipe G, where it is condensed and its thermal energy stored as already described. If a peak load develops on the high-pressure main, the valve V_1 closes, and during the continuance of this condition the steam demand of the low-pressure consumers is met by discharge of the accumulator. If a peak load develops on the low-pressure main, this is met by the wider opening of the valve V_2 and, if necessary, by the withdrawal of steam from storage to supplement the amount of steam then being obtained direct from the boilers through the valve V_1. The valves V_1 and V_2, therefore, automatically bring the accumulator into operation as a consumer of steam or as a producer of steam in dependence upon the general variation in demand.

Let us consider now the possibilities of these systems in connexion with modern hospital boiler-plants. In these buildings peak loads occur at definite periods each day, principally in connexion with hot-water supply demands for bathing, and in some cases generating plant is installed which provides heavy loads over known periods. The use of an accumulator system is likely to lead to the boiler-plant being of 60–70 per cent. of maximum capacity, and the boilers will be operating at full load with consequent saving in firing cost.

In one large hotel the problem was to equalize the load on the boiler-plant, allowing for maximum boiler efficiency and minimum boiler rating, and at the same time to provide sufficient storage of hot water to cope with the fluctuating demand for water for the hotel baths.

The plant is shown diagrammatically in Fig. 102, from which the nature of the steam-generating and steam-consuming plant will be noted.

Steam is generated in two 'Yarrow' boilers operating at 250 lb. per sq. in. pressure, supplying steam to back-pressure engines. Two of these engines were originally arranged to operate against a back-pressure of 25 lb. per sq. in., the third engine exhausting at 10 lb. per sq. in. pressure. The exhaust steam from these engines was used in the laundry and kitchen, in which departments, of course, there was a fluctuating demand for steam.

The steam pressure in the boiler-house was controlled by the first Ruths control valve. The steam pressure in the 25 lb. per sq. in. main was controlled by the second Ruths control valve. The steam pressure in the 10 lb. system was controlled by the third Ruths control valve. Each of these three valves operates normally as a 'surplus' or 'overflow' valve.

It will be seen from this arrangement that any surplus steam, either in the high-pressure main, intermediate-pressure main, or low-pressure main, will be allowed to flow over into the 'Kestner' water-heater, and be used there for the heating up of bath water.

Despite the fact that the power load and the demand for exhaust steam are both varying, the boiler load is constant.

The estimated storage of hot water for the baths was approximately 25,000 gallons. For convenience this capacity was split up between three tanks, one tank being in the basement and the other two being near the roof of the hotel. The operation of this part of the system is as follows:

During any period when there is a surplus of steam either directly from the boilers or indirectly from the steam engines, the steam flow through the 'Kestner' water-heater influences the cold-water supply valve which is thermostatically operated. This valve allows just sufficient water to flow through the water-heater irrespective of the rate of steam flow through the heater.

The outlet water temperature is approximately 140° F. This hot water flows into the main accumulator tank, in which the water-level will vary in direct proportion to the varying amount of surplus steam available. It is then pumped at a practically constant rate to the two tanks on the roof of the hotel, from which the hotel heating system and the baths system are supplied.

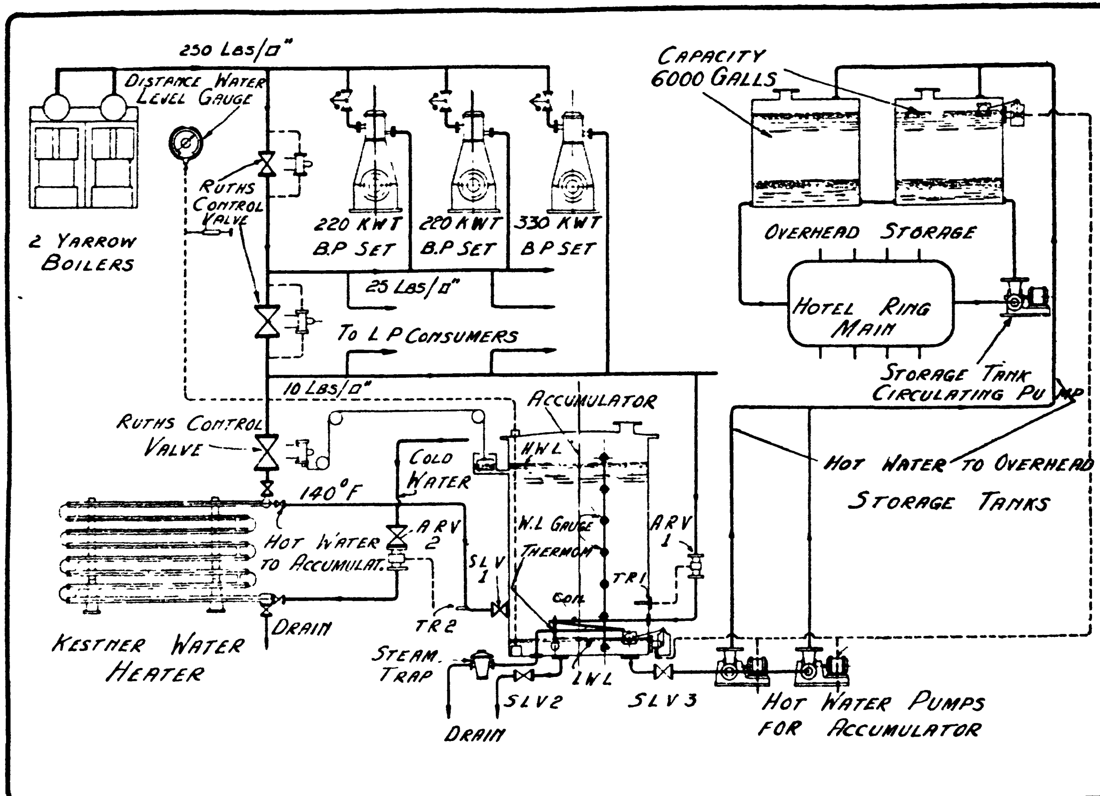


Fig. 102. Ruths system for large hotels.

In order to allow the accumulator system to operate satisfactorily under abnormal conditions the following safety devices have been added:

(*A*) *Maximum Water Control.*

In the event of surplus steam flowing through the heater over a long period when little or no water is being consumed in the hot-water system, the two roof tanks might tend to overflow. To prevent this a maximum water control, comprising a float-operated switch, will stop the hot-water pumps supplying these tanks. The accumulator tank in the basement will then fill up, and the distance water-level gauge in the boiler-house will indicate to the firemen that they are firing the boiler too heavily, and that a slight reduction in the rate of firing is necessary. Should no notice be taken of this indication, a maximum water control in this accumulator tank will automatically shut the third Ruths control valve, causing the steam flow through the 'Kestner' water-heater to cease. In this event the boiler pressure will rise, and the surplus steam will blow to atmosphere through the boiler safety-valves. This should not, however, occur, as ample warning would be given to the water-level gauge.

(*B*) *Minimum Water Control.*

During a long period when no surplus steam is available for water heating the hot-water pumps supplying the roof tanks might tend to empty the main accumulator tank.

To prevent air being sucked into these pumps a minimum water control comes into action. This is a float-operated switch which will stop the hot-water pumps until such time as the water-level in the main accumulator tank has risen above the low-water limit. In the meantime the hotel is drawing upon the reserve from the two upper tanks.

(*C*) *Auxiliary Temperature Control.*

During the night when the boiler-plant is shut down it is necessary to provide sufficient heat to the main accumulator tank which during this period should be on full. A copper coil is introduced into this tank, and the steam flow to it is controlled by a thermostatic valve. It was found that there was a sufficient reserve of steam in the boilers, under banked conditions, to maintain the desired temperature of 140° F. in this accumulator.

The system would appear to be rather complicated, but in principle it is extremely simple, and the complications are due to difficulties which had to be overcome at site. The entire system is found in practice to be most satisfactory. It is the only system in operation in this country where the variations in boiler load are levelled out by reflecting them on the water-heating system, and at the same time providing the latter system with an adequate supply of hot water at an automatically-controlled temperature.

Load Balancing with Hot-Water Boilers.

Hitherto we have only considered the advisability of averaging boiler-plant where steam boilers are concerned, but there are advantages to be gained where hot-water boilers are used. With large

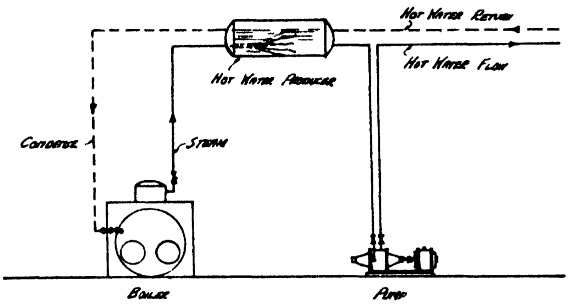


FIG. 103. 'Krantz' system.

bank buildings, for instance, where steam is not required for any service, the installation of hot-water boilers has fundamental advantages. For hot-water boiler-plants the accumulator effect may be obtained by installing boilers to deal with average load and providing hot-water storage vessels to accumulate sufficient hot water at high temperature during periods of low demand to cover the peak-load requirements. Where oil firing or automatic stokers are used a considerably higher efficiency is obtained by having the firing system in continuous operation, the heat not immediately required being conserved in the accumulator vessels.

Krantz High-Pressure Hot-Water Systems.

Of a somewhat similar principle to the thermal storage systems which have been considered is the Krantz high-pressure hot-water heating system† which is of particular use where it is required to produce hot water and steam from one boiler. Fig. 103 illustrates one application of this system where steam is taken from the boiler to a hot-water producer consisting of a cylindrical vessel half-filled with water, the upper half containing steam. The steam is injected into the water, resulting in the water reaching the same temperature

† See *The Steam Engineer*, June 1933, and April 1935.

as the steam. The heated water is then pumped through the heating system and finally returned again to the hot-water producer for re-heating. The steam which is condensed in the producer passes through an overflow and returns directly to the boiler by gravity.

Another example in Fig. 104 has been applied where there was an existing boiler working at an existing pressure of 180 lb. per sq. in. for operating a steam engine. The heating-pipe system was not suit-

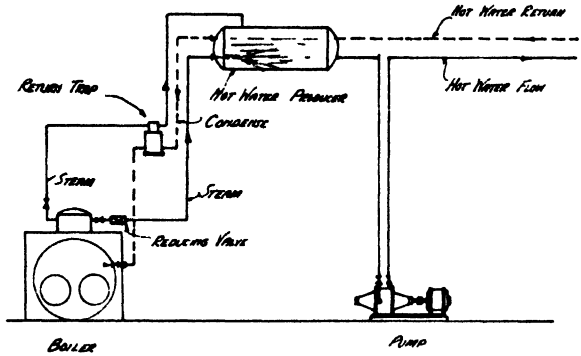


FIG. 104. 'Krantz' system.

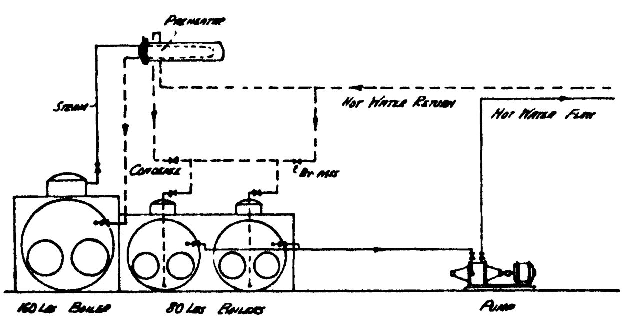


FIG. 105. 'Krantz' system.

able for a pressure higher than 100 lb. per sq. in. The method first described was not feasible for this system owing to the difficulties of returning condensation, but the application of a return trap of a special type overcame these difficulties.

A further application shown in Fig. 105 is for three boilers, one with a pressure of 160 lb. per sq. in. for operating a steam engine and the other two operating at 80 lb. per sq. in. to provide process steam and for the heating system.

At one time any additional steam required for peak loads was taken from the high-pressure boiler. For converting to hot-water heating a calorifier was employed above the engine boiler, supplying steam at 160 lb. per sq. in., the hot water returning through this heater to the boiler by gravitation. The electrically operated thermostatic control regulates the admission of steam to the calorifier to suit the requirements of the hot-water circulation.

The Treatment of Flue Gases.

Modern boiler plants demand consideration to be given to cleaning flue gases before they are discharged to the atmosphere, especially in connexion with power plants or combined power and heating plants. Particular attention was given to this point in the case of the Battersea Power Station, an experimental plant being erected to determine the efficiency of flue-gas scrubbing in removing sulphur fumes, as a result of which it was determined that spraying with water would remove 95 per cent. of the sulphur fumes.

Finally, an elaborate arrangement of scrubbers and grit-catching apparatus was designed and installed.

It is likely that similar arrangements will be applied to all large boiler plants in the future in industrial as well as residential areas.

The London County Council recently considered a long report from the Housing and Public Health Committee on the subject of the Battersea generating station of the London Power Company. The immediate occasion was a letter from the Electricity Commissioners inviting the observations of the Council on an application from the company to install three additional boilers, and thus complete the first half of the station. Opportunity was, however, also taken to review the whole of the controversy which has centred round this plant. It was agreed, in a conference with the representatives of the Battersea, Chelsea, Kensington, and Westminster Borough Councils, that it would be difficult to oppose the proposed extension on the grounds of danger to public health. At the same time, the opinion was unanimously expressed that, in order to ensure that the gas-washing plant was maintained at its present high degree of efficiency, and that advantage was taken of further developments in sulphur removal, the Commissioners should be asked to attach more definite conditions to the extension of the station than apply to the existing plant. This has already been done at the new station at Fulham, where there is a stipulation that the most efficient methods available shall be used continuously for the elimination of smoke and grit, for the prevention of discharge of sulphur and its compounds into the atmosphere, and for the avoidance of noise and vibration. It is also laid down that the Commissioners may require the installation of more efficient arrangements than those already in use, and that any such equipment shall be maintained and operated to their satisfaction. Apparatus for the continuous measurement of sulphur emission is to be provided, and the station is to be open to inspection at any time. It was therefore recommended that consent should only be given to the Battersea extensions on these conditions, that such consent should apply only to the three particular units in question, and that the local authorities concerned should be informed when any further extensions are contemplated.

W. J. Crampton has pointed out, in direct contrast to such opinions, that facts disagree with the tendency to regard the emission of sulphur fumes from power-station chimneys as highly detrimental to life and property in large cities. The usual method of presentation involves the calculation of the tons of sulphuric acid produced per day from the combustion of fuel in a large city or power station. General experience indicates that things are not so black as they are painted, and the following facts are of interest when looking into this matter.

The seventeenth report of the Atmospheric Pollution Committee states that on clear or merely misty days the whole of the acidity appears to be due to SO_2, sulphuric acid only appearing during fog. This unexpected result shows that sulphuric acid is not a normal constituent of the atmosphere.

Even in highly industrial centres such as Pittsburgh, U.S.A., which has three times the soot fall per square mile as compared with London, tests made on 304 days showed an average of only 0·16 part SO_2 per million, with maximum on 2 days of 1·1 parts per million, and less than 0·1 part per million on 141 days. Although this is an exceptionally bad centre, the SO_2 concentration on the worst days is far below the danger-point to human beings. Henderson and Haggard give 10 parts per million as the maximum allowable concentration for prolonged exposure. In fact, it is important to note that whilst abnormally high concentrations of sulphur gases are injurious, in the average concentration experienced sulphurous acids are said by some authorities to be beneficial.

Rideal states that light, ozone, and hydrogen peroxide in the country, and acids in town air, are concomitant causes of rapid destruction of disease germs in the air and that sulphurous acid has the most destructive effect on aerial microbes, especially moist. He quoted M. d'Abaddée as follows: of Sicilian labourers engaged in sulphur works only 8 per cent. to 9 per cent. suffer from intermittent fever as against 90 per cent. of those not so engaged. The sulphur works in the marshy plain of Catania protect the people in the vicinity from an evil that causes other villages to be deserted.

It seems most probable, therefore, that if the mild concentration of sulphurous acid in the air of heavily populated towns were reduced, other evils might follow, and as already shown we are a long way off the danger point.

As regards the effects of sulphurous emissions on property, this again is greatly exaggerated. The buildings in London to which attention has been drawn are many of them of considerable age, and one cannot expect that general weathering will be absent. The Mansion House, for example, is said to be now having its first wash for 200 years.

The question of the effect of sulphur on vegetation has also been raised, but the chief cause of trouble in towns is the tarry matter from domestic chimneys which deposits on the foliage, thus stopping up the breathing mechanism. Dust and dirt are evils which need no argument to emphasize them, and in the case of power-station emission these can be adequately dealt with by electrostatic precipitation or other grit-collecting device.

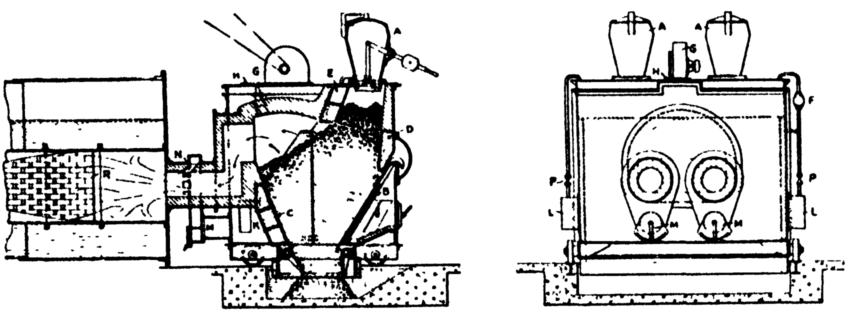
The Wollaston Producer Furnace.

Where quantities of waste combustible material or low-grade fuel are available, there is no advantage to be obtained by gasifying this in the normal producer gas plant and then delivering it to burners in the boiler plant.

In such cases a producer furnace of the Wollaston type may be usefully employed, combining the functions of the normal gas producer and automatically stoked boilers.

The furnace, illustrated in Fig. 106 in section, is a development of the Siemens principle provided with a water-sealed mouth at the bottom for the removal of ashes. The fuel, in this instance, coke breeze, is fed into the producer by means of two charging hoppers A, A, and forms a deep bed which is partly supported on the inclined grate B and refractory wall C. The lower ends of these parts terminate in an opening through which the residual ash passes into the water in the ashpit. The whole furnace is carried on runners D, D, which enable it to be removed away from the boiler when necessary, and to facilitate this movement a pivoted flap door E is provided to support the ash temporarily until the producer is moved again into its working position. Two water chambers, F and G, are provided across the whole width of the furnace, and are interconnected by the circulation pipes H, H at each end.

The upper L-shaped chamber forms the roof of the furnace, and the water therein is automatically maintained at a fixed level by means of a feed regulator K, thus providing a steam space. Low-pressure steam, at a blow-off pressure of 5 lb. per sq. in., is generated in these two chambers, and is used to saturate the primary air supply to the space below the grate. For this purpose the fan P is provided. It has a main regulating damper R, which controls the total air supply. The air passes into the back chamber and finds its way via the opening and the passages to the space below the grate. A portion of the air controlled by rotary valves is tapped off and used as secondary air to the two burners. Levers are provided for controlling the burners independently.



Wollaston PRODUCER FURNACE — HORIZONTAL FLAME TYPE

SCALE OF FEET

FIG. 106.

As in normal producer practice, the primary air is restricted so that the carbon in the fuel is partially burnt to carbon monoxide, any carbon dioxide formed being reduced to carbon monoxide in passing through the fuel bed. The steam supplied by the low-pressure boiler is added to the air to reduce the temperature and prevent fusing of the ash. Some of the steam, in conjunction with the incandescent fuel, is split up, giving hydrogen and carbon monoxide.

The gas passes through the central outlet M and divides into two streams, one to each burner. These burners are provided with a series of air ports which supply the secondary air. The gas, when ignited, forms a large flame of the Bunsen type, which impinges upon the perforated refractory brickwork walls built within the boiler flues, and causes them to become incandescent.

It is stated that the overall efficiency of the plant is 75–80 per cent. Although the illustration shows a fixed grate, which is satisfactory when coke, good breeze, or non-caking coal is used, it is usual to have a mechanical grate, either for power or hand operation, for the poorer quality fuels or those having low fusibility or high ash proportions.

Fig. 107 illustrates the arrangement of these producer furnaces as applied to the boilers of large hospital heating systems.

Chimneys for Boiler Plants.

There have been published a number of tables and charts for the determination of chimney and flue dimensions, and the widely differing results obtained by their use has led the author to produce a nomograph which facilitates the solving of chimney problems. A point which designers often fail to appreciate is that no one graph can give the chimney sizes necessary for all cases, and this has caused

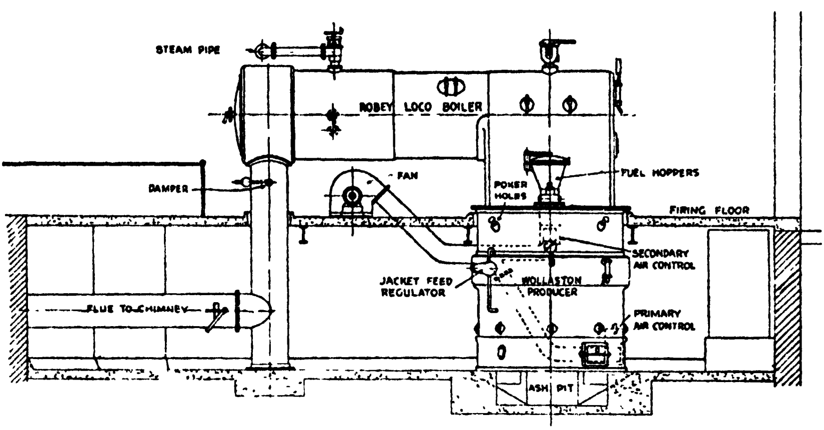


FIG. 107. Typical Wollaston producer plant.

chimneys to be used which are either too small or too large for the fuel in question. With gaseous fuels the amount of flue gases may be only one-third of those for solid fuel, whilst oil fuel produces less also.

In using the chimney nomograph, Fig. 108, the movable scale should firstly be set to the type of fuel being used, when the intersection of a line through 'B.T.U.' and 'height', with the 'area' scale, indicates the necessary cross-sectional area for the boiler rating in question. An addition of 10 per cent. to the area is usually required for an exposed steel chimney.

Several other details need attention where gaseous fuel is used, and the troubles usually experienced are either those from fumes blowing back due to bad draught, or the enormous amount of water produced during combustion. The products of combustion should leave at as low temperature as possible, so that usually the draught available is very weak, and is easily overcome by adverse winds.

With gas fuels, the amount of moisture produced is 6 gallons per 1,000 cu. ft. of gas, so that the material of which the chimney and flues are constructed needs careful choice.

Fortunately, the whole of this moisture is not condensed in the chimney, but sufficient to warrant taking notice of the effect. A material is required which is not only resistant to heat, but also to the effect of condensed moisture, which is slightly acid, and for this reason a brick chimney is not suitable unless it is carefully lined. Practical tests have shown that the materials most resistant to corro-

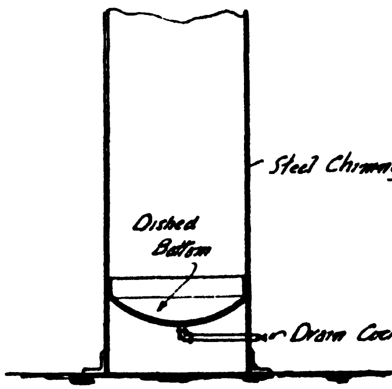


FIG. 109. Flue for gas boiler.

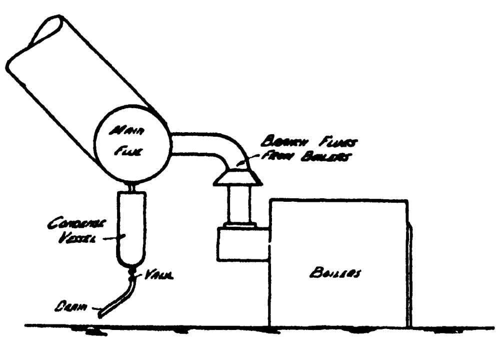


FIG. 110. Flue connexion to gas boiler.

sion are asbestos cement, aluminium, or glazed earthenware. Some of the asbestos cement products are advantageous in preventing condensation, are easy to cut, and are not too fragile.

Condensed moisture should be easily removable at the base of the flue or chimney, and on no account should it be allowed to fall back into the boiler, where it would either be re-evaporated, or remain to corrode the metal walls of the boiler. One arrangement in general use is the provision of a collecting vessel with drain cock, shown in Fig. 109, whilst another is that in Fig. 110, where a condensation tray is provided at the base of the chimney, with a drain pipe running to a convenient position.

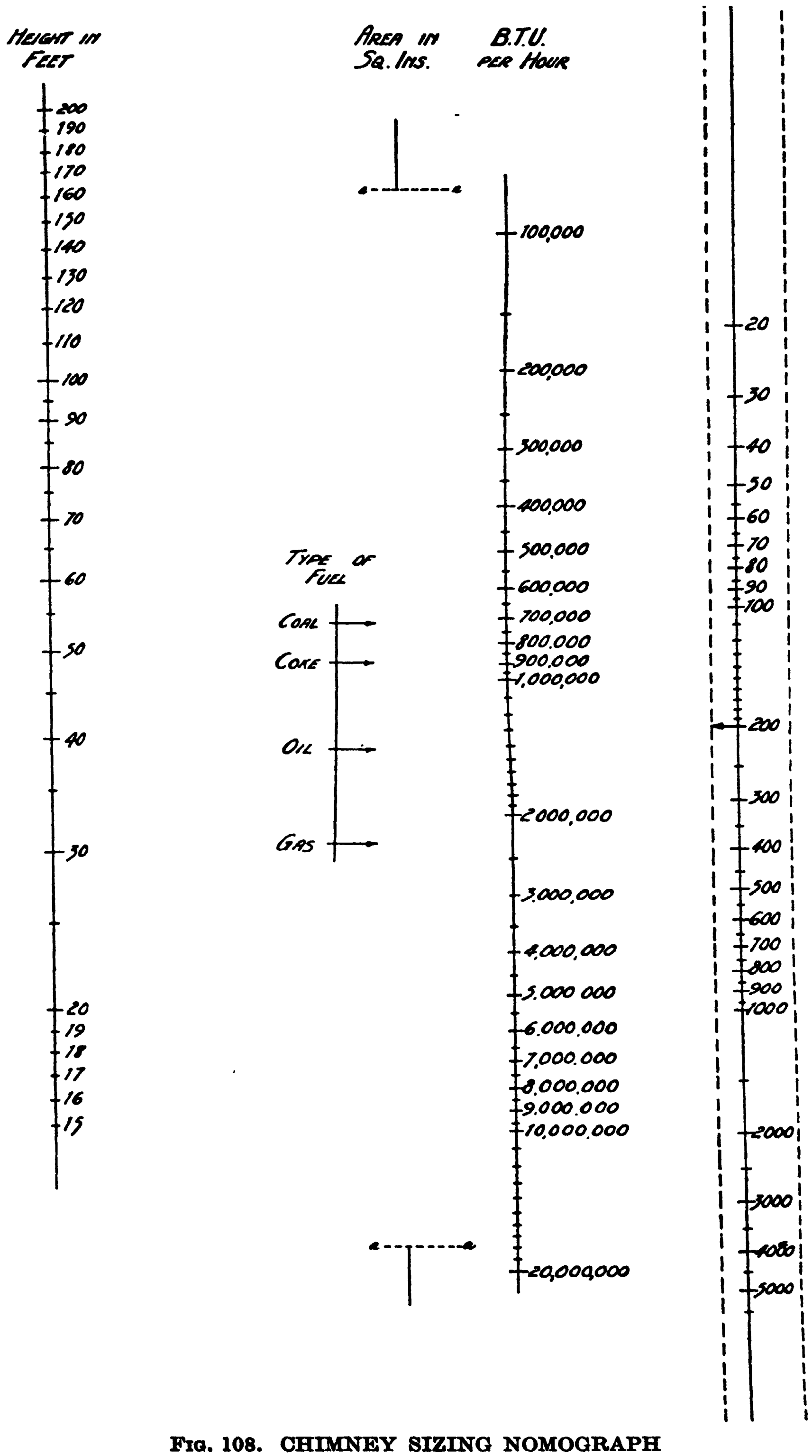


FIG. 108. CHIMNEY SIZING NOMOGRAPH

When boilers are fired by high-pressure gas or air systems, the pressure produced by the burners results in a strong draught in excess of that due to temperature differences, but the chimney should not be reduced in size below that obtained from Fig. 108. Similarly, it should always be borne in mind that there may be a remote possibility of the boilers being fired with solid fuel at some future time, which may make it desirable to have the chimney large enough for that type of fuel in the first instance.

It is a doubtful point whether it is necessary to have a break in the boiler flue connexion with a down-draught preventing cowl. For high-pressure gas firing where gas is supplied at the burners at 10 in W.G., there is little risk of down-draught, but in all cases the break in the flue certainly has the advantage of introducing a quantity of air to mix with the products of combustion, thereby lessening the risk of condensation occurring.

The engineer is concerned in the principles of the structural design of chimneys apart from the calculation of area and height. For steel chimneys,† the design may either be for the self-supporting type or alternatively for use with stays or guys. The self-supporting steel flue differs so little from the other that there is every justification for using it. The fundamental principle of design with the self-supporting steel chimney is that the weight of the stack and the resistance to movement of the foundation together with the rigidity of support on the base must be sufficient to resist the tendency to overturning exerted by wind pressure. Generally, the self-supporting chimney is not more than 25 diameters in height, as unless thick plates are used, a greater height produces excessive stresses. The base, in order to distribute the weight and resist overturning must have a taper or flare, tapering to $1\frac{1}{2}$ times the diameter, the taper joining the stack at about $\frac{1}{8}$th of the height from the bottom. In passing, it may be mentioned that although self-supporting chimneys have been built as high as 400 ft., the diameter at the top being 31 ft., it is unusual to find them more than 250 ft. high.

In working out stresses for steel chimneys, the wind effect is taken as 40 lb. per sq. ft. of exposed area, that is, for half the total area of the chimney.

The plate thickness for the conical base is usually no more than $\frac{3}{8}$ in. or $\frac{1}{2}$ in., and if openings for connexions are cut into this reinforcement is required, particularly if this should come on the leeward side where the wind effect is acting downwards on the base. Where firebrick linings are used they are divided into 20-ft. lengths, each

† See 'Steel Chimneys', by F. Johnstone Taylor, *The Power Engineer*, Feb. 1924.

supported by internal angles, with a 1-in. gap between firebrick and metal, filled with loam. The concrete foundations must be of sufficient mass to resist overturning, and at the same time cover an adequate area according to the bearing pressure of the ground:

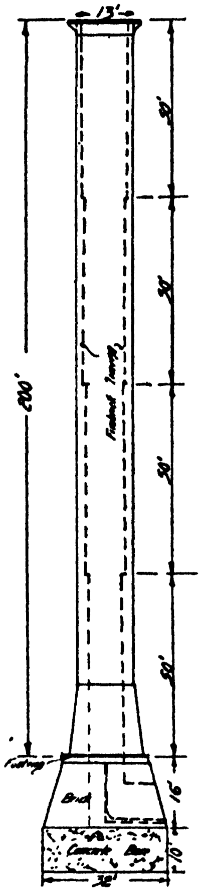


FIG. 111. Typical steel chimney.

Taking as an example the chimney in Fig. 111 we may calculate the wind pressure as follows:

Height $\times \frac{1}{2}$ circumference $\times$ 40 lb. per sq. ft.

$$= 200 \times 20{\cdot}4 \times 40$$

$$= 160{,}000 \text{ lb. approximately.}$$

The chimney is considered theoretically to be a cantilever with the bending moment due to wind pressure greatest at the base, so that this becomes:

$$\frac{WL}{2} = \frac{160{,}000 \times 200}{2}$$

$$= 16 \text{ million lb. ft.}$$

Operating against this, we have a concrete base 32 ft. square by 10 ft. thick, which at 140 lb. per cu. ft. weighs

$$32 \times 32 \times 10 \times 140 = 1{,}400{,}000 \text{ lb.}$$

This again, considered as a cantilever with a concentrated loading measured from chimney base as 21 ft., would have a moment of:

$$1{,}400{,}000 \times 21 = 29{\cdot}4 \text{ million lb. ft.}$$

Theoretically, if the resistance to turning of the base, taking into account also the weight of the chimney, is equal to the bending moment, the chimney would not be stable. The allowance of 40 lb. per sq. ft. as the effect of wind pressure is equivalent, however, to a wind velocity of 165 miles per hour (for a round chimney), which represents ample safety factor.

It should be realized also that the wind pressure is converted to a tension in the bolts on one side of the chimney, holding it to its foundation, and the sizes of these should be carefully calculated with a large factor of safety, for upon them the whole structure depends.

Ejector Chimney Draught.

The use of the ejector draught system, at one time limited to steam blast in the chimney, is now prevalent, the draught being produced by a small high-pressure blower discharging into a Venturi-shape chimney. This system allows the use of short chimneys.

The basis of design is the calculation of the volume of flue gases to be handled by the chimney, obtained from the quantity of fuel burnt and the air required for combustion. The ejector nozzle and venturi are then designed in accordance with the calculations in Chapter VIII.

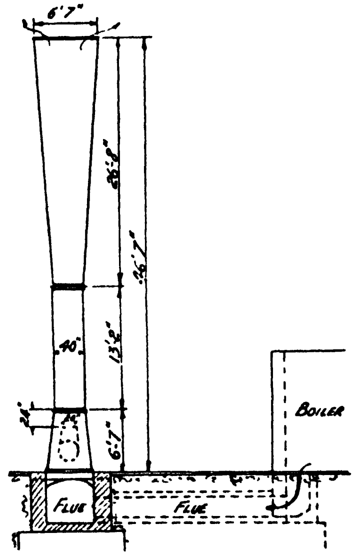


FIG. 112. Ejector draught for boilers.

Fig. 112 illustrates an arrangement of ejector draught for a chimney serving down-draught boilers.†

In this case, the chimney serves 6 boilers, each of 32·5 sq. ft. grate area, burning at the rate of 21 lb. of low-grade coal per sq. ft. The total draught required to overcome fuel bed resistances and the various losses was 1 in. W.G. The quantity of flue gases was taken for the particular fuel as 470 cu. ft. per lb. The total duty of the chimney was therefore 3,150 cu ft. per minute.

The draught was produced by a nozzle 20 in. diameter discharging into a 40-in. throat, at the rate of 1,800 cu. ft. per minute, and at a pressure of 3 in W.G. The power absorbed by the blower was 12 H.P. The calculations did not take into effect the gains from natural draught.

For natural draught a chimney at least 200 ft. high, and probably 10 ft. diameter would have been required, whereas the ejector draught system is not only permanent and unalterable by weather conditions

† 'Des trompes et de leurs applications industrielles', *Arts et Métiers*, Oct. 1927.

but the cost of the whole equipment is only about £250, which is about one-third of that for the natural draught chimney.

The provision of an alternative drive for electricity or steam is essential.

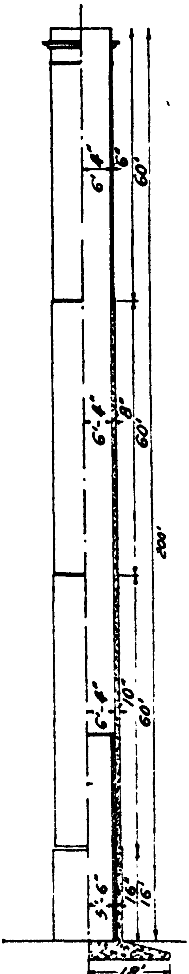


FIG. 113. Reinforced concrete chimney.

Brick and Concrete Chimneys.

The brick chimney-stack is perhaps the most popular in this country for any but industrial buildings, although some large bank buildings have internal steel flues cased in brick.

Whilst the proportions of brick stacks are perhaps standardized, it is well to calculate for unusual cases the minimum thickness of brickwork which will resist stresses due to wind pressure.

Reinforced concrete is a construction which will doubtless be used considerably in the future owing to the fact that for chimneys over 120 ft. high they are cheaper to construct.† Incidentally the maintenance cost is negligible, for they require no painting or banding in future years as brick chimneys often do.

The reinforced concrete chimney shown in Fig. 113 is interesting for comparison with the steel chimneys in previous illustrations.

The reinforcement is arranged as horizontal rings tied to vertical rods, the lower rings being calculated to resist compression due to the weight of the concrete above, and the vertical rods on the windward side being in tension and those on the leeward in compression.

The base or foundation needs only to be 18 in. thick, owing to the far greater weight compared with a steel chimney, and consists of a slab, thickened in the centre to resist the concentrated load of the chimney, with radiating reinforcing rods running to the edges. It is desirable that the first 40 ft. should be firebrick lined.

It is suggested that a factor of safety of 5 should be used, with 4 : 2 : 1 concrete for foundations and a richer mixture for the shaft, with ample proportions to the foundations.

† See 'Reinforced concrete for chimneys', by J. M. Jardine, *Indian and Eastern Engineer*, Oct. 1927.

Chapter Eight

DUST AND FUME EXTRACTION, STEAM REMOVAL, AND PNEUMATIC CONVEYING

Steam Removal.

MANY of the accidents which occur in textile mills, dyeworks, and paper-mills have in the past been caused by the obscured field of vision which the operatives have owing to large quantities of free steam and vapour being present in the air of the workshop. There are many means by which the difficulties of steam being present may be overcome, and wherever possible care is taken to remove the steam at its source.

With fruit-boiling pans for example steam could be removed through an exhaust hood fitted above and fairly close to the pans. A steam-removal system of this nature presents no difficulties as it only becomes a simple extract system.

There are other instances, however, where the whole room is on occasions filled with steam, and in these cases if air is merely exhausted from the room fresh air must enter from somewhere, usually at a lower temperature. As is well known, the capacity of air for absorbing moisture decreases with the temperature so that instead of helping to eliminate free steam, extraction generally creates a worse condition. It is apparent therefore that the basis of removing or eliminating free steam in any room must be to increase the capacity of the room for absorbing moisture. It might perhaps be thought possible to increase the temperature of the room alone, but although this naturally leads to an increase of the possibility of moisture being absorbed, saturation of the atmosphere would soon be reached with very little improvement in conditions. It is necessary that air in quantity should be moved through the room and this air must be as free from moisture when it is introduced as is possible, in order that it will be able to absorb the large quantities of moisture which free steam provides. There are two general means which may be employed for introducing air to the room for absorbing moisture:

(1) To introduce small volumes of very hot air;

(2) To introduce large volumes of air very little above the usual room temperature.

The first method is often useful where the distribution of the air can be localized, but in most cases the second method is often more practicable.

Let us assume as an example a room in which it is known that 200 lb. of free steam is provided by process work every hour, whilst the capacity of the room is to be assumed as 150,000 cu. ft.

It will be decided that to introduce air at 70° F. will normally overcome the problem provided that the air is reasonably dry. During the summer months, if it is found that the natural conditions do not eliminate steam, air will probably be available at a far lower temperature, but as a basis of calculation it may safely be assumed as the basis that the relative humidity of the entering air will not be greater than 30 per cent. From the psychrometric nomograph we may see that in this condition the air contains 33 grains of moisture per lb. It is preferable, and as a basis of calculation it would be assumed, that the air leaving the room after absorbing free steam should not have relative humidity greater than 65 per cent. At the condition of 70° F. with 65 per cent. relative humidity air would contain 70 grains of moisture per lb. It must, therefore, be apparent that the amount of moisture which would be added for each lb. of air would be $70-33 = 37$ grains. The total quantity of air required to absorb 200 lb. of steam, knowing that 1 lb. is equivalent to 7,000 grains, will be $\frac{200 \times 7,000}{37} = 37,800$ lb. of air per hour. As there are approximately 13·5 cu. ft. of air per lb. at 70° F. this is equivalent to $37,800 \times 13{\cdot}5 = 510,000$ cu. ft. per hour. As the capacity of the room is 150,000 cu. ft. the number of times which the air would need to be changed per hour would be $\frac{510,000}{150,000} = 3{\cdot}4$ times per hour. This basis of calculation would apply for any ventilation system applied with the object of eliminating steam, provided the amount of steam produced is known, but it is generally found that to provide 4 to 5 air changes with air taken from outside the building and raised to 70° F. overcomes steam difficulty.

In arranging for the distribution of the warm air it is desirable that it should be introduced to the room as near the positions in which the steam is produced as possible.

The use of unit heaters comprising self-contained heater and fan unit with a connexion to the outside air is now becoming quite common for steam removal systems.

Basis of Calculation for Fume Removal.

Many industries, particularly those like artificial silk, lead to the production of large quantities of fumes in the workshops which must be eliminated or at least diluted by the provision of large quantities of fresh air. It is generally possible to calculate from a knowledge of the chemical process involved the quantity and nature of the fumes produced, and from a knowledge of the maximum quantity which may be present in the air for health or comfort to calculate the amount of air required to dilute them to this level. It is not always possible, however, to obtain this information, and then experimental figures or the results of other installations must be taken as a guide, for according to the conditions so the air changes required might vary between 10 and 30 times per hour. As a general basis of calculation it is, therefore, necessary to know the following:

(1) The weight of fumes produced by the process;

(2) The maximum permissible weight of fumes which may be allowed in 1 lb. of air in the room.

For an example let us consider a room in which 20 lb. of sulphur dioxide is produced per hour. If it is known that the maximum permissible amount of this gas which may be allowed is 0·05 per cent., then the quantity of air which must be passed through the room in a suitable manner must be: $\frac{20 \times 100}{0 \cdot 05} = 40{,}000$ lb.; at 70° F., this would be equivalent to $\frac{40{,}000 \times 13 \cdot 5}{60} = 9{,}000$ cu. ft. per minute. We will deal later in this chapter with the specific requirements of fume removal systems as far as design and layout are concerned.

Fume Extraction in Various Industries.

Most important industries have many instances where fume-extraction systems are required, and of these the artificial silk industry is the one which offers more scope than any other for the application of fume-extraction systems. According to the type of silk which is being manufactured so the fume extraction problems will differ. In the case of viscous silk, which is the most common, sulphuric acid is thrown off from the spinning machines as a vapour into the air of the spinning-room. Another type of silk known as cuprammonium releases ammonia during manufacture, acetate silk eliminates acetic acid, whilst the nitrose cellulose or chardonne silk produces hydrochloric and sulphuric acids. It is not legally permissible for artificial silk spinning-rooms to be without a fume-extraction system, and for that matter it is impossible for operatives to work without it, for the fumes not only result in frequent sickness but there have been many instances where operatives have been blinded for long periods by fumes. Opinions differ upon the quantity of air which must be exhausted to produce conditions which are allowable for the operatives. The quantity of fumes produced varies according to the type of machine employed for spinning and also the denier or thickness of the thread which is being spun.

Referring now to the viscous silk, this is manufactured by discharging a liquid through fine nozzles or jets into a bath of sulphuric acid, the thread being solidified in the sulphuric acid bath and finally being carried to the upper part of the machines where it is wound on bobbins somewhat similar to cotton spinning. It is in passing over the guides to the bobbins that the sulphuric acid is eliminated. The quantity of air to be exhausted may be defined as a quantity per spindle or bobbin on the machine.

The following figures give the quantities employed by many of the spinning-machine manufacturers:

Parallel bobbin spinning 30 cu. ft. per minute, per spindle.

Centrifugal pot spinning 50 ,, ,, ,,

If coarse deniers are worked 70 cu ft. per minute, if fine deniers are worked 40 cu. ft. per minute. In this connexion it is mentioned that the average denier for artificial silk is 150.

An artificial silk spinning-mill consists of a large room generally of north light construction with spinning-machines running across the width of the room, and these machines might perhaps each contain 100 spindles. In the design of the fume-extraction system the first thought must be to collect the fumes from as near their source as possible, that is actually from the machines. It is not satisfactory with most fume-removing problems to attempt general extraction from the room. There are many alternative methods which have been employed for dealing with the extraction from machines, some of which will be described.

One particular system, illustrated in Fig. 114, employs an extraction hood fitted to the top of the spinning-machine for its full length, communicating by ducts from each machine to a centrifugal fan; the fans discharge into a common duct which in turn leads to a common fume-duct outside the spinning-room and thence to a chimney discharging the fumes at high level.

This method has many points in its favour, amongst which the most important is the fact that it is possible to run one or a small group of machines without the consumption of power which is normally taking place for a centralized fan system. On the other hand, the multiplicity of fans with this system generally proves expensive to install. It will be observed that this particular system serves groups of machines running lengthwise in the spinning-room.

As an alternative arrangement, if the machines had been running with their length across the spinning-room the system shown in Fig. 115 could be used, consisting of a hood over each machine with a duct above it communicating with a separate fan for each machine which discharges to a common suction discharge duct leading to a fume chimney.

Practical observation has shown, however, that a centralized system is in many ways advisable, and an alternative arrangement

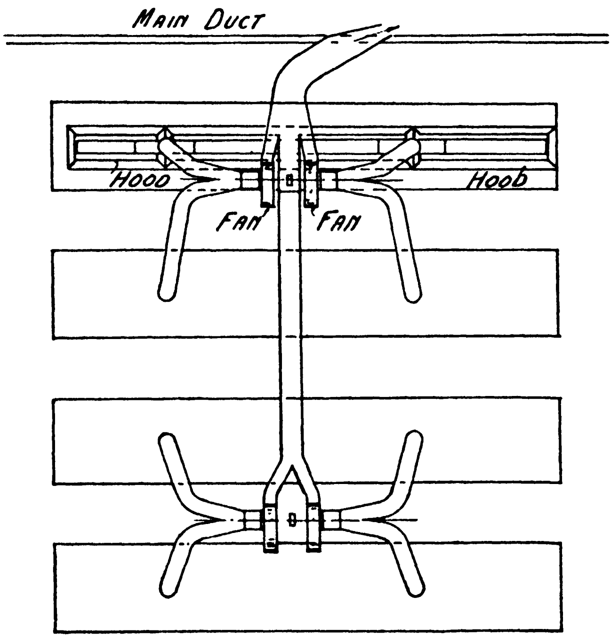


FIG. 114. Fume-extraction system.

for a similar spinning-room is shown in Fig. 116. In this case a duct is run over each machine communicating the smaller branches to hoods, the duct leading into a common suction duct outside the building which is served by a large centrifugal fan discharging to a fume chimney.

A fume-extraction system need not necessarily be served by the centrifugal type of fan, for propeller fans capable of working against comparatively high resistance are sometimes more convenient, and it would be possible to arrange a suitable hood system over the machine served by a propeller extract fan to discharge into the common duct leading to the chimney as illustrated in Fig. 117. Fig. 118 illustrates the general arrangement of the hoods over the spinning-machine.

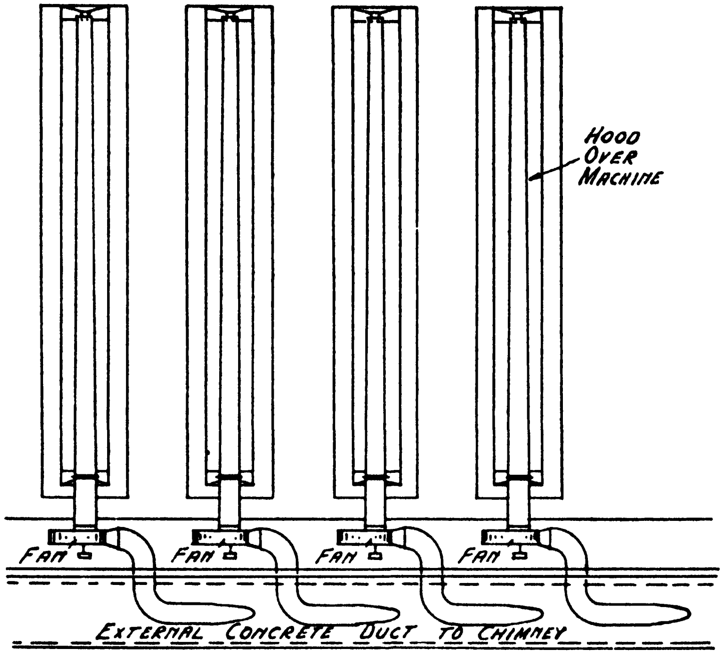


FIG. 115. Fume-extraction system.

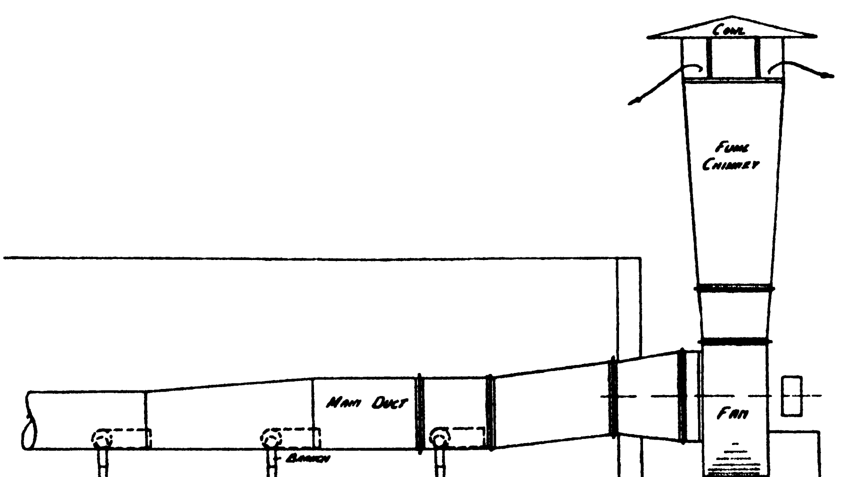


FIG. 116. Central fan fume-removal system.

Opinions have differed on the Continent upon the advisability of employing overhead ducts both for the fume-extraction system and the air-conditioning system which must necessarily be installed, without using underground ducts with their advantages of not taking up head room or obstructing light. Fig. 119 illustrates a method of extracting from a spinning-machine by ducts leading from hoods

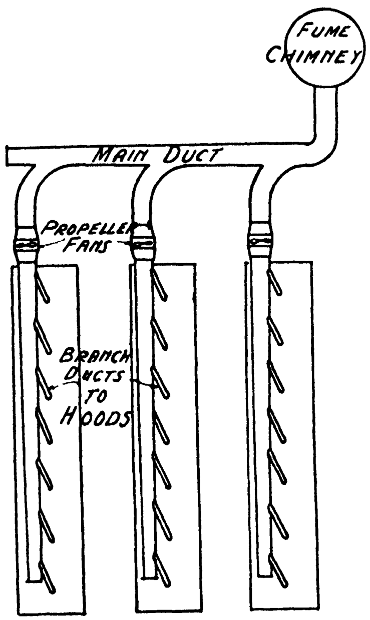


FIG. 117. Propeller fan for fume removal.

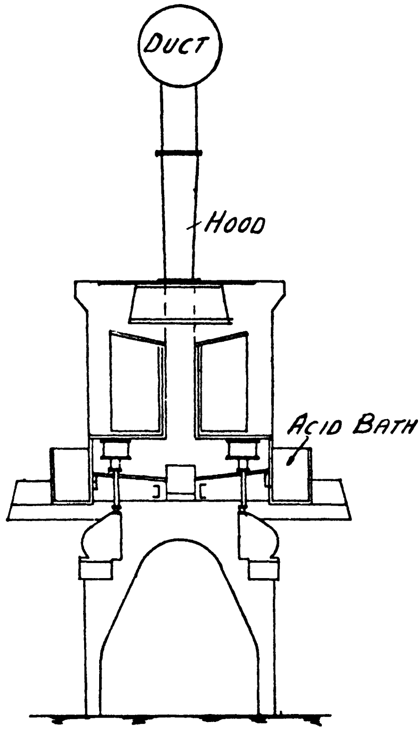


FIG. 118. Exhaust hoods from artificial silk spinning-machine.

at the top of the machine to underground main exhaust ducts; in such cases the common exhaust ducts connect together to run to a battery of centrifugal-type extraction fans which finally discharge to a chimney.

One of the largest artificial silk factories in this country employs such a system, whilst another is unique for the arrangement of exhaust fans illustrated in Fig. 120. The fume chimney is surrounded with a duct at its base on the top of which the fans are mounted with suction connexions to the duct, the discharges running obliquely into the chimney. It is felt that the alternative solutions to the problem of fume extraction in an artificial silk spinning-room amply illustrate the wide possibilities of fume extraction in general.

There are no set rules, and each problem must be taken on its own merits and decided as an engineering proposition requiring imagination and a knowledge of the general basis of design of ventilation ducts.

The velocities at which the air is carried in the various systems must depend to a great extent on the length of duct involved. With

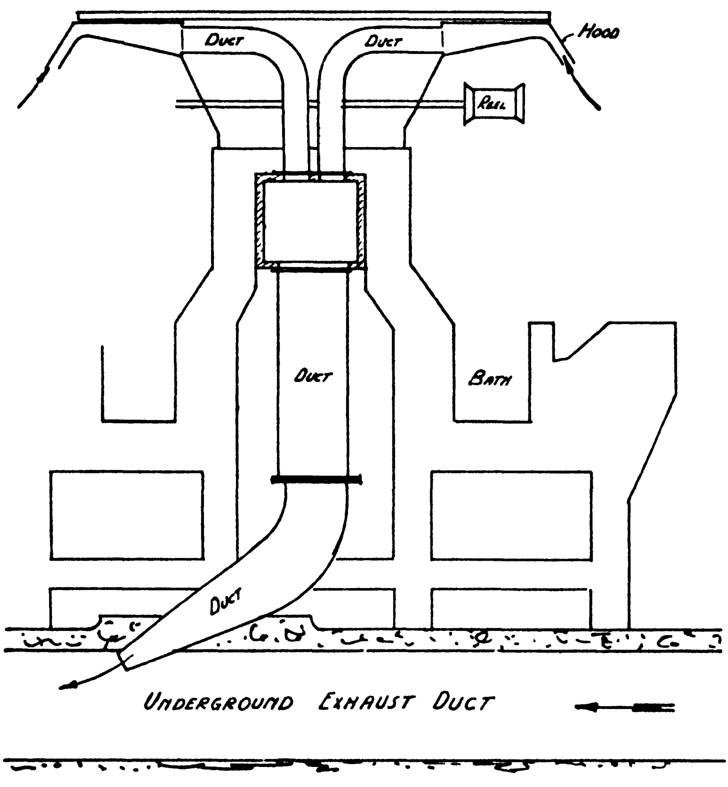


Fig. 119. Underground fume-removal system.

the majority of systems employing overhead ducts where the fan is situated adjacent to the spinning-room the velocities may vary from 1,000 ft. per minute in the branches communicating with hoods to as much as 4,000 ft. per minute in the main duct. Consideration needs, however, to be given to the possible water gauge at which the system must work, and therefore we cannot dictate definite velocities to cover every case.

With large fume-extraction systems of this nature, provided that the system is not too large, the fume chimney would be designed with a velocity of 1,500 ft. per minute at the base increasing to 2,500 ft. per minute at the top, resulting in a tapered chimney smaller at the top than the bottom. This is advantageous because the increase of velocity at discharge ensures the fumes being thrown to a reasonable height. On the other hand, the high discharge velocity necessitates high power consumption on the fans.

With comparatively small systems where the surrounding district is such that the fumes may be discharged without fear of complaint at a comparatively low level, it is then possible to arrange the fume chimneys to have a greater diameter at the top than the bottom. There have been many instances of the discharge of sulphuric acid fumes from an artificial silk mill with a fume chimney 100 ft. high leading to complaints and litigation from occupants of surrounding property. The Branston Artificial Silk Mill has a fume chimney 363 ft. high and 19 ft. 6 in. internal diameter at the top, the total weight including foundations being 5,250 tons. This chimney is, incidentally, the largest of any kind anywhere in the British Isles.

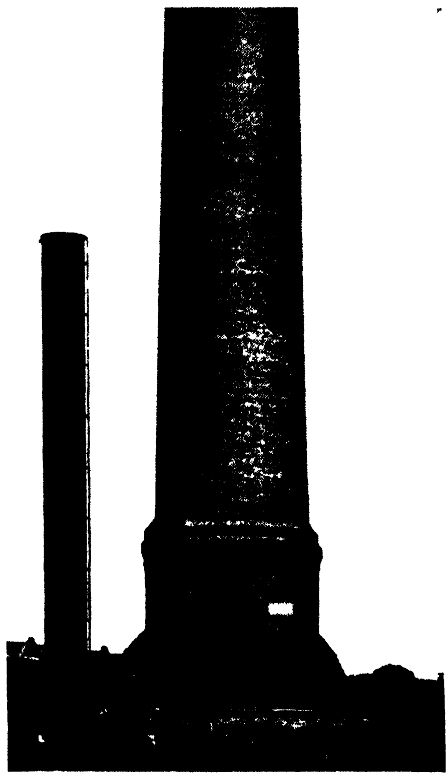


FIG. 120. Fans at base of fume chimney.

It is essential that hoods, ducting, and fans should be constructed of material which is not attacked by the fumes, so that lead-coated sheets, aluminium, stainless steel, and other forms of expensive construction generally prove necessary. The development of asbestos cement pipes offers a modern alternative to metal construction. There are many difficulties in constructing a fan which is resistant to acid fumes.

In one case a large fume-extraction fan was constructed with the housing of reinforced concrete, the fan wheel being 17 ft. 6 in. diameter, built mainly of wax-impregnated timber, the shaft being of stainless steel. Other industries have led to the construction of earthenware fans and to the use of fans coated internally with vulcanized rubber. These forms of construction, whilst they are particularly practicable are very expensive if large equipment is required.

At one stage in the manufacture of artificial silk the material is

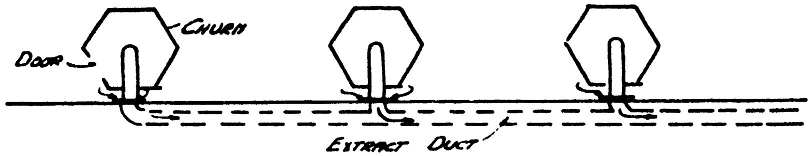


Fig. 121. Churn room exhaust system.

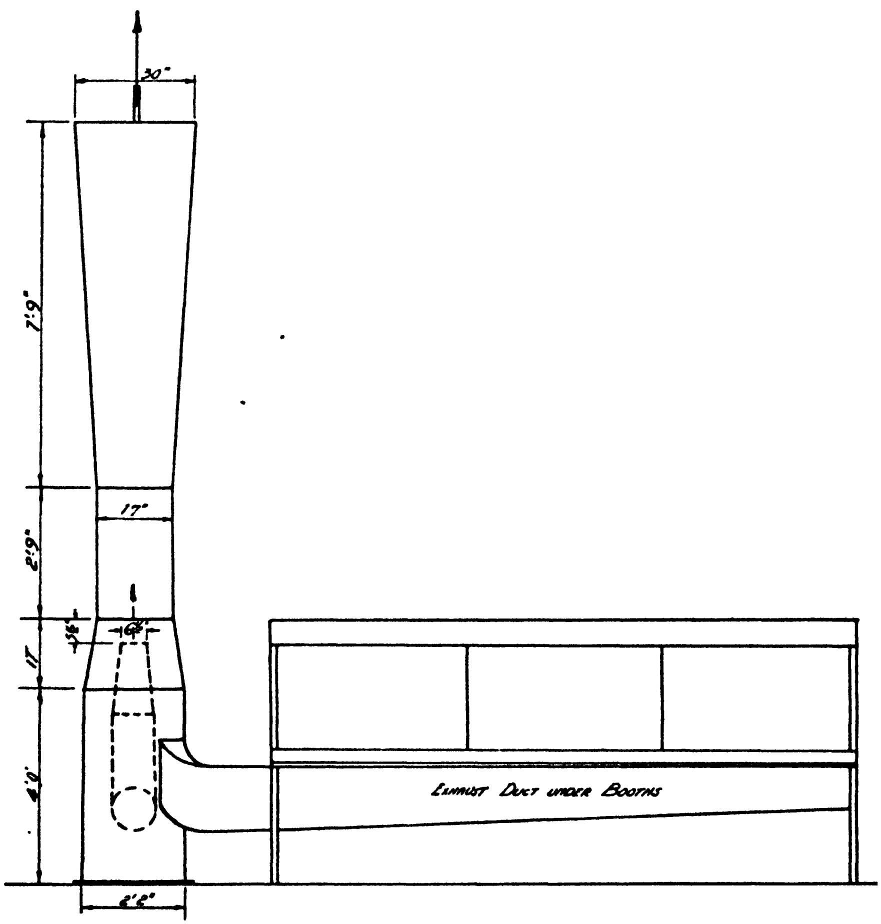


Fig. 122. Ejector exhaust from paint-spraying booths.

rotated in churns, and when the churns are stopped and the doors are opened, fumes are produced. In this case exhaust openings are arranged in the floor in such a position, as seen in Fig. 121, that the fumes will be exhausted without having opportunity to spread.

In some industries it is thought more convenient not to pass fumes through the fan. Such an instance of this is cellulose paint spraying where the cellulose would rapidly clog up the fan. One arrangement suitable for overcoming this difficulty is the use of an induced or ejector system as illustrated in Fig. 122. In this case the Venturi principle is employed, a high-velocity jet discharging or in the throat

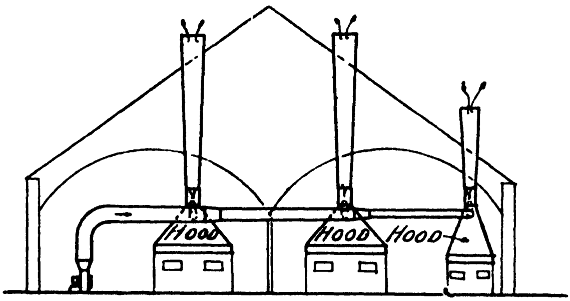


FIG. 123. Ejector exhaust from glass Lehrs.

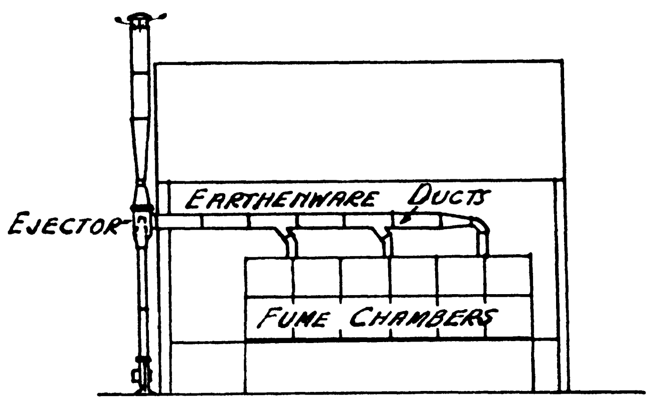


FIG. 124. Ejector exhaust from fume chambers.

of the venturi inducing a flow of air through the paint-spraying cabinets. Similar application of the ejector system is shown in Fig. 123, which is a fume-extraction system for Lehrs in a glass works.

The ejector system is also employed for exhausting air from fume chambers in chemical laboratories. In some instances the system is arranged as in Fig. 124, with an independent electrically driven blower delivering air to the venturi tube. Where fume chambers are situated adjacent to large works it is often possible to obtain a supply of compressed air for use at the ejector nozzle, whilst in other instances a steam nozzle may be employed.

It is often possible with some fumes to arrange for them to be delivered to a furnace where they may be completely destroyed; similarly they may also be discharged to a chemical neutralizing plant.

Very great care is required in designing an ejector exhaust system, and unless the system is designed with an accurately proportioned venturi it must be very inefficient. It is proposed to refer to the fundamental principles of the design of an ejector system.

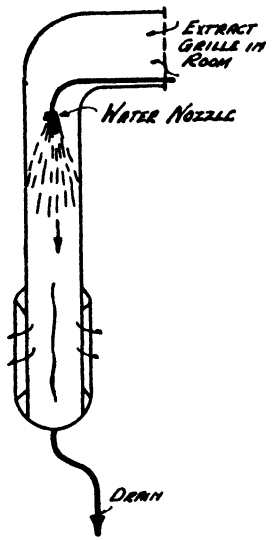


Fig. 125. Kite's water ejector ventilator.

Design of Ejector Systems.

The principle of the ejector, which relies upon the energy contained in a fluid mass, moving at a high velocity, to produce an induced flow of another mass either fluid or solid at a lower velocity, has many industrial applications.

The first application to ventilation was probably Kite's water-ventilator, which was in use some time before 1870. In this apparatus the energy of water issuing from the water-spray induces a flow of air from the room.

If the whole of the water energy is used in moving air, without any losses due to friction and changes of velocity, which would nearly be the case in Fig. 125, we may say that the kinetic energy of the motion producing mass M, moving at a velocity of V, would be exactly equal to that of the induced flow M_2, at a velocity V_2.

That is, $$\frac{MV^2}{2} = \frac{M_2 V_2^2}{2},$$

whence $$MV^2 = M_2 V_2^2.$$

This expression, which is the form given by Recknagel,† is the primary equation for any form of ejector; it holds good, only where the ratio $M : M_2$ is small and where the ejector is frictionless and without any losses.

For an ejector serving a long system of ventilating ducts, where there would be a considerable pressure loss due to friction, other factors need to be considered. No information is available on the subject in English text-books, but G. Proeschel,‡ has suggested a general method of calculation for all types of ejector based on the works of Rateau and Zeuner.

† *Kalender für Gesundheitstechniker.*

‡ 'Des éjecteurs, quelques applications dans les charbonnages', *Arts et Métiers*, Aug. 1926.

Let us consider a constricted throat ejector, as in Fig. 126, operated by air under pressure, discharged from a nozzle S_1 with a velocity V_1, this gas inducing a flow of air at atmospheric pressure, which because of the tendency to vacuum, caused by the flow V, enters at a velocity V_2, by an orifice of section S_2, which is the annular ring surrounding the orifice S_1.

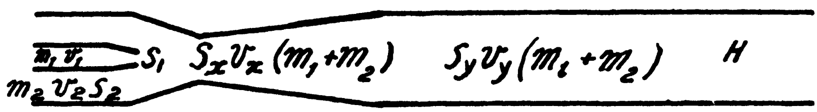


FIG. 126. Theoretical ejector.

M_1 and M_2 being the mass of the nozzle air and induced air respectively, we may say that the energy $\frac{M_1V_1^2}{2}$ of the nozzle air still exists at the end of the duct, only diminished by the losses due to changes of section.

Applying the theorems of Bernoulli and Belanger, we have:

$$\frac{M_1V_1^2}{2} = \frac{M_2V_2^2}{2} + \frac{M_1}{2}(V_1-V_x)^2 + \frac{M_2}{2}(V_2-V_x)^2 + \frac{M_1+M_2}{2}(V_x-V_y)^2 + \frac{(M_1+M_2)}{2}V_y^2 + (M_1+M_2)gH, \quad (1)$$

where H is the loss of head due to friction of the mixture of nozzle and induced air, measured in feet of air column, equal in density to that in the duct.

In this equation, and those which follow, no account is taken of the loss due to friction in the ejector itself, as this loss is always very small, and is usually compensated by the choice of a suitable coefficient of efficiency.

The ejector might equally well take the form shown in Fig. 127, with a short duct terminating in a constricted orifice.

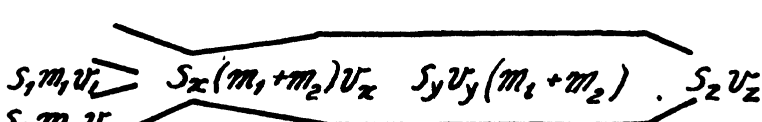


FIG. 127. Theoretical ejector.

In this case, the general formula (1) becomes

$$\frac{M_1}{2}V_1^2 = \frac{M_2V_2^2}{2} + \frac{M_1}{2}(V_1-V_x)^2 + \frac{M_2}{2}(V_2-V_x)^2 + \frac{M_1+M_2}{2}(V_x-V_y)^2 + \frac{M_1+M_2}{2}V_y^2 + \frac{M_1+M_2}{2}V_z^2, \quad (2)$$

where the term $(M_1+M_2)gH$ of the first equation is replaced by the term $\frac{M_1+M_2}{2}V_s^2$.

Section S_s may be considered as the equivalent orifice of a duct.

Many systems are in existence for ventilation, sometimes combined with humidification, in which the ejector principle is applied. Either steam, water, or compressed air is used as the motion producing fluid, and several forms of apparatus have been patented both in this country and elsewhere. The steam ejector is well known in its application to induced draught for the flues of locomotive engines, but for general ventilation purposes water at 175–215 lb. per sq. in. or air at 60–70 lb. per sq. in. is most often employed.

In mines, ejector ventilators are used for circulating the air through the various galleries in the workings, in which case the nozzles are fed by compressed air at a pressure of 4 to 5 atmospheres, the proportion of compressed air to induced being 5 per cent.

In commencing the design of an ejector system, the principle thing to be determined is the ratio of the nozzle air volume to the induced volume of air.

For fume-extraction systems it is usual to take the nozzle volume as 40 per cent. of the induced volume. We must, however, at this stage take account of the efficiency of conversion of energy, which for the air ratio mentioned may be taken as 60 per cent. Let it be assumed also that we are to design an ejector exhaust system to deal with six spray painting booths, as in Fig. 122, each having a face area of 10 sq. ft., a ventilation rate of 100 ft. per minute being maintained over the face area.

The volume of air to be exhausted is thus $6\times10\times100 = 6{,}000$ cu. ft. per min., and we will assume that the pressure loss in the system ignoring ejector losses has been calculated as 0·375 in. W.G.

This is equivalent to a velocity of $4{,}000\sqrt{0{\cdot}375} = 2{,}450$ ft. per minute.

We must therefore have a system where 60 per cent. of $\frac{M_1V_1^2}{2}$ balances the energy of the induced air stream and mixed air stream.

It is convenient to devise an energy unit for use in ejector calculations by employing units to represent thousands of cubic feet per minute and thousands of feet per minute velocity.

The various amounts of energy required are then as follows:

(1) Resistance of duct system,

$$\frac{m_1+m_2}{2}V_s^2 = \frac{2{\cdot}4+6}{2}\times2{\cdot}45^2 = 24{\cdot}5.$$

(2) Energy for giving velocity V_y, assumed in design of ducts to be 1,700 ft. per min. This is provided for in the calculated water-gauge loss of the system.

(3) The velocity V_2 on the inlet side is also taken as 2,000 ft. per min., and allowed in the water-gauge calculation.

(4) The loss in expanding from the nozzle to the throat velocity will be:

$$\frac{m_1}{2}(V_1-V_x)^2,$$

in our case, $$\frac{2\cdot 4}{2}(V_1-V_x)^2 = 1\cdot 2(V_1-V_x)^2.$$

(5) The energy in contracting from the velocity on the suction side to that in the throat will depend upon the angle of convergence, but for 30° included angle may be taken as

$$0\cdot 38\frac{m_2}{2}(V_2-V_x)^2, \text{ that is, for } V_2 = 2{,}000,\ 1\cdot 14(2-V_x)^2.$$

(6) The energy consumed in expanding from the throat to the duct at 1,700 ft. per min. for an expander of 7–8° included angle may be taken as:

$$0\cdot 3\frac{m_1+m_2}{2}(V_x-V_y)^2$$

in our case, $$0\cdot 3\frac{(2\cdot 4+6)}{2}\times(V_x-1\cdot 7)^2 = 1\cdot 26(V_x-1\cdot 7)^2.$$

As the total energy required must be balanced by that produced at the nozzle, namely $0\cdot 6\frac{(m_1V_1^2)}{2}$, in our case, $0\cdot 6\frac{(2\cdot 4V_1^2)}{2} = 0\cdot 72V_1^2$, we may now equate as follows:

$$0\cdot 72V_1^2 = 24\cdot 5+1\cdot 2(V_1-V_x)^2+1\cdot 14(2-V_x)^2+1\cdot 26(V_x-1\cdot 7)^2, \quad (3)$$

and at the same time we know that the useful energy produced by the nozzle must also exist in the throat, so that

$$0\cdot 72V_1^2 = \left(\frac{m_1+m_2}{2}\right)V_x^2, \text{ in our case, } \frac{2\cdot 4+6}{2}V_x^2 = 4\cdot 2V_x^2,$$

that is $$0\cdot 72V_1^2 = 4\cdot 2V_x^2. \quad (4)$$

We may assume now values for V_x, the throat velocity, and determine the equivalent values for V_1, the nozzle velocity, choosing such value of V_1 as will satisfy the equation (3) of nozzle and consumed energy.

By trial it may be found that with $V_1 = 11$ the value of V_x from equation (4) $= 5\cdot 2$, and these values satisfy equation (3).

The nozzle therefore is to be designed for 11,000 ft. per min. velocity, equivalent to $\left(\frac{11{,}000}{4{,}000}\right)^2 = 7\cdot 6$ in. W.G.

The nozzle area required will be $\frac{2,400}{11,000} = 0{\cdot}218$ sq. ft., equivalent to $6\frac{1}{2}$ in. diameter, approximately.

The throat area of the ejector would be designed for 5,200 ft. per min. velocity, requiring $\frac{8,400}{5,200} = 1{\cdot}61$ sq. ft., equivalent to 17 in. diameter.

The discharge end of the expander, on the initially decided velocity of 1,700 ft. per min. would be $\frac{8,400}{1,700} = 4{\cdot}95$ sq. ft. area, equivalent to 30 in. diameter.

The inlet diverging duct would need to be $\frac{6,000}{2,000} = 3$ sq. ft. in area, with an addition of 0·3 sq. ft. obstructed by the nozzle, making 3·3 sq. ft. total.

The duct from the blower to the nozzle should be sized on 50 per cent. of the nozzle velocity, to reduce friction losses, and would need to be $9\frac{1}{2}$ in. diameter.

The blower duty, allowing calculated losses in the discharge of 1·4 in. W.G., would be 2,000 cu. ft. per min. at 9 in. total W.G.

As is well known, the ejector system is extremely inefficient compared with direct fan movement of air, and for the example taken the ratio of efficiency would be $(6,000 \times 0{\cdot}375) : 2,400 \times 9 = 1 : 9{\cdot}7$, or approximately 10 per cent.

Although we have assumed the nozzle air to be 40 per cent. of the induced volume, there is no reason why it should not be any other proportion. If a greater value than 40 per cent. is taken, however, it must be remembered that the discharge ducting will be increased in size. With a smaller value the ducting will be decreased in size, but the efficiency of conversion may drop to as low as 15 per cent. or less. The average values of conversion factors are as follows:

Nozzle air per cent. of induced	*Conversion factor*
5	0·15
10	0·25
15	0·35
20	0·40
25	0·45
30	0·50
35	0·55
40	0·60
45	0·63
50	0·70
55	0·72
60	0·75

In proportioning the ejector great care is necessary, and the expander from the throat should have an included angle not greater than 7°. The nozzle must be placed absolutely central in the ejector and should be so arranged that the angle included by lines drawn from the edge of the outlet to the outside of the nearest end of the throat is 110°. Although the theoretical ejector or venturi does not include a throat of any length, it is necessary to have this 1·75 times the throat diameter for efficient conversion of energy. For ejectors operated by compressed air this is not essential, and the nozzle may then be arranged at such a distance behind the throat that lines from the centre of the nozzle to the outer edges of the throat include an angle of 40° as shown in Fig. 131.

It will be interesting now to consider the effect of an increased water-gauge on the proportions of an ejector exhaust system. We will assume that instead of 0·375 in. W.G. the system has extended discharge ducting of such a length that the W.G. is 1 in.

A water-gauge of 1 in. represents a velocity of 4,000 ft. per minute, that is $V_z = 4{,}000$ ft. per min.

The energy represented by this will be $\dfrac{2\cdot4+6}{2} \times 4^2 = 65\cdot7$ units,

but on attempting to satisfy the conditions of equation (3) it is found that the losses due to velocity changes in the ejector always exceed the available energy, which indicates that the relation between M_1 and M_2 which was chosen is not possible, and in order to reduce the losses a higher volume of nozzle air is required.

We will now assume 50 per cent., that is 3,000 cu. ft. per min., the losses being as follows:

(1) Duct system, $\dfrac{3+6}{2} \times 4^2 = 72$,

(2) Nozzle loss $= 1\cdot5(V_1 - V_x)^2$,

(3) Inlet contraction $= 1\cdot5(2 - V_x)^2$,

(4) Expansion to duct, $\dfrac{0\cdot3(3+6)}{2}(V_x - V_y)^2 = 1\cdot35(V_x - 1\cdot7)^2$.

The conversion factor for 50 per cent. nozzle air is 0·7 so that the two fundamental equations become:

$$1\cdot05\,V_1^2 = 4\cdot5\,V_x^2,$$

$$1\cdot05\,V_1^2 = 72 + 1\cdot5(V_1 - V_x)^2 + 1\cdot5(2 - V_x)^2 + 1\cdot35(V_x - 1\cdot7)^2.$$

From the first of these equations values of V_x corresponding to assumed values of V_1 may be found, and these values of V_1 and V_x substituted in the second equation until the equation is satisfied.

It is found that this is done with $V_1 = 17\cdot5$ and $V_x = 8\cdot5$.

The nozzle area is therefore $\frac{3,000}{17,500} = 0{\cdot}172$ sq. ft. $= 5{\cdot}6$ in. diameter.

The throat area is $\frac{9,000}{8,500} = 1{\cdot}06$ sq. ft. $= 14$ in. diameter.

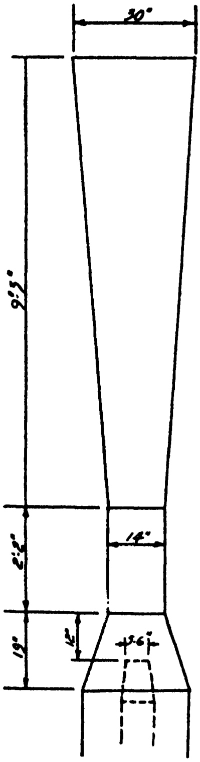


FIG. 128. Proportions of ejector for 1″ W.G.

The blower would need to deliver 3,000 cu. ft. per min. at $\frac{17,500^2}{4,000} = 19$ in. W.G., with an additional allowance for duct losses from the fan.

Fig. 128 indicates the proportions of the ejector for working against 1 in. W.G. in accordance with this last calculation.

It has been interesting to observe from this calculation how the relation of M_1 and M_2 depends to a large extent on the W.G. which the ejector must produce. Ejectors must always be based on trial calculations guided by experience of likely relations of M_1 to M_2. If calculation shows that the ratio chosen is not possible, alterations must be made, or alternatively velocity conditions in the ducts altered to vary the total W.G. which must be produced.

As a further example, we will take the same system to produce $\frac{3}{8}$ in. W.G., but assume that a supply of compressed air is available at 72 lb. per sq. in. for use in the ejector nozzle. We will assume that M_1 is 5 per cent. of M_2, that is, $\frac{6,000 \times 5}{100} = 300$ cu. ft. per min.

This volume is of course to be the volume issuing from the nozzle at the reduced pressure.

The energy required is as follows:

(1) Duct system, $\frac{0{\cdot}3+6}{2} \times 2{\cdot}45^2 = 19$,

(2) Nozzle loss, $\frac{0{\cdot}3}{2}(V_1-V_x)^2 = 0{\cdot}15(V_1-V_x)^2$,

(3) Inlet contraction, as before $= 1{\cdot}5(2-V_x)^2$,

(4) Expansion to duct, $\frac{0{\cdot}3(0{\cdot}3+6)}{2} \times (V_x-1{\cdot}7)^2 = 0{\cdot}95(V_x-1{\cdot}7)^2$.

The conversion factor for 5 per cent. nozzle air is 0·15, so that

$$0{\cdot}15V_1^2 = 3{\cdot}15V_x^2,$$
$$0{\cdot}15V_1^2 = 19+0{\cdot}15(V_1-V_x)^2+1{\cdot}5(2-V_x)^2+0{\cdot}95(V_x-1{\cdot}7)^2.$$

It is already decided that the pressure of air at the nozzle is 72 lb. per sq. in., which is equivalent to a velocity of 65,000 ft. per minute, or allowing for nozzle losses a coefficient of 0·83 this becomes 54,000 ft. per min.

$$0{\cdot}15 \times 54^2 = 3{\cdot}15 V_x^2, \text{ whence } V_x = 11{\cdot}8.$$

On attempting to satisfy the equation of nozzle and consumed energy, it is found that the losses are too great, so that they must in some way be reduced. The nozzle loss is unavoidable, but we must vary inlet and outlet velocities or construction. Any variation must not affect the W.G. of the system, however.

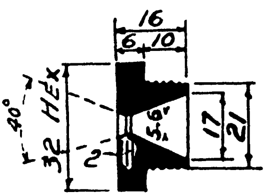


FIG. 129. Section of compressed air jet.

One of the largest losses is that of the sudden change of velocity of the induced air from the velocity of entry to the throat velocity. If, however, the included angle of the diverging nozzle is 10°, the loss at this source is approximately $0{\cdot}25 \frac{m_2}{2}(V_2 - V_x)^2$.

The equation may now have known values substituted, calling the loss at the inlet a, as follows:

$$440 = 19 + 265 + a + 97,$$

whence $$a = 59,$$

so that $$0{\cdot}25 \frac{m_2}{2}(V_2 - V_x)^2 = 59$$

$$0{\cdot}25 \frac{6}{2}(V_2 - 11{\cdot}8)^2 = 59,$$

and $$V_2 = 2{\cdot}95.$$

But to increase V_2 will slightly increase the W.G. in the suction duct, and it would be found necessary to reduce V_y to 1·2, when $0{\cdot}95(V_x - 1{\cdot}2)^2$ becomes 107; but although this is increased, it is likely that owing to the expander being lengthened, no variation need be made.

The system would be proportioned as follows:

Compressed-air nozzles are usually available in standard orifice diameters, the general proportions being shown in Fig. 129, whilst the following table gives the volume of air discharged at various pressures.

Nozzle orifice	*Compressed-air pipe diameter*	*Free air C.F.M. at lb./sq. in.*						
		20	30	40	50	60	72	100
⅛ in.	¾ in.	10	12·3	14·2	15·8	17·3	19	22·4
¼ ,,	¾ ,,	40	49	57	64	69	76	90
⅜ ,,	1 ,,	90	110	127	143	156	171	202
½ ,,	1½ ,,	160	196	226	253	276	304	358
⅝ ,,	2 ,,	250	307	355	396	432	475	560
¾ ,,	2½ ,,	360	442	510	570	623	684	808

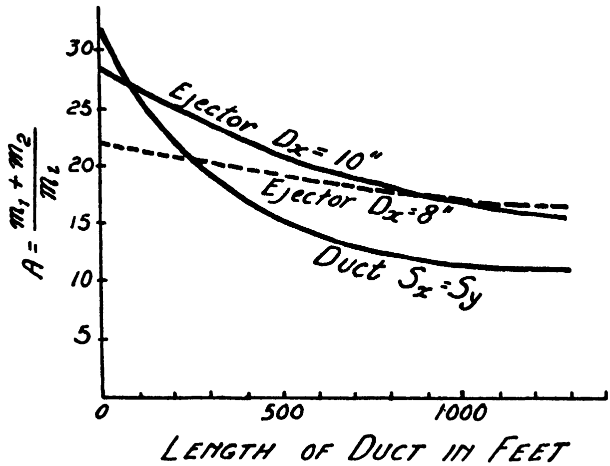


FIG. 130. Ejector output curves.

In the example taken we require 5 per cent. of 6,000 = 300 cu. ft. per min., calling for a ½-in. nozzle which delivers 304 cu. ft. per min. at 72 lb. per sq. in.

The ejector throat area, with $V_x = 11{\cdot}8$ would be $\frac{6,300}{11,800} = 0{\cdot}535$ sq. ft., equal to 9·9, say 10 in. diameter.

The inlet at 2,950 ft. per min. would need to be $\frac{6,000}{2,950} = 2{\cdot}03$ sq. ft.

The expander would have an outlet area of $\frac{6,300}{1,200} = 5{\cdot}25$ sq. ft., equivalent to 31 in. diameter.

It will have been appreciated that the overall efficiency for a compressed-air ejector is considerably less than for a low-pressure system, and indeed is usually varied between 1½ and 4 per cent. For cases where large supplies of compressed air are available the high-pressure ejector is extensively used.

Reference must be made to the use of an ejector nozzle in a parallel duct without venturi contraction. For high-pressure systems, owing to the greater difference existing between V_1 and V_x, the W.G. for a given ratio of M_1 and M_2 is considerably lower, owing to increased loss in nozzle expansion, and the overall efficiency rarely exceeds 1½ per cent. with the exception that for very low water gauges, as shown in Fig. 130, which is a theoretical output curve for the ejector in Fig. 131, the straight tube shows an apparently higher efficiency.

Having designed an ejector system, variations either of the throat diameter or the amount of air discharged by the nozzle will of course effect the volume of induced air. It has been found that to increase or decrease the throat diameter results in decreasing considerably the volume of induced air. Tests taken† on an ejector using air at 72 lb. per sq. in., the nozzle delivering 50 cu. ft. of air per minute, resulted in the following variations in output:

Diameter of throat (in.)	*Delivery cu. ft. per min.*
6½	891
7⅛	918
7½	1,057
7⅞	1,134
8⅛	817

This shows the importance of accurate construction of the ejector to correct calculated throat dimensions. For the same ejector, with 7⅞ in. throat diameter, increasing the volume of nozzle air to 168·23 increased the output to 2,072, or approximately M_2+M_1, for the same ejector, so that a slight variation from calculated nozzle conditions has but little effect on output. For a jet operating in straight piping without a venturi throat, tests by W. Hancock‡ for a nozzle pressure of 71 lb. per sq. in. operating in a 12-in. diameter tube gave these results:

FIG. 131. Ejector for mine ventilation.

M_2/M_1	*Nozzle diameter*	*Total C.F.M.*	*Total W.G.*	*Static W.G.*	*Velocity W.G.*	*Efficiency*
7·7	⅜ in.	1,478	1·00	0·8	0·22	2·361
13·9	¼ ,,	1,130	0·74	0·625	0·125	1·582
19·3	⅛ ,,	386	0·34	0·325	0·015	0·969

† 'Local air conditioning underground', by W. Hancock, Ph.D., etc., *The Colliery Guardian*, May 20, 1927.

‡ Ibid.

These figures are of interest as they show how the overall efficiency of the ejector, working on a fixed ducting system, is lowest for the small quantity of nozzle air. On the other hand, if the tests had been carried further it would also have been apparent that large increases in nozzle air had little effect on increasing efficiency, as a critical velocity would be reached.

R. A. H. Flugge-de Schmidt is of opinion that the fundamental of venturi ejector design is the avoidance of sudden changes in the rate of acceleration of flow of air in the ejector, and he has based his designs on dimensions drawn from the sine curve as representing perfect harmonic motion. Commencing with a given initial velocity, and assuming changes of velocity per unit of time to correspond with an acceleration which conforms to a sine curve with relative values of say 0, 1, 3, 4, 3, 1, 0, we obtain certain definite velocities for decreasing diameters. The diameters corresponding to the velocities are plotted, spacing them according to distance travelled in each unit of time, thus giving an ideal shape for a streamline effect. Reversing the process, the curve for increasing diameters is obtained. The change from large to small diameters may be effected quickly as the pressure tends to keep down eddies, but the expansion side requires harmonic declaration. The initial velocity assumed, and the relative values of the sine curve, the compressed-air pressure, and the type of nozzle are variable factors and therefore the actual shape of the ejector will vary accordingly. One such design on which tests were carried out is shown in Fig. 132.

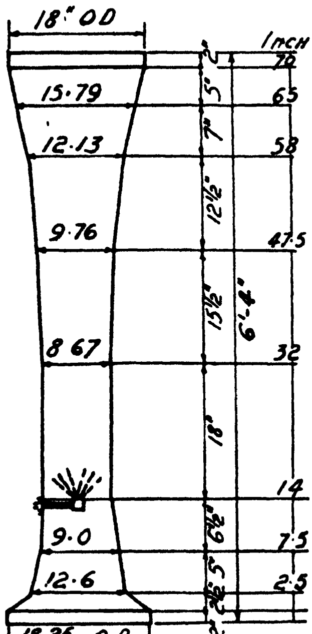


FIG. 132. Sine-curve ejector.

Professor Mellanby has given† much interesting information concerning small high-pressure ejectors which forms a useful guide in designing and testing ejectors of all types.

Collection and Conveying of Dust.

The subjects of dust removal and pneumatic conveying are so closely allied that it is not possible to give consideration to the one

† 'Fluid jets and their practical applications', A. L. Mellanby, D.Sc., etc., *Trans. Inst. Chem. E.*, vol. vi, 1928.

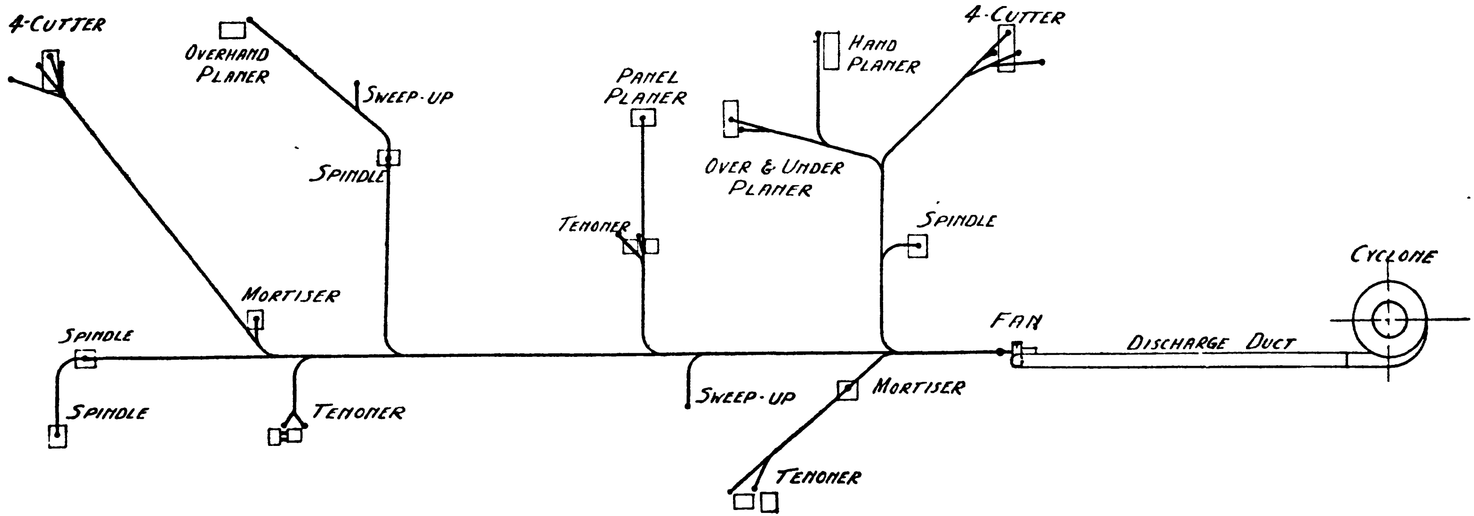
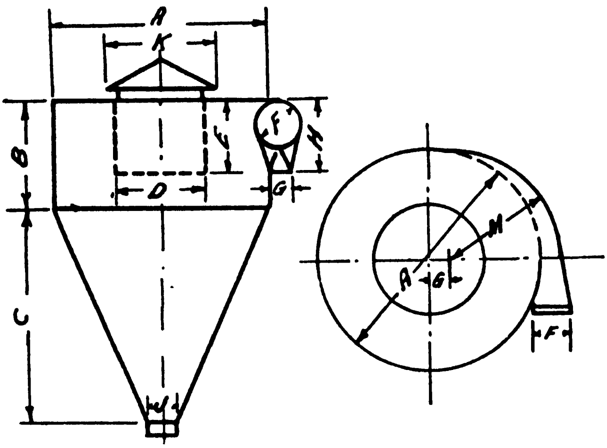


FIG. 133. Layout of chip and shavings exhaust plant.

without at the same time encroaching on the field of the other. Design in connexion with many systems is largely a question of experience and a knowledge of proportions of the various parts of the plant which have been proved to be successful with other installations.



Size	A	B	C	D	E	F	G	H	J	K	L	M
4	24	12	24	8	8	4	2	7	2	11	8	12
5	28	14	28	10	10	5	2½	9½	3	13	10	14
6	30	15	30	12	10	6	3	10	4	15	12	15
7	36	18	36	14	11	7	3½	11	6	17	14	18
8	42	21	42	16	13	8	4½	13	6	19	16	21
9	46	23	46	18	15	9	5	15	6	21	18	23
10	50	28	50	20	16	10	6	16	10	23	20	25
12	54	30	54	24	17	12	8	17	10	28	24	27
14	58	36	63	26	20	14	8	20	10	30	26	29
16	62	40	70	28	23	16	9	23	10	32	28	31
18	66	44	74	32	26	18	10	26	10	36	32	33
20	72	48	80	36	28	20	11	29	10	40	36	36
22	78	52	88	40	30	22	12	32	10	45	40	39
24	84	56	96	44	32	24	12	38	12	49	44	42
26	90	60	104	46	36	26	13	42	12	51	46	45
28	96	64	112	·50	36	28	14	44	12	55	50	48
30	102	68	102	52	38	30	15	48	12	57	52	51
32	108	72	128	56	40	32	16	52	12	61	56	54

FIG. 134. Cyclone separator.

Of the many materials dealt with by dust-extraction plants, probably the most common system is that of extracting shavings, sawdust, and chips from woodworking machinery, of which Fig. 133 is a typical arrangement. In its essentials the system comprises an exhaust ductwork system connected by suitable hoods to the machinery, the hoods being arranged in positions where they are able to collect the chips as they are formed. The shape of the hoods has become standardized for most types of woodworking machinery. Similarly, the broad basis of design is to maintain certain minimum velocities in the pipe-lines, these velocities being the minimum at which the material remains suspended in the air stream. Systems such as these are served by centrifugal fans of the paddle-blade type which may be providing a suction of 4 to 8 in. of water-gauge, depending upon the length of the exhaust piping. The following table gives the sizes of branch pipes applied to various types of machine. When the chips are exhausted from the machine by the fan they are discharged into a cyclone separator, comprising a large conical expansion chamber in which owing to the sudden drop in velocity the chips are deposited and the air leaves through the upper part of the apparatus. Fig. 134 gives the dimensions and structural details for a standard range of cyclone separator. Briefly summarized, therefore, the method of design consists in choosing the correct size branch pipe from the data given, working out the correct velocity for the particular type of material, the volume of air which would be passed by the branches, and finally proportioning main branches back to the fan on similar velocity bases. It is found in practice that a plant operates more successfully if the length of suction ducting is reduced, and particularly in large installations it is customary to place the fan in a central position so that any long distance from the woodworking shop to the position in which the cyclone is situated is on the discharge side of the fan.

An important point to be observed in the construction of the apparatus is the provision of tightly fitting dampers to shut off any branches not in use. The system which has been described may be regarded as a low-pressure exhaust and conveying system, and may be applied for conveying many types of material, and the following table gives suitable air velocities for various materials.

Velocities Required for Conveying

Class of work	*Velocity in ft./min.*
Gases, fumes, and light materials	1,000– 3,000
Cotton wool	3,000– 5,000
Shoe machinery, sawdust, polishing, buffing, and grinding	4,000– 6,000
Heaving cut wood	3,000– 4,500
High-speed wood machinery, or green timber	4,000– 6,000
Lead dust	4,000– 6,000
Corn, wheat, and oats	5,000– 6,000
Salt, sand, etc.	5,000– 8,000
Fine coal	4,000–10,000
Fur and felt machinery	6,000– 8,000
Elevating and crushing machinery	6,000
Pottery process exhaust	6,000

Details of Connexions for Woodworking Machinery

Machine	*Diameter of connexion*
Circular saws up to 16 in. diameter	4 in.
,, ,, 18–24 in. diameter	5 in.
,, ,, 26–40 in. ,,	6 in.
,, ,, 40–60 in. ,,	7 in.
Matcher heads, planers, moulders, stickers, shapers, jointers, etc., with knives:	
up to 10 in.	6 in.
,, 10–20 in.	8 in.
,, 20–30 in.	10 in.
Band saws up to $1\frac{1}{2}$ in. wide	4 in.
,, ,, $1\frac{1}{2}$–3 in. wide	$4\frac{1}{2}$ in.
,, ,, 3–4 in. ,,	5 in.
,, ,, 4–6 in. ,,	6 in.
Belt sanders up to 4 in. wide	5 in.
,, ,, 4–8 in. wide	6 in.
,, ,, 8–12 in. ,,	7 in.
Disk sanders up to 24 in. diameter	5 in.
,, ,, 24–36 in. ,,	6 in.
,, ,, 36–48 in. ,,	7 in.
Sweep-ups	6 in.
Drum sanders, 24 in. long	4 in.
,, ,, 30 in. ,,	5 in.
,, ,, 36 in. ,,	6 in.
,, ,, 48 in. ,,	9 in.
,, ,, 60 in. ,,	11 in.

Apart from the necessity of removing or conveying materials from one part of a building to another there are many instances where dust must be collected due to it being either injurious to the worker or a favourable product, or alternatively providing risk of explosion. In many cases dust must be exhausted by low-pressure fans and passed through spray washing-chambers of many types. The size and type of the dust largely influences the general principles of the plant. It is possible to ensure the elimination of dust from an air stream by discharging it into a large expansion chamber where it is settled due to the drop in velocity. It may generally be taken that if a dust contains even a small portion of particles below 10^{-2} cm. in diameter it cannot be successfully collected by means of a settling chamber. In some instances, where fine dusts are being dealt with, the air stream containing the dust is passed through a series of three or four cyclone separators arranged in series, but more usually a fabric filter is applied for the collection of such dusts.

One general type of filter used for this purpose has cloth screens made up in the shape of bags or sleeves. These bags are fastened inside a cylindrical metal container with a conical bottom, the bottom of the sleeves being opened and the top closed. The open ends of the sleeve are attached to a metal plate so that the dust-laden gauze or air entering below the plate must pass inside the sleeves, the dust remaining inside the sleeves and the air leaving at the top of the filter: the hopper at the bottom provides a space for the collection of the dust.

Some types of this filter are arranged so that a current of compressed air periodically shuts the damper in the outlet to the exhaust fan and at the same time operates a motor shaking the frame to which the tops of the filter sleeves are attached, thereby shaking off the dust from the inside of the screen and causing it to fall into the hopper, which has of course a very tight valve which is only opened periodically for the removal of dust.

It is useful to remember that any gases or dust-laden air passing through filters of this type should not exceed 250° F. in temperature when woollen material is used or 260° F. when cotton cloth is employed.

The viscous filter which is commonly used for removing the dust in air introduced into a ventilating system is often employed in connexion with small exhaust systems from grinding wheels, a blower exhausting air from the wheel and discharging it into the viscous filter cell.

As a general basis of design for conveying systems handling materials weighing between 40 and 60 lb. per cu. ft. it may be taken that 30 to 40 cu. ft. of air is required per lb. of material for successful conveying.

Pneumatic Conveying of Cotton.

The low-pressure pneumatic conveying system has been employed for the following purposes in cotton-mills:

(1) Conveying cotton after the bales have been opened to the mixing arm.

(2) Conveying the mixed cotton to the openers.

In cotton manufacture pneumatic conveying has many advantages over the belt or similar conveyors, for conveying by means of air in itself frees the cotton from much of its dust content, removes some of the moisture, and also ensures a thorough mixing. The cotton may be conveyed for as much as 1,000 ft. from one point to another, which is advantageous in allowing a works layout to be planned without regard to sequence of operations.

With one general system of conveying a feed hopper is provided to the belt breaker and a fan exhausts the cotton from the hopper and discharges it through ducts into mixing-rooms where it is deposited. An alternative arrangement is substantially similar except that the air conveying the cotton is blown into cyclone separators, the cotton falling from the bottom into the mixing rooms, an additional fan exhausting the air, which would of course contain a large amount of dust, from the top of the cyclone, discharging it into a chimney. From the mixing-rooms another pneumatic conveying system carries cotton by a suction system delivering it to a cyclone which discharges into the hopper of the opening machine. In some instances the air intake to the conveying system is provided with an air heater where it is desired to change the moisture content of the material.

Pneumatic Conveying of Wool.

In the manufacture of wool from the raw material we have an example of extensive pneumatic conveying of the raw wool conveyed from the store to the washing and drying process, from there to the carding machines, and in some cases the total distance through which the material is conveyed by one system is as much as 1,200 ft.

It is generally found with cotton and wool conveying that approximately 30 cu. ft. of air is required per lb. of material.

Low-pressure Conveying in Artificial Silk Manufacture.

An opportunity occurs for the application of the pneumatic conveying system in the manufacture of viscose artificial silk, where the cellulose alkali is placed in a hopper communicating with mixing-machines and is finally exhausted by a fan and discharged into a cyclone from which it feeds to the desired position. With this system in order to have the material in the correct condition for work the air is exhausted from the cyclone passing through a heating and humidifying chamber to the feed pipe from the hopper, the air being in a closed circuit.

The Ejector Conveying System.

There are occasions when it is not desirable to pass the material which is to be conveyed through a fan, particularly where it is of a friable nature, and for this reason the ejector system is often employed. Fig. 135 illustrates a typical application of this system for materials such as coal-dust, sugar, chemicals, cinders, and clinker from boilers which may be conveyed by this method. The ejector may be served by compressed air, or steam, according to the nature of materials and the quantity to be conveyed, or from a high-pressure fan. For fine coal 0·05 in. diameter which was conveyed by one particular system although 1,000 ft. per minute velocity is sufficient to move the particles it was necessary to have a velocity of 5,000 ft. per minute to ensure that they did not cling to the surface of the conveying pipes.

In the case of the system illustrated in Fig. 135 the fine coal dust enters the system through a feed hopper and is discharged into a storage bin 75 ft. away. This plant consisted of a fan delivering 3,360 cu. ft. per minute at a velocity pressure of about 13 in. W.G. at the ejector nozzle, the quantity of dust being conveyed amounting to 70 tons in 16 hours, or 165 lb. per minute, so that the amount of air used was about 20 cu. ft. per lb. of dust.

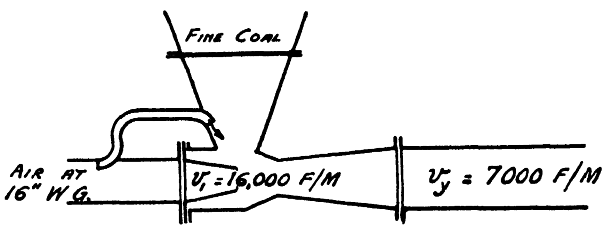


FIG. 135. Ejector conveying plant.

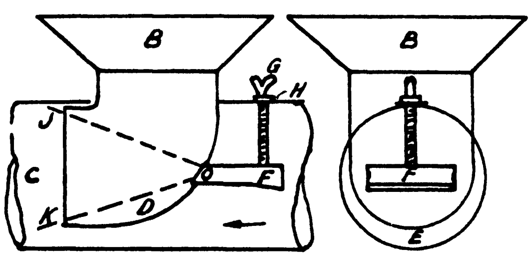


FIG. 136. Adjustable ejector for pneumatic conveying.

In proportioning the ejector the general principles outlined for the design of ejectors for fume-extraction systems should be followed, bearing in mind that the resistance of the piping system is increased by the presence of the material conveyed.

Another form of ejector is used for conveying systems for conveying grain. The compressed air actually enters the ejector by the annular space surrounding a central grain inlet, this being adjustable so that the best position of the nozzle may be obtained. It must be remembered that the power required in any pneumatic conveying system is the following:

(1) That necessary for changing material from a state of rest to motion;

(2) That for maintaining the velocity after the plant has started;

(3) Power for balancing the loss due to friction in the pipe system;

(4) The power for lifting the material through a height equal to the vertical discharge of intakes and discharges.

It is sometimes dangerous with plants operating on the ejector principle for conveying textile materials such as cotton, kapok down, and other similar materials to arrange a system blowing air along a duct, the material being fed in by gravitation at a convenient point and thus joining the air stream. Fig. 136 illustrates an adjustable ejector-valve for pneumatic systems of this type. The feed hopper together with a bend which projects into the air duct may be adjusted in position so that the annular space surrounding it through which the air passes may be varied in area, which in fact is the same as varying the throat area of the normal ejector.

Card-cleaning Systems.

With all carding machines large quantities of fibre accumulate in the machine, and there are many devices in use for removing this by a pneumatic suction plant. There are several patent devices for stripping the dust and fibre from the machine, one comprising a revolving brush engaging with the clothing of the cylinder or buffer. To clean out the material which is stripped from the revolving brush a four-wing spiked stripper is used. A light metal casing encloses the brush and stripper, the pneumatic suction duct connecting a flexible connexion to the device. Each machine is provided with this fitting and the ducts connect together to run to an exhaust fan, the dust finally being collected in any convenient type of settling chamber or filter. A somewhat similar plant is applied in rope-works, in which case the dust emanating from drawing and roving machinery is exhausted through a pipe-line connecting to exhaust headers in convenient parts of the room, the dust being discharged into collecting chambers.

Pneumatic conveying systems and dust-extraction systems all follow the one general principle, but every installation needs to be considered on its own particular merits.

Apart from the low-pressure and the ejector exhaust systems and conveying systems which have been described, there is also the high-vacuum pneumatic conveying system for conveying coal and grain and the high-vacuum cleaning systems for floor cleaning and industrial cleaning processes, but the former, however, scarcely fall within the scope of a ventilating engineer's work.

The Keith-Blackman Dust-separator Fan.

This device, which has been devised by Messrs. James Keith, Blackman Co., Ltd., combines in one compact unit a centrifugal fan and a centrifugal dust settler. Primarily evolved to meet the increasing demand for flue-dust and grit arresters in connexion with induced-draft fans for boilers, this separator fan may be employed for collecting any dry non-clogging dust, if local conditions permit of the dust being collected near the point chosen for the fan. The principle of centrifugal separation is employed in all the various

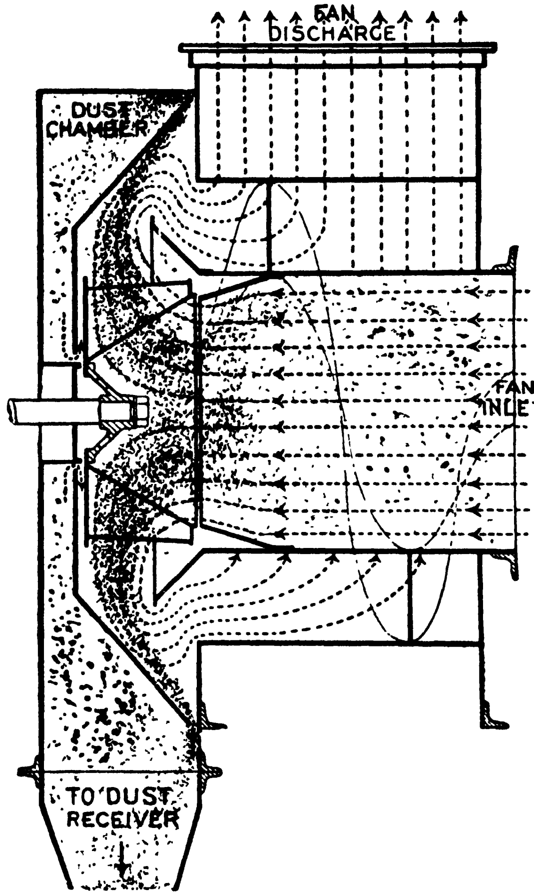


FIG. 137. Keith-Blackman dust-collecting fan.

forms of 'cyclone' dust collectors, but in this dust separator fan advantage has been taken of the fact that, under any normal working conditions, the whirl set up by the runner is considerably faster than that produced in any 'cyclone'; and, by using the momentum energy of the dust particles without disturbing the continuity of air flow, a high efficiency of dust separation is obtained at relatively little sacrifice of the manometric and mechanical efficiencies of the fan.

The final design and proportions of the apparatus are the result of a long series of practical experiments, and the principle of operation will easily be understood by reference to the accompanying diagrammatic sectional view, Fig. 137. The dust-laden flue gases (or air) are inducted through the circular inlet duct to the eye of the fan runner, at which point the flow is diverted by the runner from an axial to a radial direction. This change of direction is the first step towards the final separation of the dust, in as much as the particles tend to pass towards the backplate of the runner, along which they slide, impelled by the vanes. On reaching the periphery of the runner the dust is projected, under centrifugal action, on to the smooth slope of the deflecting cone, along which it is conducted, until, on reaching the corner formed with the casing side-plate, the dust is carried through the narrow annular space at this point into the surrounding dust chamber, assisted by a slight air current operating on a closed air circuit.

Whilst the dust is conducted centrifugally to the dust chamber—falling into the hopper at the bottom of same, and thence to any convenient receptacle—the main delivery stream of gases or air (still retaining its rotational energy) escapes into an annular helical diffuser around the outside of the inlet duct, and so, by a suitable connexion, to the chimney or atmosphere, as the case may be. When using sample of boiler flue dust in a demonstration dust-arresting fan installed at the maker's works, the proportion thus recovered is no less than 97 to 98 per cent.

Design of Central Vacuum Cleaning Systems.

Nothing contributes so much to the efficiency of a modern building as a central vacuum cleaning plant suitably designed for the duty required, especially as the initial and running cost of such a plant represents such a small proportion of the prime cost of the building and its maintenance charges.

There are two distinct systems of vacuum cleaning:

(*a*) The high-vacuum low-volume system, which maintains a maximum vacuum at the exhauster of 10–12 in. Hg (i.e. in. of mercury).

(*b*) The low-vacuum high-volume system, maintaining 4–6 in. Hg of vacuum at the exhauster.

Theoretically, if the kinetic energy of the air entering the cleaning tool is the same in each case, both plants should be equally efficient in cleaning. In practice, however, it is found that for floor and wall cleaning, which forms the greater part of the work in public buildings, the low-vacuum high-volume system is more desirable and effective, although its first cost is slightly greater than the other system.

As far as air volumes are concerned, it may be taken that the high-volume system should be capable of exhausting 50–60 cu. ft. of air per minute with a suction of 1–1½ in. Hg at the renovator hose connexion, which is equivalent to an expanded volume of 60–75 cu. ft. per minute at the exhauster. The low-volume system would handle 20 cu. ft. per minute at the renovator, or an expanded volume of 35 cu. ft. per minute at the exhauster. Higher figures are of course reached when the cleaning tool is not in use, and in such cases the working volume may rise to 100 cu. ft. at the tool for the high-volume system. The use of high-vacuum systems in public buildings is now very rare.

Suction Points Required.

The design and layout of pipe-lines and hose connexions requires careful consideration, as the successful working of the plant to a large extent depends upon this. We will consider in detail a theatre having a seating accommodation for 2,500 persons. To deal adequately with this at least 11 suction points would be required, arranged so that the whole of the building could be cleaned with a hose length not exceeding 50 ft. at each point. In the Pit 2 points would be provided, in the Stalls 1, in the Grand Circle 2, in the Upper Circle 2, and in the Gallery 3, whilst another would be provided in the Flies, about 20 ft. above the stage. The low-volume system usually has flexible hose connexions 1½ in. diameter, with a complete set of cleaning tools the principal of which are a general purpose cleaner 12 in. wide with ⅜ in. slot and an upholstery cleaner about 4 in. wide.

Capacity of Exhauster.

It may be safely assumed that in a theatre or cinema the exhauster plant should be designed to handle sufficient air for 25 per cent. of the renovators to be working at the same time, so that, for the high-volume system which we are considering where all work is done on carpets or upholstery, the exhauster would have an exhausting capacity of about 180 cu. ft. per minute, to serve three renovators. The turbo-exhauster is actually a multi-stage centrifugal fan, having four to eight multivane runners assembled in one cast-iron or pressed steel casing, the air being directed from the periphery of one runner to the inlet of the next and so on, so that the volume of air handled by the exhauster is equal to the capacity of one runner, although the total suction or vacuum produced is equal to the sum of the suctions produced by each of the runners. The peripheral velocity of turbo-exhausters is necessarily high, being from 10,000 to 15,000 ft. per minute, with corresponding revolutions of 2,500 to 4,000 per minute. The cost of running a three-renovator vacuum cleaning set for a theatre or cinema with current at 1½*d.* per unit should not exceed 9*d.* per hour.

Dust Separator.

The combined dust separator and air filter, of which many types exist, consists of a casing containing a cloth filter-sleeve through which the air must pass before entering the turbo-exhauster. The dust in the air is deposited on the filter sleeve, some of it dropping to the bottom of the separator casing and the remainder clinging to the surface of the filter cloth. In order to prevent undue resistance to the passage of air through the filter a sleeve shaking device is often provided, operated by a cam and suitable gearing from the motor driving set. The dust is removed from the bottom of the separator through a small clean-out door when the set is stationary. The resistance of the separator may be taken as 0·5 in. Hg when the filter sleeve is not clogged; when the sleeve is badly choked with dust this may be more than doubled.

Design of Pipe-lines.

Care should be taken in designing the pipe-lines that the velocity of the dust-laden air should not fall below 2,500 ft. per minute, to prevent material collecting in the pipes. The maximum velocity in the pipe-line should be 5,000–6,000 ft. per minute, as above this value the loss of vacuum due to friction becomes excessive, especially in the smaller pipes. In the flexible hose the velocity may reach a maximum of 9,000 ft. per minute.

As some indication of the various items which comprise the total vacuum to be produced at the exhauster, in a low-vacuum high-volume system, the following is given:

Approximate Vacuum Required in Low-volume Systems

Loss in renovator and 50 ft. of flexible hose	= 1·0 to 1·5 in. Hg
Loss in separator	= 0·5 to 1·0 in. „
Loss in pipe-line	= 0·75 to 2·0 in. „
Vacuum maintained at renovator	= 1·0 to 2·0 in. „
Total vacuum required at exhauster	= 3·25 to 6·5 in. Hg

As a guide in calculating the loss of vacuum in pipe-lines for a system operating at an average vacuum of 5 in. Hg, the nomograph in Fig. 138 has been prepared, and is sufficiently accurate to give also losses in suction hoses.

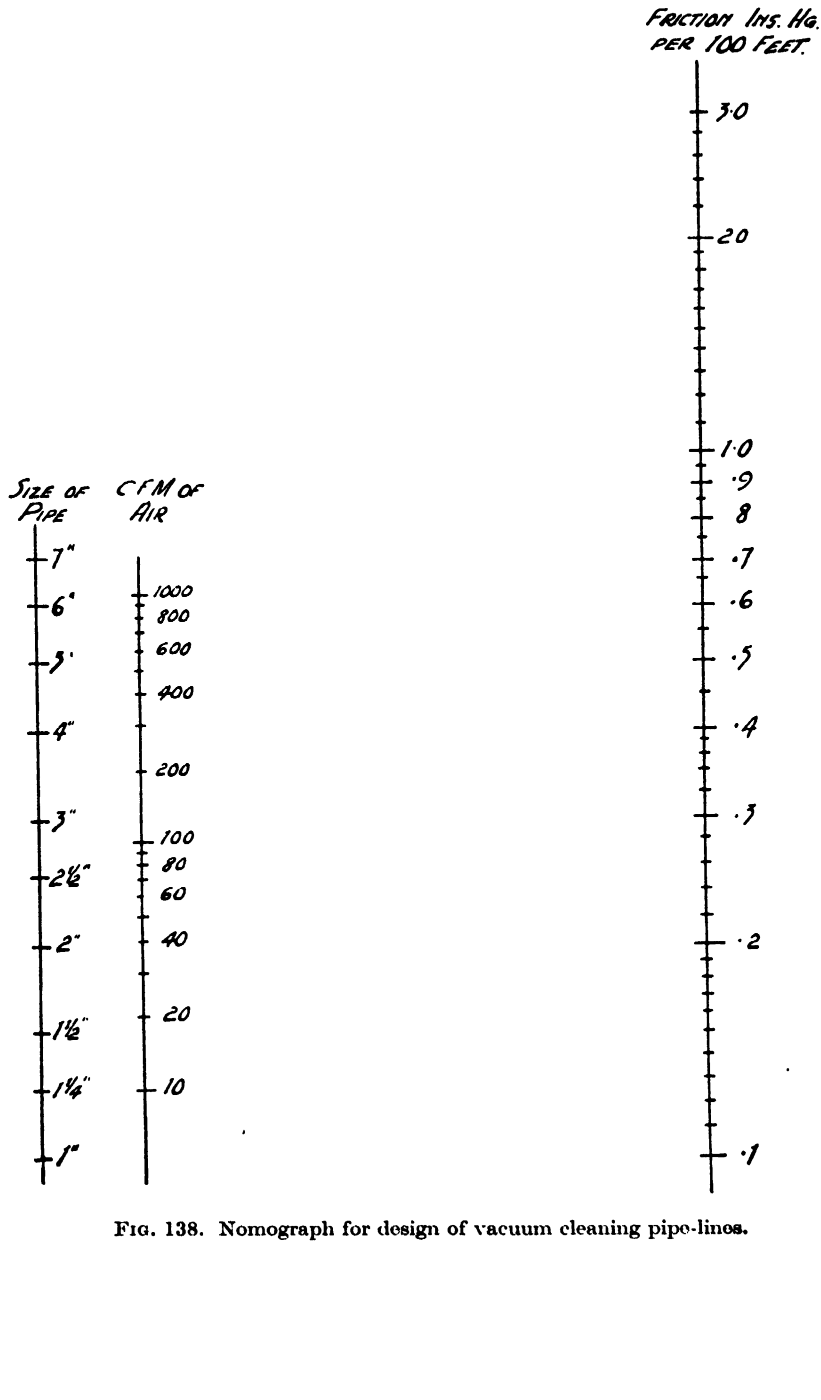


FIG. 138. Nomograph for design of vacuum cleaning pipe-lines.

Construction of Pipe-lines.

Pipe-lines should be constructed of 'steam' quality steel tube, with fire-made bends, having a radius not less than 12 in. Clean-out boxes should be provided at any change of direction to enable rods to be used for removing any stoppage. In all cases the pipes should be cut square, reamered clean, and threaded so that the end of the tube butts hard into the fittings.

The couplings at each suction point should be arranged to shut off automatically when the flexible hose is disconnected, this being arranged either by a spring or drop flap. Suitable hose racks and cleaning tool cabinets should be provided in convenient positions, although modern competition tends to eliminate these and results in tools and hoses being kept in any available store.

Power Consumption.

The power consumption of a vacuum cleaning plant varies considerably according to the nature of the material upon which the renovator is being used. A plant designed to pass 60 cu. ft. of air per minute at the cleaning tool with the tool on a carpet surface would probably actually pass 90–100 cu. ft. per minute on bare floors and walls and up to 130 cu. ft. per minute when not in contact with any surface, depending upon the characteristics of the turbo-exhauster. The larger the number of renovators served, or in other words the greater the capacity of the set, the lower is the power consumption per renovator.

The power may vary from 1·5 to as much as 4 H.P. per sweeper according to the type of work being done, the capacity of the plant, and the type of exhauster and dust separator employed, so that motors should be capable of heavy overloads for short periods. Some multi-stage centrifugal exhausters are particularly useful owing to no overloading characteristics and capability for adjusting power consumption according to the work being done with a constant speed.

It is of interest to mention that the space required for accommodating a central vacuum cleaning plant is approximately 15 sq. ft. per sweeper capacity of the plant, with a minimum of 40 sq. ft.

Testing Complete Plants.

A test on a vacuum cleaning plant must have as its object the determination of the following main factors:

(*a*) Volume of air removed from each renovator.

(*b*) Vacuum maintained at renovator.

(*c*) Volume of air handled by exhauster.

(*d*) Vacuum at exhauster.
(*e*) Loss of vacuum in pipe-line.
(*f*) Loss of vacuum in hose.
(*g*) Loss of vacuum in separator.
(*h*) Efficiency of separator with regard to dust removal.
(*i*) Overall efficiency of plant.

Another point of general interest which should be observed is the quantity of dirt removed in a specific time, particularly when comparing two plants of different make.

With regard to item (*h*), one method suggested for testing the efficiency of the separator is to connect as many cleaning tools as the plant is intended to serve, and distributing on the floor, over an area of 50 sq. ft. for each point, 6 lb. of dry sharp sand (screened through a 50-mesh screen), 3 lb. of wheat flour, and 1 lb. of powdered charcoal. Bare-floor cleaning tools are then attached, using 50 ft. of hose and the material is picked up, recovered from the dry separator, spread out again and again picked up. This is repeated four times, and then the exhausters are examined, and no trace of any of the above materials should be found if the separator is efficient.

Portable Vacuum Cleaning Sets.

It is often found convenient in existing buildings to use a portable cleaning set, as it is then found to be far simpler to provide power-points in the building rather than to fix a series of large pipes which would be required for a centralized plant. In this case the principle and operation of the plant is exactly the same. A portable set for a theatre should have an air displacement of 60 cu. ft. per minute, working on carpets with a vacuum of 2 in. Hg, and a power consumption not exceeding 1·5 H.P.

Chapter Nine

SWIMMING BATHS AND PUBLIC WASH-HOUSES

PUBLIC baths and wash-houses offer more scope than any other class of buildings for the work of the heating and ventilating engineer, who is called upon to deal not only with the heating and hot-water supply services, but in addition with general steam and power supply, special treatment in Russian vapour and Turkish hot-air baths, aeration and filtration plant in connexion with the swimming baths, mechanical washing and drying apparatus for use in both establishment laundry and public wash-houses, cooking apparatus required in connexion with staff kitchens, artesian well installations, and also fire services.

The modern tendency towards the use of more efficient apparatus in this class of buildings is evidenced by the plants recently installed in some of the new swimming baths, the total cost of engineering services for each of which could not have been much less than £25,000.

It would perhaps be interesting to recall some of the conditions which might be made for governing the general arrangement of engineering services in connexion with a typical public baths.

Extracts from Conditions and Regulations.

(1) *Boiler-house, Engine- and Pump-room, etc.* Particular stress is invariably laid upon the necessity for an arrangement for the delivery of coal and removal of ash and clinker, and strict attention is needed in determining the position of the boilers and other engineering plants with a view to enabling them to be removed either for replacement or repairs. Sufficient boiler power, with ample margin, must be provided not only for heating the buildings throughout and also for warming the water actually used in the swimming baths, but also for supplying hot water for slipper baths, steam supply to public wash-houses and establishment laundry, steam supply to staff kitchens, and the many other services required.

One of the most important points to be observed is that room should be made for a liberal storage of coal, or oil fuel if this latter method of firing is used. In the case of solid fuel, layout should be determined only after careful consideration has been given to ease of stoking and attending to the boilers.

Further space is called for by ventilation, plant rooms, pump and engine rooms, and engineers' workshops.

(2) *Public Wash-houses.* Generally, a public wash-house must be

provided with a well-lighted and ventilated washing-room with a smaller ironing and mangling room, store room, drying-rooms, and workshop adjoining. Provision must be made for separate washing compartments with the necessary hot- and cold-water supplies, boiling-troughs, hydro-extractors, and at least one drying-horse for each washer in the room, whilst the ironing room only calls for such items as ironing tables and mechanical mangles.

(3) *Aeration, Filtration, and Reheating Plant.* Some form of water-purification plant must be provided to deal with men's first-class and second-class baths, and ladies' bath, the duty being provided for by duplicate apparatus each capable of two-thirds of the maximum duty at normal load.

This plant is required for filtering, cleansing, aerating, and reheating the water from the baths, exhaust steam being used as far as possible in connexion with the plant. It is generally specified that the plant should be capable of effecting a complete circulation of the water in the swimming baths at least once in eight hours, and maintaining a constant bath-water temperature of 70° F. Provision is also to be made for filling the baths independently of the aerating and filtration plant.

(4) *Artesian Well and Cold-water Storage.* Consideration should be given to the advisability of providing for an artesian well with the necessary pumping equipment and storage tanks, in order that the baths shall be entirely independent of the Water Board supply; an emergency main supply should be provided for use in the event of the pumping plant being out of operation for a protracted period.

(5) *Heating and Ventilation.* The whole of the buildings must be adequately heated, and where necessary mechanically ventilated, in accordance with the L.C.C. or other local regulations. Particular attention should be given to the general ventilation of the swimming-bath halls. A plant should also be provided for warming and conditioning (by which is understood both extract and inlet ventilation) for the first-class men's swimming bath hall which will be used as a public concert hall during the winter months.

(6) *Establishment Laundry.* A suitable plant comprising soap-boilers, washing-machine, hydro-extractor, drying-machine, and mangle should be provided for the establishment laundry, which is intended to deal with all the swimming costumes and towels used in the three baths, the apparatus being arranged progressively, that is to say, that each stage should be from machine to machine, and finally to linen store, without the linen moving in a backward direction. A separate drying-machine is required for this section.

(7) *Turkish Baths and Russian Vapour Baths.* The heating arrange-

ments should provide for warming the frigidarium, tepidarium, calidarium, and laconicum to the usual temperature schedule required in these establishments, ventilation systems being provided for all the rooms, and also the necessary hot- and cold-water services to units in shampoo-rooms, lavatories, spray and plunge baths, etc. The services should also provide for Russian vapour baths where these are included.

(8) *Fire Services.* Consideration should be given to the question of providing fire services to conform with the Fire Brigade and Insurance Company Regulations, this section comprising ordinary hydrants with hose connexions at suitable positions in the baths.

It is proposed to deal in a fairly detailed manner with each of the various services for a building containing, for example:

(*a*) Men's first-class swimming bath, 100 ft. long by 40 ft. wide.
(*b*) Men's second-class swimming bath, 80 ft. long by 40 ft. wide.
(*c*) Women's swimming bath, 75 ft. long by 35 ft. wide.

The Cost of Engineering Services.

Before commencing upon the actual requirements of the various systems it would, perhaps, be interesting to give some idea of the approximate costs of the various sections of the plant, and these are enumerated below:

Approximate Estimate for Engineering Services at Typical Public Baths and Wash-houses

	£
Boilers, accessories, and pipework	7,000
Artesian well, pumping plant	3,800
Storage tanks	800
Aeration, filtration, and reheating plant	3,000
Hot-water supply services	3,000
Public wash-house equipment	3,000
Establishment laundry equipment	1,000
Heating apparatus	2,000
Insulation	500
Side-tip wagon and truck for boiler-house	100
Steam-engine and dynamo for lighting services	900
Electric wiring for lighting, pumps, fan, and other motors	1,400
Supplementary gas lighting	400
Cooking apparatus for staff quarters	100
Mechanical ventilation plant	2,500
Heating and ventilating equipment for Turkish and Russian baths	850
Fire hydrants	500
Cold-water services	850
Sundry machinery for workshop	150
Total	31,850

This estimate is, of course, representative only of large and high-class baths and wash-houses, where every possible service which is likely to be conducive to efficiency is included.

Filtration, Aeration, and Reheating Systems.

The increase in the number and popularity of public swimming baths has brought into prominence the condition of the water and has rendered important the question of standards of purity to be maintained in public baths water. The possibility of spreading infectious diseases has been frequently suggested, and many urge this argument against the use of public baths. If, therefore, the potential user can be made to realize that this possibility may be entirely eliminated by a suitable process of purification, there is more likelihood of his availing himself of the bathing facilities provided. On purely aesthetic grounds more people would use a bath in which the water is clear, colourless, and attractive than if it is turbid or coloured. It must be remembered that the appearance of the water is the only criterion by which the user may form an opinion upon its purity. Chemical and biological analyses are not available for him, and if they were, would in the majority of cases be unintelligible.

Until recent years it has been the practice to change the bath water at regular or irregular (usually the latter) intervals, governed sometimes by the number of baths, but more often by time or cost. At some baths the water is changed each week, at others twice a week, whilst some allow a fortnight or an even longer period to elapse before refilling. The water becomes hourly more polluted and discoloured from the time when the bath is filled, the rate depending upon the number of users; the bottom at the shallow end is soon invisible, and the water very unattractive. The practice of periodical filling conduces to a rush of bathers on fresh 'water days', while for some time before the bath is emptied the water is so discoloured that only a comparatively few people will use it.

This state of affairs is rapidly giving place to improved conditions. Progressive authorities are installing apparatus for continuously purifying the water so that it appears always in a clear, bright, and colourless condition to the bather. The bath is only filled at long intervals, usually not more than once or twice a season, but the bathers are at all times certain of water of good quality at a comfortable uniform temperature and free from any possibility of infection. Continuous purification promotes economy in the use of public water-supply, where this is used for filling the bath, the importance of this during warm dry periods being apparent.

The necessity for such purification will be self-evident, both from the point of view of public hygiene and also from that of the attractiveness of the water used in the baths, the latter being of particular importance in the case of those cities whose supply, drawn from peaty gathering grounds, is already strongly coloured when it enters the bath. In such cases the refilling of the bath every week, or even more frequently, affords only a temporary alleviation, whereas by the installation of a swimming-bath purification plant of modern type the water can remain continuously in the bath during the whole season without replacement, whilst maintaining at all times a standard of purity both as regards physical appearance and bacteriological condition, comparable with a pure public drinking-water supply.

The advantage offered by such systems may be briefly summarized as follows:

First. Continuous clean and clear water for all bathers.

Second. Probable increase in baths.

Third. Economy in consumption of town's water-supply.

Fourth. Large saving of fuel for heating the supply.

Fifth. Less labour or attention required.

Attempts have been made to effect purification by the periodical addition of a disinfecting compound to the bath water, usually at night after a day's bathing is over, but this is only a palliative. Satisfactory purification connotes the improvement of the physical, chemical, and biological condition of the water; the most that can be expected of a disinfecting agent is a reduction in the bacterial contamination, and a variation, not necessarily an improvement, in the chemical condition as revealed by analysis.

It does not fall within our scope to discuss at length the many systems of filtration employed for swimming-bath purification.†

Plant Required for Typical Baths.

We now propose to discuss the plant which would be required for the typical swimming baths which is to be considered.

The capacity of the three baths would be found as follows:

Men's first-class, 100 ft. long by 40 ft. by 7 ft. 6 in. –3 ft. 6 in.	= 22,000 cu. ft.
Men's second-class, 80 ft. long by 40 ft. by 7 ft. 6 in. –3 ft. 6 in.	= 17,600 „ „
Ladies' bath, 75 ft. long by 36 ft. by 7 ft. 6 in. –3 ft. 6 in.	= 14,850 „ „
Total capacity	= 54,450 „ „

That is, 340,000 gallons approximately.

† For details see 'The mechanical equipment of public baths and wash-houses', by A. T. Henly, *The Heat. and Vent. Eng.*, vol. ii.

It is desirable that the plant should be capable of circulating this quantity in, at the most, 8 hours, so that the duty required of the plant would be $\frac{340,000}{8} = 42,500$ gallons per hour.

This duty would call for approximately four to six filters according to the type employed and the manner in which the three baths are served.

Requirements of Reheating System.

Generally speaking, we may assume that on occasions when the bath is being refilled, water will be available at a minimum temperature of 50° F., and that it must be heated to 75° F. maximum, this being accomplished by means of an ordinary direct contact non-storage calorifier.

If the filling is to be done over a period of say 10 hours, the quantity of water to be heated and the B.T.U. required for each of the three baths would be as follows:

Men's first-class:

$$\frac{(22,000 \times 6{\cdot}25 \times 10)}{10} \times (75-50) = 3,440,000 \text{ B.T.U. per hour.}$$

Men's second-class:

$$\frac{(17,000 \times 6{\cdot}25 \times 10)}{10} \times (75-50) = 2,650,000 \text{ B.T.U. per hour.}$$

Ladies' bath:

$$\frac{(14,850 \times 6{\cdot}25 \times 10)}{10} \times (75-50) = 2,320,000 \text{ B.T.U. per hour.}$$

It should be remembered, however, that when a filtration plant is installed, filling occurs at very infrequent periods so that it would not be necessary to consider the whole of these heat requirements in determining boiler power, particularly as filling is likely to occur during the night, when the other services (wash-houses, slipper-baths, etc.) will not be in use, and moreover, it is extremely unlikely that all three baths will be filled during the same periods.

Whilst the baths are in actual use there will be very little heat to be provided, and that actually required is only necessary for counteracting the losses due to surface evaporation and radiation and a certain amount of loss by conduction through the walls and bottom of the baths. This loss may usually be taken to be served to a large extent by exhaust steam from the pumps used in conjunction with the filtration system.

Hot-water Services.

We must now give our attention to the requirements of the various hot-water supplies throughout the building to such services as shower- and slipper-baths, Turkish baths, sinks, establishment laundry, and public wash-houses.

Slipper-baths.

For the purpose of determining the quantity of water required in connexion with the typical baths being considered we will assume that the slipper-baths are distributed as follows:

Men's first-class	. . . .	35 baths.
Men's second-class	. . . .	25 ,,
Women's bath	. . . .	20 ,,
Total	. . . .	80 ,,

Assuming each bath to be used twice per hour, over a maximum load period of say 6 hours, would give 960 baths as the total number likely to be taken in that period.

This is equivalent to 24,000,000 B.T.U. if we decide to base calculations on the basis of each bath being 25,000 B.T.U.

It will be interesting to see what it would mean if the whole of this were conserved in storage cylinders.

If the temperature attained in the cylinder is taken to be 180° F., the heat content of 1 gallon of water in the cylinder would be 1,300 B.T.U., so that the storage capacity required would be:

$$\frac{24,000,000}{1,300} = 18,500 \text{ gallons.}$$

This will show how impracticable the storage system of hot-water supply is for a building of this nature, particularly when it is remembered that the boiler power required to deal with the system would even then be approximately equal to an evaporation of 4,000 lb. of steam per hour, taking into account the various losses encountered throughout the system during a heating up period of six hours.

Considering the non-storage system, we may see that the maximum demand in any one hour will be that required for 160 baths, namely 4,000,000 B.T.U. per hour.

If, as may be expected, this section of the plant is served by a calorifier having, say, 400 gallons storage capacity, equivalent to approximately 500,000 B.T.U., then it will be possible to reduce the hourly heating requirements, as it is only reasonable to conclude that the calorifier will be full of heated water when the 'draw-off' period commences.

When baths are first drawn off the capacity of the system is 500,000 B.T.U.

Whilst bathing is taking place the calorifier is heating water in readiness for the next batch of baths, until the commencement of the last batch, when theoretically steam could be shut off.

Therefore, the balance of the heat requirements, namely, 24,000,000 $-$500,000 $=$ 23,500,000 B.T.U. per hour, can be provided in the time required for the first 11 batches of baths, that is $5\frac{1}{2}$ hours.

The duty required must thus be:

$$\frac{24,000,000}{5\cdot5} = \text{say, } 4,400,000 \text{ B.T.U. per hour.}$$

This figure will be carried to the summary in deciding the steam requirements of the whole building.

Total Water required by Slipper-baths.

For the maximum demand of 960 baths the cold water required, based on 35 gallons each, would be $960 \times 35 = 33,600$ gallons, that is:

$$\frac{33,600}{5\frac{1}{2}} = 6,000 \text{ gallons per hour approximately.}$$

It is somewhat difficult to decide what storage of cold water should be allowed for a large swimming bath, as this must depend upon the town's supply available and whether there is a possibility of the supply being completely shut off for any length of time.

Generally, however, it is found that a storage of 20,000 gallons is sufficient.

The storage tanks should be arranged where possible in the form of a water tower at least 40 ft. high to provide sufficient head for distribution at the various points in the building.

In connexion with this service too great stress cannot be laid upon the advisability of giving careful attention to the sizing of the distributing pipes by methods previously explained.

For the 6,000 gallons per hour required in this typical building (assuming 35 ft. head from the highest tap to the storage tank and a travel from the tank through the supply pipe to the cylinder and thence to the farthest point in the building of 200 ft.), and remembering that each bath should be filled at the rate of 35 gallons (allowing for cleaning out) in 2 minutes, that is 17·5 gallons per minute, a total of 1,400 gallons per minute, or for one-third in use at the same moment 470 gallons per minute. From Fig. 38 for $\frac{T}{H} = \frac{200}{35}$, that is 5·7, a pipe of at least 4 in. diameter should be provided to ensure an adequate cold-water supply.

The cold-water storage tank itself, which would be about 20 ft. long by 16 ft. wide by 10 ft. deep, might well be of the cast-iron sectional type, as it would otherwise be an extremely difficult and costly proposition to erect the tank, not to mention even greater difficulties in manufacture and transport.

Turkish and Russian Baths.

Very little has been written of the plant installed for warming and ventilating the Turkish and Russian baths.

The Turkish bath, as must be well known, consists of a suite of rooms with a temperature ranging from 70° to perhaps 230° F. or more, arranged on the principle of 'continuous flow', whereby persons using the baths pass firstly into the normal temperature room and then consecutively through to the hottest room and back again through gradually cooling zones to normal temperature. The Turkish bath is, of course, not a bath as it is generally understood, but a warm-air bath.

The following temperature schedule is generally employed for this type of establishment:

Frigidarium	(normal 70°–75° F.)
Tepidarium	140°–160° F.
Calidarium	160°–180° F.
Laconicum	200°–230° F.

It will be appreciated that there must be some difficulty in designing a plant to maintain these temperatures.

The most convenient system is that employing the usual warm-air plant, consisting of filter fan and air heater, situated adjacent to the Turkish bath suite.

Ventilation is required on a basis of 4–5 air changes per hour.

Fig. 139 shows a typical arrangement of a Turkish bath suite with cubicles for 18 persons. The plunge bath, situated in the tepidarium, may be 20–30 ft. long by 8–12 ft. wide by 3 ft.–4 ft. 6 in. deep, according to requirements.

Russian vapour baths are, as their name implies, a different type of bath from Turkish baths, for, unlike the latter, a Russian bath is designed as a room containing saturated vapour instead of hot air, and is maintained at a temperature of 100°–110° F., steam entering from low-pressure nozzles.

This type of bath is rarely employed, although there is very little work required in connexion with it.

Early practice in Turkish baths favoured the use of what were known as either convoluted stoves or reverbatory furnaces, whilst the first baths of which any record exists was served by a hypocaust, or furnace, the whole of the gases resulting from the combustion of fuel being passed at a high temperature through hollow spaces under the floors and in some cases in the walls.

The use of stoves and furnaces now tends to be entirely superseded by a system of hot-air ventilation. With this system, filtered air is introduced at a temperature of 400°–450° F. into the laconicum, passing from this through the other two hot rooms, by-passes being

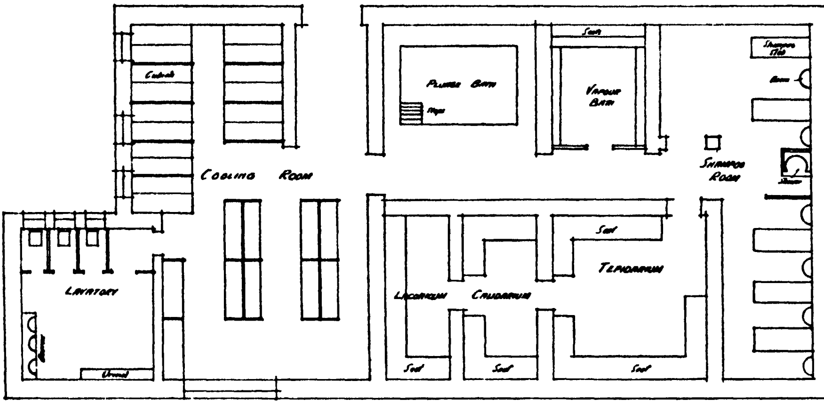


FIG. 139. Typical Turkish bath suite.

arranged so that if necessary air can be taken direct to them. Whilst gas heaters prove convenient for smaller establishments, it is desirable that the larger buildings which we are discussing should serve the Turkish bath rooms from the central steam boiler plant.

For the purpose of the typical baths we will assume that the various heating requirements will be as follows:

Room	B.T.U. per hour
Frigidarium	55,000
Tepidarium	75,000
Calidarium	37,500
Laconicum	40,000
Shampoo, etc.	15,000
	222,500

Grand total, say 223,000 B.T.U. per hour, to be carried to summary of heating requirements.

The Public Wash-house and Establishment Laundry.

In the general design of any public building such as a combined baths and wash-house considerable forethought must be given to the need of the particular district in which the building is to be situated before planning the public wash-house, as it is found that no two districts are identical in their requirements as far as this is concerned. In one neighbourhood, for example, it may be found that there is a distinct preference, by those of the inhabitants who are in the habit of using the public wash-house, for the use of washing-tubs for the majority of their work, whilst others may favour the more modern washing-machines.

It should be remembered that it is customary in many districts for one person to undertake a large quantity of washing, in which case the washing-machines are more likely to be required, as they are generally hired out for periods of a few hours a day, as required. On the other hand, when the only work to be done is that of a single family, the wash-tub holds the field.

Generally speaking, in the modern public wash-houses the tendency is towards the use of ranges of wash-tubs at the end of each of which a suitable hydro-extractor is provided for removing the greater portion of the water from the clothes after washing. The two processes of washing and rinsing are carried out in the same compartment, both steam and hot- and cold-water services being provided for each. A range of drying-horses is provided for the ends of the wash-tubs by the hydro-extractors usually along one side of the hall.

The washing and rinsing machines are usually situated at a convenient position at one end of the room.

Figure 140 illustrates a typical layout recently applied for a public wash-house showing how the principle of continuous flow may be applied to obtain maximum efficiency and economy, as it is possible for people entering the wash-house to obtain their tickets at the office, depositing prams, etc., in a store near by, and thence passing into the wash-hall where the progress is gradually made from wash-tubs to hydro-extractors and finally to the drying-horses.

The actual washing machinery installed,† with the various steam, water, and waste connexions required, and details of drying chambers, where the horses may be removed from the chamber on rollers, the clothes hanging over the rails before they are pushed back again into their places, are all obtainable from manufacturers. The necessary warm air required for drying is supplied by a fan and heater battery combination which may either be placed on top of the chamber containing the horses or in a convenient position nearby. The exhaust air from the drying-chamber is allowed to escape by way of brick flues to the atmosphere after it has removed the moisture from the clothes.

† See footnote to p. 338.

Where it is found convenient to arrange for the washing and rinsing machinery to be driven by means of steam power, it is often desirable to employ any exhaust steam emanating from these sources for heating the batteries serving the clothes-drying compartment, but this is of course entirely dependent on the circumstances of the particular installation.

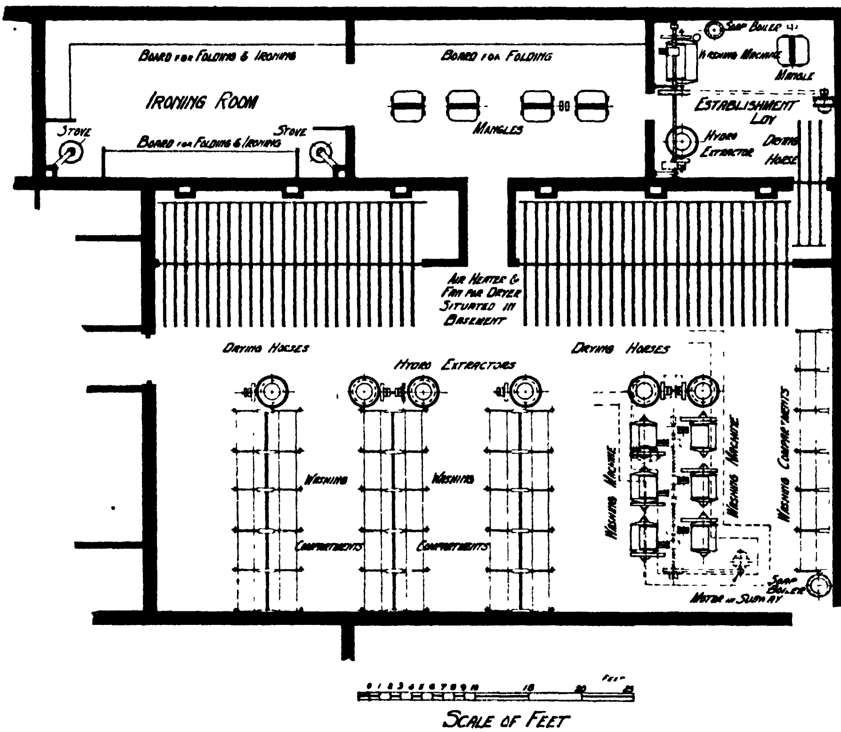


FIG. 140. Typical public wash-house layout.

In some arrangements of washing-tubs and hydro-extractor, dryers, etc., provision is made for removing any free steam from the neighbourhood of the wash-tubs by means of exhaust pipes fitted to each compartment, and so arranged as to convey any such vapours away from the persons using the tubs, to discharge through the roof when the covers of the compartments are opened; this is essential if the atmosphere round the wash compartment is to be maintained clear.

Apart from this provision, it is desirable to provide a mechanical extract system of ventilation combined, where cost is not of great importance, with a warm-air inlet system to ensure complete removal of steam from the whole of the wash-house, allowing for an air change of at least six times per hour.

Before discussing the question of the establishment laundry, which of course is intended for dealing with towels and swimming costumes used in the swimming baths section of the building, it is intended to determine as far as possible what the various water and steam requirements are likely to be for the different apparatus used in the wash-house.

For the typical baths, we may take the following as indicating the requirements:

Allowing for 34 washing-troughs, each requiring 40 gallons of water at 150° F. per hour, would call for $34 \times 40 = 1{,}360$ gallons per hour.

Providing for 20 troughs being used for boiling, each having a capacity of 20 gallons to be raised from 150° to say 212° F. by means of free steam, that is, through a temperature rise of 62° F., in say 15 minutes, would call for 400 gallons, 150°–212° F., in 15 minutes, the steam required being:

$$\frac{4{,}000 \times 62 \times 4}{966} = \text{say } 1{,}000 \text{ lb. per hour.}$$

Generally speaking, it is unusual for the water to be boiled more than twice per hour. It will be assumed that no washing-machines are employed.

Drying-closets.

Drying-horses and closets should be provided on the basis of one per washing compartment, so that we should require 34 compartments in our example, with a duty of perhaps 300 lb. of moisture to be evaporated per hour, by means of warm air at 150° F. maximum temperature.

On the average basis of $1\frac{1}{2}$ lb. of steam per lb. of water evaporated we should need $300 \times 1\frac{1}{2} = 450$ lb. per hour.

The volume of air passed through the drying-closets would be approximately 3,750 cu. ft. per minute.

Cold Water for Troughs.

Approximately 80 gallons of water per hour per trough may be taken as the maximum cold-water requirements, giving a total of 2,720 gallons per hour.

Sundry Equipment.

In addition to the washing-troughs which could well be arranged in four rows, it would be necessary to supply four hydro-extractors.

The Establishment Laundry.

The establishment laundry is intended to deal with the whole of the swimming costumes, towels, etc., employed at the baths, and as the articles to be dealt with are of more or less standard nature and dimensions the problem of choosing suitable equipment resolves itself into a considerably simpler problem than that existing in the case of the public wash-house.

Generally this section will call for the use of one or more of each of the following: combined washing, boiling and rinsing machines, soap-boiler, hydro-extractor, and power-driven mangle.

One typical arrangement of the necessary plant is shown in Fig. 141, and it will be noticed that in this case ordinary drying-horses,

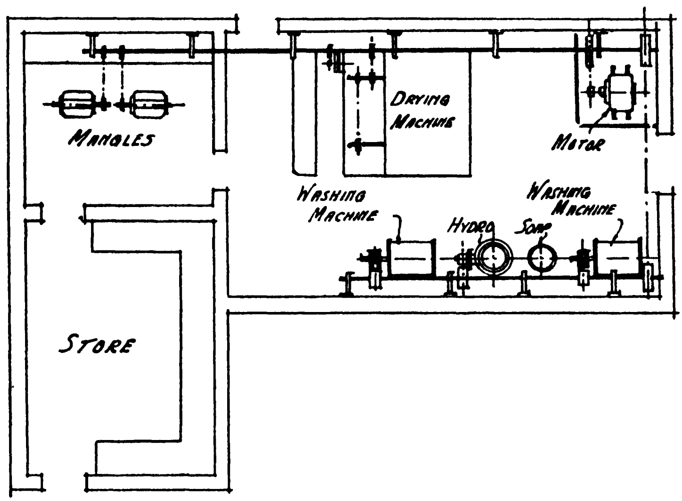


FIG. 141. Typical establishment laundry layout.

served by the warm-air plant serving the main drying-chamber, are employed.

The better-class baths will, however, invariably use a drying-machine of the continuous type which is arranged to convey the towels, etc., through the drying tunnel, with a consequent saving in time required for drying and in labour.

Requirements of Typical Establishment Laundry.

For the typical baths being considered we may assume the requirements to be 5,000 towels and 1,000 swimming costumes. This will be the requirements during an eight-hour working day. The amount of water required may only be assumed approximately to be:

700	gallons	at	150° F.
1,300	,,	,,	50° F.
Total 2,000	gallons per hour,		

whilst the steam required will be taken as 500 lb. per hour.

It would be advisable to supply one soap-boiler, two washing-machines, one 26-in. hydro-extractor, one continuous drying-machine, and two mangles, the equipment being arranged generally as shown in Fig. 141, so that as the goods leave the drying-machine, which must of course be capable of handling the work turned out by the washers, they are taken from the table at the back of the dryer, and then passed through the mangle to another table, when they are ready for distribution either to store or to the ticket office.

In this instance it will be noted that the whole of the equipment is served by an electric motor and overhead shafting, which has the advantage that the person in charge has complete control of the plant.

Artesian Wells and Cold-water Equipment.

Although the modern development of aerating and filtering in connexion with swimming baths has to a large extent nullified the use of independent artesian wells, it is considered desirable to deal with this type of equipment at some length, as many baths still find it necessary to look elsewhere than to the water-supply companies, especially in those cases where public wash-houses are also provided.

It is invariably extremely difficult to decide upon the probabilities of any strata from the water-bearing viewpoint as, although the general stratification of the principal towns and cities are well known, there are always the inevitable pitfalls for the unwary. Generally speaking, it may be taken that it is essential to make a thorough survey, and to take trial borings before saying anything definite upon the cost of the plant. Even when these precautions have been taken, very few artesian well engineers are prepared to give a fixed price for the boring operations, owing to the risks entailed. As a rule, prices are quoted for the pumping plant and storage tanks, the boring being done on the day-work basis, which actually is far more satisfactory for all concerned.

When considering the question of providing an artesian well in connexion with public baths and wash-houses, the following three important factors against the sinking of wells for public baths particularly must receive attention:

(1) The low temperature of the water-supply obtainable from artesian wells, which is likely to average 54° F. for the whole of the year, so that the water has to be heated, with a consequent expense due to fuel consumption.

(2) That owing in the majority of cases to the baths being very large water consumers, preferential charges are allowed by the water companies.

(3) The most important factor against artesian wells is the extensive use of aerating, reheating, and chlorinating plants. When these plants are used the water is used over and over again, so that the consumption is reduced to such an extent that a saving in the cost of obtaining water is nothing like great enough, when compared with the loss of interest on capital expenditure for the pumping plants.

On the other hand, where pumping plants are already installed, it is of course unwise to abandon them. Many baths in this country, particularly those built twenty-five to thirty-five years ago, were equipped with artesian-well plants.

Water-storage Tanks.

It would obviously be bad practice to supply a pumping plant capable of dealing with the highest demand likely to occur, and for this reason it is customary to install large water-storage tanks to provide against any contingencies such as high loads. With a water-storage tank arranged as a water-tower with a capacity of 20,000 gallons, the tank may be divided into two, so that one-half may be cleaned while the other is used.

In cases where it is particularly essential to maintain the cool-water supply it is usual to provide some form of insulation to the tank either of the usual plastic or cork or wood covering with thatched roof. As mentioned previously, it is invariably found cheaper to provide a sectional cast-iron tank rather than a large tank made in one piece, particularly in view of the labour and transportation charges.

In the case of the modern public baths and wash-houses, when an independent artesian well is provided it is considered desirable to have an emergency supply coupled to the Water Board's mains for use in the event of the pumping plant being out of operation for a protracted period. The plant should be arranged so that the water level in the storage tank is maintained constantly by means of automatic electric float control gear which starts the electric motor driving the pump only when the water level in the tank falls below a definite fixed point. The provision of this apparatus, besides making the plant to a large extent fool-proof, also leads to an appreciable saving in time for the engineer in charge of the plant. In the design of the well and pump-room care should be taken that a trap is provided in the floor above this room to permit of the withdrawal of tubes from the well should this be necessary.

Consideration of Typical Baths.

We will now consider in some detail the water requirements and the plant likely to be installed in connexion with the typical baths.

As mentioned it is not desirable to provide the pumping plant to cope with the full demand, and in view of this, we must decide to what extent the pump plant may be reduced to be an economical proposition.

When considering to what extent the hot-water supply service should meet the highest likely demand it is usual to base calculations upon the number of baths likely to be required, ignoring such things as sinks and lavatory basins; and in a similar manner we may decide upon the capacity of the artesian well plant for both baths and wash-houses. We have seen previously that the total capacity of the three swimming baths taken as an illustration was 340,000 gallons, and assuming that the plant was capable of filling the three baths in a period of 30 hours, a well would be required to give 12,000 gallons per hour.

This capacity should be ample to meet the requirements of the wash-house, slipper-baths, and other services.

We have considered at some length the various types of plant which must be dealt with in equipping a modern public baths and wash-houses, and we have discussed to a certain extent the heating requirements of the various sections of the plant. It remains, therefore, to summarize these requirements and to determine what would be the maximum demand for the boiler-house plant to serve the typical installation which we have up to now considered.

It will be remembered that earlier we referred to such items as the lighting system, which would usually be taken to be served by a steam-driven generating set. For such items as these it is intended to assume arbitrary values for the steam consumption likely to be required, as it is not possible to devote space to a lengthy description of the types and efficiencies of apparatus required in connexion with such plants. Similarly, the fire hydrant equipment will not require extensive explanations. It should be remembered, however, that the swimming baths will certainly not be complete without some such installation.

As far as the boiler-house plant is concerned, we must consider under this head the boilers themselves, feed pumps and sump pumps, and hot wells; also such questions as super-heating and economizers, all of which must receive consideration in an efficient plant. In addition some reference must be made to fuel storage and handling problems and also to the difficulties which may be encountered in ash removal if solid fuels are used. We must also consider the types of oil-burning equipment which are likely to be most suitable for such buildings.

Little, if any, reference has been made to the actual heating system for use throughout the building. It is not necessary to go in detail into the various facts influencing the design of the heating installation as these have been fully referred to elsewhere. Generally there are two systems which suggest themselves for use:

(1) Accelerated hot-water system;

(2) Vacuum steam system.

The accelerated hot-water system has perhaps the great merit of simplicity both in design, installation, and maintenance, and, moreover, is generally more likely to meet with the approval of the local authorities than any steam-heating system, although this may be at or below atmospheric pressure. The atmospheric steam system is, however, equally efficient provided that great care is taken in the design and installation. Compared with the hot-water heating system it has the doubtful advantage of quickly cooling down when this is desired, but as this is not likely to be required in the average swimming baths it is difficult to advocate the use of either one or the other system, and this must, therefore, be left entirely to the taste of those concerned in the design and layout.

In calculating the boiler power we shall therefore assume a reasonable figure to cover the demands made by the heating systems.

Heat Requirements of the Building.

We may summarize the various heating requirements as follows:

	B.T.U.
Warming system	750,000
Hot-water supply system	4,400,000
Public wash-house equipments	1,450,000
Warming swimming-baths, including an allowance for quick heating up	3,000,000
Turkish bath suite	225,000
Establishment laundry equipment	500,000
Total	10,325,000

Allowing the margin to cover any contingencies we may assume a total demand of 11,000,000 B.T.U. per hour, equal to an evaporation of approximately 11,500 lb. of steam per hour.

It must be obvious for this large duty that either the well-known Lancashire type or the lesser known, but certainly more compact and efficient, economic boilers are suitable.

We decided that the full heating load of the typical building under consideration should be equivalent to an evaporation of 11,500 lb.

of steam per hour. We have now to consider what types and arrangements of boilers will best meet these requirements.

It will obviously be thought desirable to supply more than one boiler in order to make some provision for the dangers of a breakdown. Generally it is considered that to supply three boilers, each capable of an evaporation equal to two-thirds full load, that is, 7,000 lb. of steam per hour, normal rating is best. It should then be possible to use one boiler as a stand-by, and moreover when the demand is not up to maximum one boiler would sometimes be ample.

It will, as a rule, be found desirable to work the boiler plant at a pressure of 80–90 lb. per sq. in. reducing the pressure for the various services as found necessary in practice.

There are three types of boilers which can reasonably be considered for the above evaporation duty:

(1) Cornish type;
(2) Lancashire type;
(3) Economic type.

The first of these must, however, be ruled out owing to the fact that the capacities are far too low.

This type of boiler, as must be well known, is of the single furnace type, but otherwise very similar to the Lancashire type.

The Lancashire type of boiler, details of which are given in the accompanying table, is one of the most suitable types for our purpose.

Lancashire Boilers: Capacities and other Details.

Diameter	*Length*	*Evaporation lb. of steam per hour*	*Thickness of plate*	*Weight*	*Weight of mountings*
7 ft. 0 in.	28 ft. 0 in.	5,000	$\frac{1}{2}$ in.	15 tons	$2\frac{3}{4}$ tons
7 „ 6 „	30 „ 0 „	6,000	$\frac{17}{32}$ „	17 „	3 „
8 „ 0 „	30 „ 0 „	6,600	$\frac{9}{16}$ „	$18\frac{1}{2}$ „	$3\frac{1}{4}$ „
8 „ 6 „	30 „ 0 „	7,300	$\frac{5}{8}$ „	$21\frac{1}{2}$ „	$3\frac{1}{2}$ „
9 „ 0 „	30 „ 0 „	8,000	$\frac{3}{4}$ „	25 „	4 „

The evaporation capacities given are those obtained by easy firing; they may be increased by hard firing. The figures given are of course only average and may vary slightly for different makes of boilers.

We may see that for the three-boilers arrangement we might employ the 30 ft. by 8 ft. 6 in. type, or if a little hard firing is not disliked the 30 ft. by 8 ft. To use the smaller size is, however, usually a false economy. For instance, when burning 25 lb. of coal per sq. ft. of grate surface per hour 10 lb. of water may be evaporated per lb. of coal, but if the rate of combustion is increased 50 per cent., or to 38 lb. the rate of evaporation may become reduced to $7\frac{1}{2}$ lb. of water per lb. of coal consumed.

In any case, it should be remembered that whatever type of boiler is used it should conform to the general requirements set out below:

(1) Ample size for the duty required, without hard firing.

(2) Simplicity of construction, with all parts easy of access for thorough internal and external inspection, cleaning, and repairs.

(3) Durability and rigidity of construction.

(4) Ample strength, with freedom from excessive deteriorating strains, resulting from unequal expansion and contraction.

(5) Sufficient elasticity to allow expansion by heat.

(6) Ample heating surface, formed to facilitate circulation and arranged in the best position for the efficient absorption of the available radiant heat and also for abstracting the greatest possible amount of heat from the products of combustion.

(7) Efficient natural circulation of water, with uniformity of temperature throughout the boiler.

(8) Ample steam space and water surface to effect silent release of steam from the water, and to secure dry steam at constant pressure.

(9) A furnace arranged in such a manner as to effect the most complete combustion of the fuel and the development of the greatest possible quantity of heat.

(10) Fire-grate surface properly proportioned to the rate of combustion of fuel and the heating surface.

(11) A combustion space sufficiently large to ensure mixture of air and fuel gases and to complete their combustion.

(12) It should have no joints or seams of plates exposed to direct impingement of flame.

(13) It should be so rated that it is capable of developing at least $33\frac{1}{3}$ per cent. more than its rated capacity by means of hard firing.

These points may be said to define the requirements of any type of boiler.

The 'Economic' type of boiler is actually a return-tube type, in which the products of combustion are passed from the furnace tubes into a brick combustion chamber at the back of the boiler, and thence returned through tubes to a smoke-box at the front of the boiler. The great advantage of this type is the fact that the boiler length is never more than about 18 ft. for the largest type, although the duty is comparable with a Lancashire boiler 30 ft. long, and about the same diameter.

Fuel Consumption.

It will be interesting at this stage to consider the fuel consumption that is likely to be required by the boiler plant.

If we take the 11,000,000 B.T.U. per hour maximum duty and allow for fuel of 14,000 B.T.U. per lb. calorific value, the quantity required per hour on a 70 per cent. efficiency basis will be:

$$\frac{11,000,000}{14,000} \times \frac{100}{70} = \text{say } 1,100 \text{ lb. per hour.}$$

Or, approximately, for every million B.T.U. per hour, equivalent to 1,000 lb. of steam, 100 lb. of fuel must be burned.

The annual fuel consumption would amount to perhaps 1,400 tons, based on comparative data for various baths.

Fuel Economizers.

In any boiler plant, as must be well known, the amount of heat in the flue gases forms a considerable part of the total calorific value of the fuel. If, therefore, some method could be employed to regain the whole or even part of the heat in the flue gases a certain economy must be effected.

The Economizer has been employed for this purpose for many years, with very effective results, considering the cost of the plant, compared with the saving in fuel. It actually consists of a water-heater formed of pipes placed in the main flue between the boiler and chimney for the purpose of economizing fuel by utilizing, in heating the feed water, a portion of the heat of the products of combustion which would otherwise escape to waste up the chimney.

It might occur to some that the length of a Lancashire boiler could be increased, with the object of providing a greater area of absorbing surface for the fuel gases to traverse, and thus in a way fulfil the functions of an Economizer. This, however, is not possible, as it has been found that the use of an Economizer is far more effective and efficient than when the ratio of length to diameter of a Lancashire boiler exceeds 4 to 1, owing to the fact that the tube surface of an Economizer is kept free from soot by self-acting scrapers, whereas the back surfaces of a boiler would in any case be thickly coated with soot, which, we need hardly mention, is an extremely bad conductor of heat. It must be remembered also that the difference of temperature of the back of the boiler and that of the escaping gases would also be far too small to ensure adequate transmission if the boiler were lengthened.

On the other hand, where the flue gases are brought into contact with water passing through the pipes with an inlet temperature of about 100° F., which it might conceivably be when taken from a hot well, there must be an efficient transmission.

The original type of Green's Patent Fuel Economizer consisted of a number of pipes about 4 in. diameter and 9 ft. long connected to top and bottom headers, the feed water entering at the bottom header and leaving at the top. An empirical formula for determining the size of the Economizer was to take $2\frac{1}{2}$ pipes per sq. ft. of fire-grate surface. The temperature of the feed water leaving the Economizer varies between 230° and 320° F., according to the flue gas temperature, and a saving of fuel amounting to 15 to 25 per cent. is stated to be effected by its use.

This would amount on the lowest saving to $\frac{15}{100} \times 900 = £135$ per annum for the typical plant which we are considering.

Feed-water Heating.

Apart from the use of Economizers for heating feed water, an appreciable economy can be effected by the use of independent steam-heated feed-water heaters. These are of two types, either the closed type, very similar to an ordinary calorifier, or the open type, where steam is injected direct into the feed water.

In addition to the fuel economy obtained, feed-water heaters are advantageous in that they ensure a constant temperature of the feed water entering the boiler, reducing to a certain extent the effects of unequal expansion and contraction in the boiler.

Superheating.

There is very little field for the use of superheating systems in the modern public baths and wash-houses unless an independent lighting engine is employed, in which case the efficiency of the engine may be increased. It is not proposed, therefore, to deal in detail with this matter.

Depreciation and Maintenance Charges.

When developing a plant for a building of this nature some thought must be given not only to the maintenance charges but to depreciation.

The rate of depreciation of any plant depends greatly upon the design and quality as well as the care taken in its use, and it is in these respects that careful thought should be given to the advisability of paying a little more for the equipment when the life of the plant can be increased.

Sundry Equipment.

Apart from the various specialized equipment for use in public baths and wash-houses discussed in previous articles of this series, there are various items which although they do not require much explanation must nevertheless be referred to for completion of this series.

One of the most important items in connexion with the boiler-house plant is, perhaps, the question of oil fuel. Oil-firing systems should always be advocated in the class of building under discussion, both from the point of view of efficiency and general cleanliness, so obviously great care is necessary to ensure that the surrounding atmosphere is as dust free as is possible. It is not proposed to discuss in detail the various fuel-oil systems. Care should, however, be taken in the layout of the plant to ensure that ample space is available for oil-fuel tanks, which should be arranged near to or, if possible, under the roadway to facilitate the filling operations. The actual system of firing used should be carefully chosen for its purpose, and in any case duplication of the most important items of the plant should be arranged for, to guard against the chances of a break-down.

We need hardly stress the advantage of oil-firing systems compared with coal-firing. For the latter, the extensive bunker space, ash and fuel runways, etc., are a far more expensive proposition than the slight additional cost of oil-firing equipment.

Another very important point to which we have made little, if any, reference is the question of boiler flues, and this is, unfortunately, a point which is all too often given little attention. One of the most important points to be observed as far as concerns the boiler plant is to obtain an efficacious draft in the flues, which can only be obtained by the rational design of the boiler flues and chimney stack. The chimney can either be too large or too small. In the first instance the result would be that far too much air would be drawn through the boiler furnaces, with excessive fuel consumption, whilst apart from the excess of fuel burnt (in an uneconomical manner) there is a risk of the boiler being damaged, due to overheating and the formation of clinker.

In the latter instance, of course, combustion would be neither sufficient nor complete and the boiler could not cope with the requisite duty.

It is not possible to calculate in anything approaching an accurate manner the correct size for a chimney, but by the judicious application of the nomographic calculator, Fig. 108, coupled with some experience of the variations likely to be met with in different localities, it should be possible to obtain reasonably accurate results.

A word now as to forced or induced draught. By the use of either a forced or induced draught system it is possible to obtain increased efficiency and output from a given boiler plant, but, generally speaking, for the public bath and wash-house boiler plant the additional cost of the plant involved will not be justified unless the plant is of a very large order. Moreover, the additional space required and the extra attention to be given in running the plant are both inclined to favour discarding induced or forced draught.

Similarly, we may say that the use of a pre-heater system, by which is understood some system by means of which the air used for supporting combustion is heated by indirect contact with the hot flue gases, is also rarely to be used for a public bath and wash-house system, as this involves the use and control of one or more fans. Here, again, the efficiency of the plant may be increased a certain amount, but it is questionable whether the additional capital expenditure is warranted.

Apart from any consideration of plant, there is one item that should certainly receive close attention, and that is the question of recording and indicating instruments, as by their use alone it is possible to obtain the best possible efficiency.

Instruments should be provided for indicating the following items:

(1) Flue gas analysis.
(2) Steam pressure and temperature.
(3) Steam flow.
(4) Feed water temperature.
(5) Calorifier temperatures.
(6) Quantity of hot water used.

These and other similar indications are to the intelligent engineer in charge the only true guide to the efficiency of the entire plant, and by the intelligent use of the figures thus obtained it should be possible to obtain maximum efficiency and indication of whether the plant is being correctly handled.

Ventilation and Air Conditioning of Bath Halls.

The bath hall should always be ventilated at such a rate as will give four air changes per hour. In small baths this is accomplished by propeller extract fans in the end walls. In other baths a central extract system is installed. There is much to be said for the idea of extracting air at low level round the edge of the bath, the warm fresh air entering at high level to form a downward ventilation system. This system ensures that air is drawn off primarily from the area in which the occupants are situated.

Care should be taken that any bath hall which is to be used as an assembly hall should be provided with complete heating and ventilating equipment to conform with local requirements or standing order, the air change being determined according to the required volume of air to be allowed per person.

Heating of Open-air Bathing-pools.

The problem of the open-air swimming-pool is one receiving constant attention in view of the large number of pools now being used. It is evident that the open-air pools will, under certain conditions, lose an enormous quantity of heat, and in extreme winter weather it is hardly practicable to maintain the 70–75° F. water temperature usually required, for a wind will lead to high water evaporation and consequent heat losses.

It is usual to base the boiler plant on sufficient power to raise the contents of the bath through 5 degrees per hour, which provides ample margin for the initial heating up and the losses, which generally account for 3 degrees per hour temperature loss.

Chapter Ten

SPECIAL REQUIREMENTS OF FACTORIES, SCHOOLS, ETC.

Factory Heating Schemes.

HEATING schemes for factories of modern construction call for many special arrangements not generally met with in other classes of building. The heat loss of a factory building according to construction may average $1\frac{1}{2}$–4 B.T.U. per cu. ft. of space heated, and when galvanized iron or corrugated asbestos walls and roof on steel frames is used, even higher than this. The necessity for leaving floor-space free eliminates in most instances any chance of concentrating radiators at low level, so that overhead pipes or unit heaters become the only means of providing a heating system at reasonable cost.

When using overhead pipes it is not unusual to find the temperature near the roof 15°–20° F. higher than at the breathing-level, which not only results in high roof-transmission losses but also lowers the emission of the overhead heating pipes which are situated in the higher temperature. This remark is particularly applicable to hot-water heating where the mean temperature of the pipes is low. The author has found that it is not possible to obtain more than 60 per cent. of the usual emission as measured with required room temperature to be effective at low level, in spite of theoretical considerations which may show greatly increased emission for pipes suspended in free air in large factory buildings. This does not mean to say that the amount of heat to be supplied to a building compared with the normal calculated heat loss would be $\frac{100}{60}$: 1, that is, 1·7 to 1, for the heating surface is inefficient and does not transmit normal heat. For hot-water heating, in deciding upon the amount of overhead piping, the following allowances are desirable:

Size of pipe	*Temperature difference between mean water and desired room temperature with emission in B.T.U. per ft. run*			
	120° F.	110° F.	100° F.	90° F.
2 in.	100	92	84	75
$2\frac{1}{2}$ in.	115	105	88	86
3 in.	140	128	117	105
4 in.	175	160	146	131

The heat loss should be calculated in the usual manner, assuming

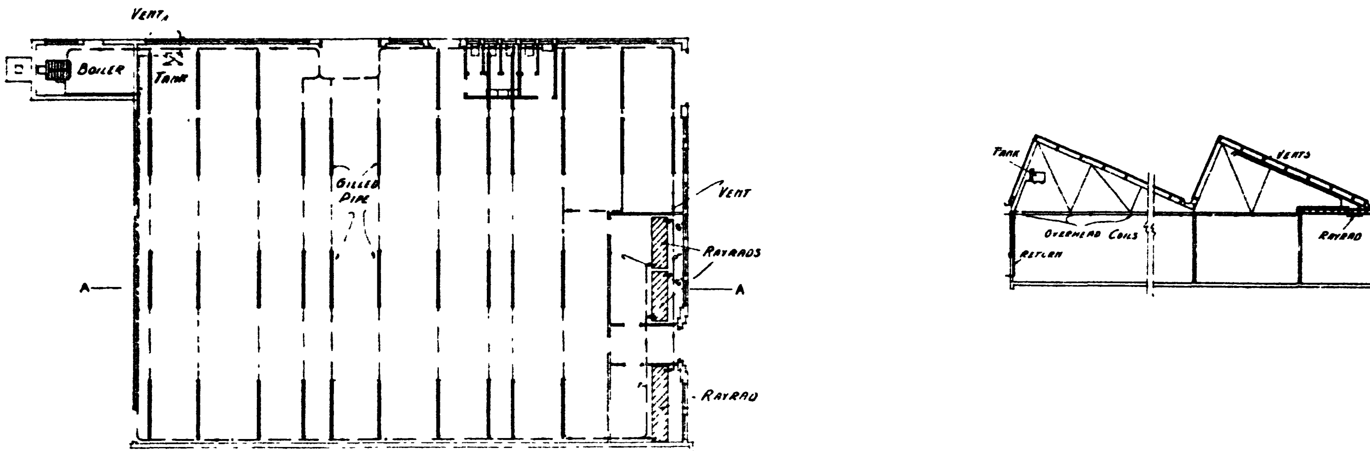


FIG. 142. Factory heating scheme with overhead gilled pipes.

the whole of the surfaces to be surrounded with air at the desired internal room temperature. The surface should then be determined from the above figures, but as they represent effective heat at low level and not actual transmission, 25 per cent. should be added to obtain actual transmission in calculating pipe sizes.

The use of overhead pipe coils for heating a factory building leads to 25 per cent. increase in fuel consumption due to high roof temperature.

With hot-water heating in particular, it is usually impossible to have the whole of the heating surface in the form of plain pipe coils

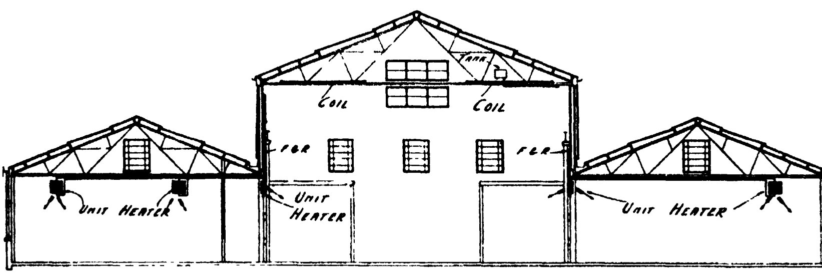


FIG. 143. Section through factory showing unit heaters.

and resort must be made to lengths of gilled tube to give heating surface in a small space. Usually with overhead heating the transmission coefficient of gilled pipes is 1 B.T.U. per sq. ft. per degree difference in temperature, but there is a practical limit to the closeness of gills for obtaining high heating surface.

No gilled pipe should be used having a ratio of total surface to pipe surface greater than 10:1, for closer spacing of gills or increased diameter lowers the transmission coefficient so much that the increased surface emits less heat than less surface with wider spaced and smaller diameter gills.

Fig. 142 shows a typical factory-heating scheme where gilled pipe is used, with accelerated hot-water circulation.

Apart from any question of initial cost, the use of unit heaters is a far more efficient arrangement, for the difference in temperature between roof and floor with this system is very slight and no heat is required for maintaining a high-level temperature in excess of the desired room temperature. For large factories there is no system which is so efficient or can be so cheaply installed as unit heaters, and a typical recent example of a large factory heated on this principle is given in Fig. 143.

The Factory Power Plant.

Many factories produce large quantities of wood or other refuse during the manufacturing process, and in such cases it is a good arrangement to use the refuse or waste in a gas-producer plant which will produce sufficient gas to run an engine for generating current for power and lighting purposes and for heating the works. The gas produced by this means is of low calorific value, about 135 B.T.U.

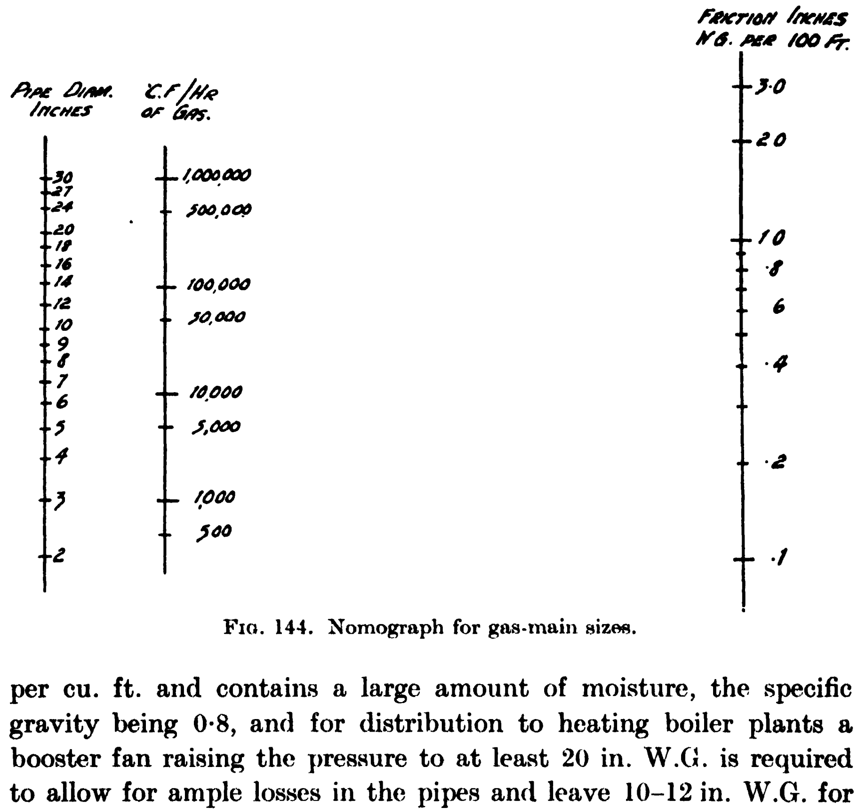


FIG. 144. Nomograph for gas-main sizes.

per cu. ft. and contains a large amount of moisture, the specific gravity being 0·8, and for distribution to heating boiler plants a booster fan raising the pressure to at least 20 in. W.G. is required to allow for ample losses in the pipes and leave 10–12 in. W.G. for burner pressure.

Owing to the low calorific value large volumes are required, and the high specific gravity causes friction losses to be high. Consequently the gas-distributing mains are large. The nomographic calculator in Fig. 144 will be found useful in deciding upon suitable sizes of gas mains. Scales are given for the pipe diameters, quantity of gas in cubic feet per hour, and loss of pressure in inches W.G. per 100 ft. per hour.

Whilst there are several types of boiler designed for firing by gas alone, the cast iron sectional boiler is so cheap that it is often employed for gas firing, and in these instances a pressure gas system is desirable, particularly with gas of low calorific value, to keep to reasonable dimensions of burning equipment.

The burning equipment most suitable for boilers of this class is the radiant type of burner illustrated in Fig. 145, which has the

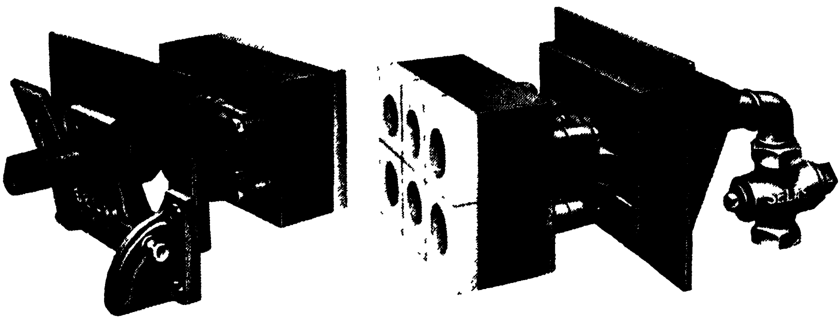


FIG. 145. Radiant gas burners.

advantage of giving a relatively low temperature of combustion and distributes heat by radiant flames. This type of burner relies largely upon natural chimney draught to induce the air for combustion, requires very little excess air, and gives a combustion efficiency of 80 per cent.

The burners consist of multiple refractory firing-blocks, a cast-iron gas manifold, and air regulator and frame. Gas control is effected by various standardized thermostatic devices, with an air control operated by gas pressure.

Fig. 146 shows a works boiler served by the type of gas burner described. It should be pointed out that with some boilers partial fire-brick lining may be necessary to prevent flame impingement.

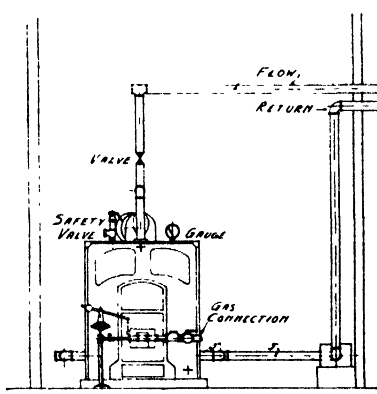


FIG. 146. Factory gas-fired boiler.

The Heating of Schools.

School planning has changed in recent years, and few completely new schools are built unless they are of the open-air or semi-open-air type, so that difficulties arise when the heating scheme is being designed.

With existing school buildings one of the most important points to be observed in the application of central heating is even distribution of radiator surface. It is not satisfactory to have radiators sufficient to heat the room placed on one external wall only. At least two sides of the room should have radiators and preferably pipes should run at the back of the classroom.

Whilst it is universally agreed that an internal temperature of 60° F. is satisfactory for a school, the question of air interchange is one which is still creating much trouble. Practical observations suggest that the most satisfactory basis of design is an allowance of an air interchange of three times per hour, but even so instances occur where the obstinacy of the teaching staff, insisting on all windows being wide open, possibly on two sides of a classroom, results in the actual air change being many times the allowed figure, and the room being cold.

Investigation of a complaint invariably shows this to be happening, but the unreasoning outlook on air requirements by those who have neither the understanding nor experience of what is actually required does not permit them to draw a parallel with their own homes, where they would not hesitate to shut the window if the fire was not providing them with sufficient comfort.

The modern type of open-air school is usually heated by either floor or ceiling radiant heating.

The question of air change, in spite of theories on radiant heating, is a matter which proves difficult, and, generally speaking, no theory is evolved which is capable of deciding upon the adequacy of any amount of heating surface to heat the classroom of an open-air school. Most of the schools where low-temperature ceiling radiant panels are used have surface equivalent to 60 sq. ft. per 1,000 cu. ft. of space, but even so a wind on one side is sufficient to nullify the effect of the heating system.

The floor-heating system is one which has been extensively applied in schools, and this exists in many forms, which are illustrated in Figs. 31–33.

Practically the whole floor area is devoted to heating surface at low temperature. Similar systems are also of great value in the heating of large churches and halls.

It is to be regretted that no authority has yet given sufficient consideration to the problem of the mechanical ventilation and air conditioning of schools as to have a school equipped in this way. An unfortunate investigation in America demonstrated that schools which were ventilated by natural means had a higher percentage of attendances and less illness than any others, which led to general condemnation of mechanical ventilation. There is, however, no doubt that a well-designed air-conditioning system can produce an atmosphere far better than any resulting from the open window in both summer and winter months, but to be able to do this air-washing, filtration, and accurate temperature and humidity control are essential.

With the modern single-story school planning the layout of an air-conditioning system becomes simple, with a false ceiling in the corridors for conditioned air and extract ducts in the floor of the corridors. To be effective at least six air changes per hour are required, and above all the air must be introduced and extracted in a draughtless manner.

Cinemas, Theatres, and Assembly Halls.

Regulations imposed by local authorities call for efficient ventilation in all cases for cinemas, theatres, and other buildings where large numbers of people congregate, but in few cases are the regulations sufficiently rigid to ensure adequate ventilation.

It is to be regretted on how few occasions adequate provision is made by the architect for heating and ventilating equipment. When planning a cinema, the projection room invariably is fully provided for, but on many occasions anything but adequate space is allowed for the air-conditioning plant, whilst the boiler room receives in some instances no more attention.

Apart from its primary use as a suitable room for the boilers, the boiler house will doubtless need to provide sufficient space for accelerating pumps, a hot-water storage cylinder, and, if solid fuel is being provided, ample space for fuel storage. For buildings such as cinemas and theatres the space requirements remain fairly consistent for a given size building where the heating and air-conditioning schemes are designed on clearly defined lines.

The following table indicates the necessary space for the boiler house under various conditions:

Space Requirements of Cinema and Theatre Boiler House

Seating capacity	*Area required in sq. ft. for various fuels, including fuel-storage space, and all plant*		
	Solid fuel	*Oil*	*Gas*
1,000	210	170	130
1,500	285	230	180
2,000	360	290	210
2,500	435	350	250
3,000	510	410	310
4,000	660	530	400
5,000	810	640	480

The general lines of the building will, of course, dictate to a large extent the shape of the portion allocated to the boiler house, but generally the ratio of length to width should be two to one as a maximum, anything more elongated than this providing unsatisfactory arrangement of the plant and other equipment. As may be seen where oil or gas firing is employed, there is considerable saving in the space required when compared with solid fuel.

In the comparatively early stages of the building development it is necessary to define the dimensions of the chimney. It will be appreciated that the necessary cross-sectional area of the chimney will depend not only on the size of the building but also on the height to which the chimney is carried, as has been explained elsewhere. The following table gives for cinemas or theatres suggested dimensions of chimneys under various conditions. It must be remembered that the chimney is almost the key to the successful working of the boiler plant. Too much care cannot be taken in seeing that its dimensions are adequate. It is true that in some cases chimneys of smaller dimensions prove to be satisfactory, but even so, there are occasions during adverse winds or atmospheric conditions when the necessary draught is not available.

Sizes of Boiler Chimney for Cinemas or Theatres of Various Seating Capacities

Seating capacity	*Internal dimensions of chimney for various heights above boiler house floor*				
	30 *ft.*	40 *ft.*	50 *ft.*	60 *ft.*	70 *ft.*
	in.	in.	in.	in.	in.
1,000	18 × 18	18 × 18	14 × 18	14 × 14	14 × 14
1,500	18 × 24	18 × 24	18 × 18	14 × 18	14 × 18
2,000	..	24 × 24	18 × 24	18 × 24	18 × 18
2,500	..	24 × 24	24 × 24	18 × 24	18 × 24
3,000	..	24 × 30	24 × 24	18 × 24	18 × 24
4,000	..	30 × 30	24 × 30	24 × 24	24 × 24
5,000	..	30 × 36	30 × 30	24 × 30	24 × 30

Apart from space considerations the architect needs also to contend with providing bases and ashpits for the boilers and bases for the hot-water storage cylinder and circulating pumps. Another important factor which in many cases in cinemas and theatres needs careful attention is the provision of adequate ventilation to the boiler house; this ventilation, as with most other buildings, is provided naturally, and it is necessary to provide inlet grilles or louvres, the air being drawn in by the natural chimney draught. The following are the suggested free areas of openings for providing fresh air to cinema or theatre boiler houses:

Areas required for Air Inlets to Boiler Houses for Cinemas and Theatres

Seating capacity	*Area of air inlet in sq. ft.*
1,000	1·5
1,500	2·25
2,000	3·0
2,500	3·75
3,000	4·5
4,000	6·0
5,000	7·5

It should be remembered that this ventilation is not required for the sole purpose of providing a reasonably cool atmosphere free from fumes in the boiler house, but also to provide sufficient air for combustion. If this provision is not made the boiler draught is likely to prove inadequate.

Accessibility to the boiler house is a desirable feature, and, where the planning permits, some means of access should be provided from the interior of the building, for the cinema and theatre boiler-house plant is invariably looked after by a man who has other duties, and it is therefore not desirable that he should have to leave the building to obtain access to the boiler house. In all cases, however, access should be provided to the exterior as a safety precaution in the event of any emergency necessitating immediate escape.

In considering the requirements of the air-conditioning and ventilating system we must first bear in mind what constitutes a good system. This will consist of two independent systems, the one being the fresh-air system and the other the extract. In the case of the fresh-air inlet system an electrically driven centrifugal fan is provided which induces a flow of air through heater batteries and an air-washing chamber before delivering it through ducts into the auditorium. The extraction system consists only of an electrically driven fan extracting air through ducts from conveniently arranged extraction grilles. The disposition of fan chambers is a simple matter, and by far the most important point is the area provided and the means and space for carrying away main fresh-air and extraction ducts.

In some cinemas it is possible to arrange the plenum chamber, in which the air-conditioning plant is situated, at a corner of the building near to the stage, at any convenient level. In some instances the fan chamber has been arranged under the stage, whilst there are many examples of fan chambers constructed on the main roof of the building. The following are the space requirements of air-conditioning and extract ventilating plants for various sizes of cinema or theatre, together with the area of fresh-air inlet louvres required to the plenum chamber. The height of the fan chambers, it will be noticed, is also stated.

Space Requirements for Cinema or Theatre Air-conditioning and Extract Plants

Seating capacity	*Area in sq. ft. for air-conditioning plant*	*Height (minimum), ft.*	*Area of inlet louvres, sq. ft.*	*Area in sq. ft. for extract plant*	*Height, ft.*
1,000	120	7	30	80	7
1,500	220	9	45	95	9
2,000	280	10	60	110	10
2,500	360	11	75	125	11
3,000	400	12	90	140	12
4,000	520	13	120	170	12
5,000	640	13	150	200	12

With many of the cinemas built on competitive lines at the present time it is usual to introduce fresh air through a large grille situated each side of the proscenium opening, and, indeed, this method of air distribution holds good for all cinemas up to 1,000 seating capacity. With regard to the necessary area, it may be taken that for an air-conditioning system to comply with any existing regulations the provision of 7 sq. in. free area per person for these grilles will be adequate. In the preliminary planning of the building it is also useful to have some idea of the areas of the main duct leading from the air-conditioning and extraction plants. In the case of the air-conditioning plant $2\frac{1}{2}$ sq. in. per person are required, whilst for the extract plant 2 sq. in. per person will suffice.

It should be pointed out that whilst the information given is intended to be of assistance to the architect in his preliminary planning, it is very desirable that the assistance of a consultant be obtained in the early stages, as it is only by this means that a perfectly designed plant can be obtained. To leave the heating and ventilating equipment as one of the last points to be considered leads only to the installation of plants which, though they may not give cause for complaint, would on careful test prove to give inadequate ventilation and faulty air distribution.

The author has found by long experience that the various requirements of cinemas and theatres in particular are capable of standardization in the form of graphs.

The weather conditions of recent years have proved conclusively that the modern cinema should be provided with air-conditioning and cooling equipment capable of maintaining comfortable internal conditions during excessive outside temperatures. There has been a growing tendency during the last few years to neglect this type of equipment, principally owing to its first cost, which is a matter which seems to come before everything with competitive cinema building. It is too late, unfortunately, to contemplate installing cooling equipment when the weather conditions make its use desirable.

The maximum external temperature which has been experienced has been in places 85° F., which is the basis of temperature on which cinema-cooling equipment would normally be designed in this country, so that a cinema without cooling equipment may well be said to be unfortunate. It is usual to find, almost without exception, a considerable fall in temperature during the night hours. This fact can be made some use of in any cinema which has a plant capable of introducing fresh air to the auditorium.

With a crowded cinema there is an enormous amount of heat emanating from the occupants, and, to make matters worse, during excessively warm weather the temperature of the external air is high so that in many cases its use as a cooling agent becomes futile. It is customary to run the ventilating equipment only during those periods when the house is occupied. In view of the low night temperatures, it would be definitely advantageous at these times to run the plant continuously during the night until, perhaps, 10 o'clock in the morning.

Not only will this result in the auditorium air being definitely cool, but, what is more important, the structure and internal fittings are given the opportunity of cooling down considerably. It is not suggested that the plant should be run between 10 o'clock in the morning and 1 o'clock, when the normal house is opened, for during this period the external atmosphere will have increased in temperature and running the plant would only result in undoing the benefits that have been obtained during the night. When the ventilating plant is again started up at the opening of the house it is necessary to maintain pure internal conditions apart from any question of temperature.

Most modern cinema-ventilating plants are provided with air-washers, and we need hardly say that these are necessary during summer weather conditions; and where the suggestion of running the plant through the night is adopted, the air-washer should also be in use. It must be remembered that with an external temperature of 85° F. and a relative humidity of 60 per cent. it is possible to cool the air in the air-washer, recirculating the spray water, to approximately 74° F.

By using water in the spray chamber direct from the company's supply it is possible to cool air to a temperature as low as 65° F. But the water shortage which on many occasions accompanies the heat-wave conditions would not justify using such large quantities of water and running them to waste. Some cinemas are, however, situated adjacent to factories or other undertakings normally using large quantities of water, and in these cases it would be a comparatively simple and rapid matter to connect the mains supply to the air-washer sprays, rearranging the air-washer pump to deliver the water, after it has been used for cooling the air, to temporary storage tanks in an adjoining works. A similar procedure could be followed if the cinema adjoined a large public building, in which case the water could be delivered to the tanks normally providing water for sanitation purposes.

Where the cinema owner is not fortunate enough to possess an air-washer there are several things which could possibly be done to assist in cooling the auditorium. In some cases the plant is provided with filters of the viscous or similar type consisting of small filter cells arranged in a frame, the air passing through these cells. When the plant is run during the night hours it is possible to remove these filters, which would result in an increased output from the plant due to removing the resistance to the filter cells. This is a procedure which should not be carried out without advice, in case the increased output is likely to overload the motor driving the fan. It may be said that removing the filters will allow impure air to pass into the building, but there are very few cases where this would prove to be deleterious for such short periods as we have in mind, and, in any case, during the night hours dust is not so prevalent.

In the case of plants which have no filters the arrangement may be such that it is a comparatively easy matter to arrange for a by-pass so that the air does not pass through the heater battery which is used during the winter months; this would remove resistance and lead to increased capacity.

The provision and use of ceiling fans is another matter which could easily be arranged for. The high-velocity movement caused by the use of these fans, which need only be temporary fixtures, produces a considerable cooling effect, in many instances equivalent to several degrees drop in temperature.

The use of ice for cooling the air entering the auditorium has often been discussed and abandoned as an expensive matter, whilst such experiments as have been tried have had very little measure of success owing to their not being properly applied. It is not necessary to provide a temperature in summer as low as that normally maintained in the auditorium during the winter months, and if the air-conditioning plant can maintain a temperature 10° F. less than that prevailing outside, a definitely comfortable effect is achieved. For example, a cinema with a seating capacity of 1,500 people would, during extreme conditions, have the following sources of heat tending towards a rise in temperature in the auditorium:

	B.T.U. per hour
Heat of occupants	600,000
Lighting and power, average	15,000
Transmission through structure	50,000
Radiation from sun	80,000
Total	745,000

It is probable that the amount of air being introduced by the air-conditioning plant is 1,500,000 cu. ft. per hour, and as 1 cu. ft. of air absorbs approximately 0·02 B.T.U. in rising 10° F., the air must be introduced

$$\frac{745{,}000}{1{,}500{,}000 \times 0{\cdot}02} = 25^\circ$$

approximately below the desired internal temperature. In other words, to provide an atmosphere internally which is 10° F. cooler than it is externally, the air must be introduced 35° F. cooler than the external temperature, that is, at about 50° F.

Under these conditions the approximate total amount of heat to be removed in cooling the air to 50° F., ignoring the question of humidity, is $1{,}500{,}000 \times 35 \times 0{\cdot}02 = 1{,}050{,}000$ B.T.U. per hour. The amount of heat absorbed by 1 lb. of ice rising in temperature to 50° F. is 162 B.T.U., so that $\frac{1{,}050{,}000}{162} = 6{,}500$ lb. per hour, or about 3 tons in all, would be required. This may seem a large amount, but, as will be seen later, it is cheaper to use than a refrigeration plant for the same duty. There are several ways in which the ice may be employed, either by use in the air-washer tank, in a separate spray-water cooling apparatus, or for direct-air cooling by passing the current of air over the ice.

Considering this last method, it would be necessary for there to be an exposed area of about 2,000 sq. ft. of ice for average conditions in order to be able to melt sufficient of it to absorb the necessary heat. According to the author's investigations, the rate of melting varies enormously with the velocity and humidity of the air passing over the surface. It is possible, therefore, to have too much surface exposed and to cool the air to too low a temperature, which would be wasting ice. It will usually be found that with direct cooling by ice it is advisable to crush the ice and to have an apparatus in which four hours' supply can be exposed to the incoming air.

It will be found that the period in which ice-cooling is absolutely essential to comfort is rarely in excess of 150 hours at full load, calling for the use of 450 tons during the season, which would cost about 30*s*. per ton if a general demand existed for ice for cinema-cooling, or a total cost of £675 per season. The cost of the equipment for enabling ice to be used efficiently would be, perhaps, £150. The annual costs would be approximately as follows:

Cost of ice	£675
Depreciation, etc., on plant per annum . . .	£10
Total	£685

Assuming, therefore, that this must be recovered in a period of 50 days, and taking an average price of 1*s*. 6*d*. per seat, just over 180 seats per day would need to be sold to recover the cost of cooling. With judicious propaganda this should not be difficult.

It may be considered that ice-cooling is not capable of use owing to the cost of running, but when it is remembered that the capital outlay of a refrigeration plant to do similar work when required would be at least £5,000 (and several buildings in this country and many abroad are equipped with them), ice-cooling is surely a proposition.

If the refrigeration plant is installed it is essential to have the services of an engineer to run the plant. The annual costs would be approximately as follows:

Depreciation and repairs on plant . . .	£250
Engineer's wages, per annum	£225
Cost of power and water	£350
Total per annum	£825

Moreover, few cinemas are able to face the additional expenditure involved in a refrigeration plant, but they would be able to consider the expense of purchasing ice to meet definite requirements.

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