FACTORY STEAM PLANT

A PRACTICAL GUIDE TO FUEL ECONOMY

BY

G. E. H. LEWIS

M.INST.F., A.M.I.E.E.

WITH A FOREWORD BY

JOHN D. TROUP

M.I.MECH.E., M.I.CHEM.E., F.INST.F.

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FOREWORD

During the late war period an extensive fuel economy campaign was inaugurated, and operated throughout industry. The Fuel Efficiency Committee of the Ministry of Fuel and Power played a leading part in this campaign.

It is a sad reflection that a Government department found it necessary to take such a course, but in the stress of war it had to be done.

This Committee with its band of engineers—many of them giving voluntary service—did excellent work. They filled many of the inefficient gaps in the use of fuel, and so saved the country many millions of tons of our native coal.

All this work has laid the foundation for further building, and the work is being pushed ahead by the activities of the Joint Educational Committees of the Ministry of Fuel and Power and The Institute of Fuel.

But there is a great deal more to be done, and the author of *Factory Steam Plant* has provided a most valuable guide in bringing together those essential factors which govern fuel economy in the steam plant. Mr. Lewis is in the fortunate position of an author who has had the practical experience of helping to fill the inefficient fuel gaps during the war. He realizes that important fuel savings are to be made, not only in the generation of steam but also in its utilization.

The author deals with his subject from the fuel economy point of view. He has covered a wide field for possible fuel saving, but has concentrated on the points where the greatest efficiency is to be attained. Not only does he deal in some detail with the plant concerned, but he makes clear in simple language the why and how on the operational side.

The days of cheap fuel have passed, and we can no longer afford to tolerate waste in any part of our steam plants. This book is a practical guide to the course that must be followed for the efficient use of fuel, and, when followed, will result in profitable returns that are out of all proportion to the relatively small investment required.

JOHN D. TROUP

PREFACE

DURING the intensive fuel economy drive of the war period, the author, as a part of his duties, visited many factories, widely differing as to type and size, and talked to numbers of engineers and managements. As a consequence, he had a unique opportunity of assessing the attitude of both to their own steam plant. The period could be described as a "Steam Renaissance." Fuel was both scarce and expensive, so that the attention of all steam users was perforce focused on the boiler house and the steam utilization plant.

Industry was suffering from past apathy, bred in the days of cheap and plentiful fuel. The boiler house had always been very much of a Cinderella of factory departments, and when the time of stress came, the revelations resulting from the investigations carried out in the interests of fuel economy were frequently startling.

Many executives, persuaded for the first time into a lively interest in a department hitherto pushed much into the background, expressed astonishment that there was "so much in it." Many others attacked the problems involved with vigour, and carried them through to successful conclusions. All evinced a thirst for further information, an appetite which the Ministry of Fuel and Power did much to assuage.

Operatives too, were equally enthusiastic, especially where they were assured of the wholehearted backing of their managements. Such information as could be applied to their own particular case was eagerly devoured. Much was done; much remains to be done.

In far too many cases bad original selection of plant, and badly planned extensions to a growing factory, have resulted in a state of affairs that only complete and drastic revision can alleviate.

Certain it is that coal will never be cheap again, or at least it may be said with safety that the pre-war level of prices will never be attained. Also, the supply position—a matter of grave concern during the war—has worsened rather than improved, and he would be bold indeed who dared prophesy abundance of solid fuel for at least the next five years.

In the meantime Industry must live, and both from the standpoint of cost and availability of coal, the boiler house can never again be relegated to the position of a conveniently forgotten adjunct to the maintenance department.

Efficiency must take the place of expedients, and improvization must give way to ordered planning.

It is to be of some assistance to engineers and executives alike that the present work is written.

Inevitably, in a book of this nature considerable space must be taken up by descriptive matter, for the simple reason that the correct selection of plant is vital to efficient working, and only knowledge of the characteristics of the proposed apparatus, together with an exact appreciation of the conditions to be met with, can enable that choice to be made.

Some of the methods of calculation are unorthodox, but all have stood the test of practical application; all are very quickly carried out and capable of adaptation over a fairly wide range. The endeavour has been to show the principle to be applied rather than to go deeply into detail, although specific examples are given where necessary, to make a principle clear.

The author has been embarrassed throughout by the necessity of keeping the book within reasonable limits, and he is conscious of many omissions and shortcomings.

When it is considered, however, that each of the chapters might well be the subject of a complete work, these unavoidable omissions may be forgiven.

The grateful thanks of the author are due to the many firms who co-operated with him in the provision of blocks and photographs, many of which were specially prepared for the book, and particularly to Mr. L. G. Northcroft, who supplied a complete set of drawings for the chapter on "Steam Trapping"; also to the proprietors of *Cheap Steam* for permission to publish matter written by the author for their pages.

Full acknowledgments are made under the appropriate illustrations. Thanks are also due to Mr. D. Boulsover, for his skill and care in the preparation of many of the curves and drawings.

Finally, the author presents his book in the sincere hope that it may be of real assistance to the works engineer in the solution of some of his problems, and that also it may help the busy executive who, without the time to go deeply into the more technical aspects, may welcome some guidance on steam matters in general, if only from the standpoint of knowing that his own plant is or is not running at reasonable efficiency and cost,

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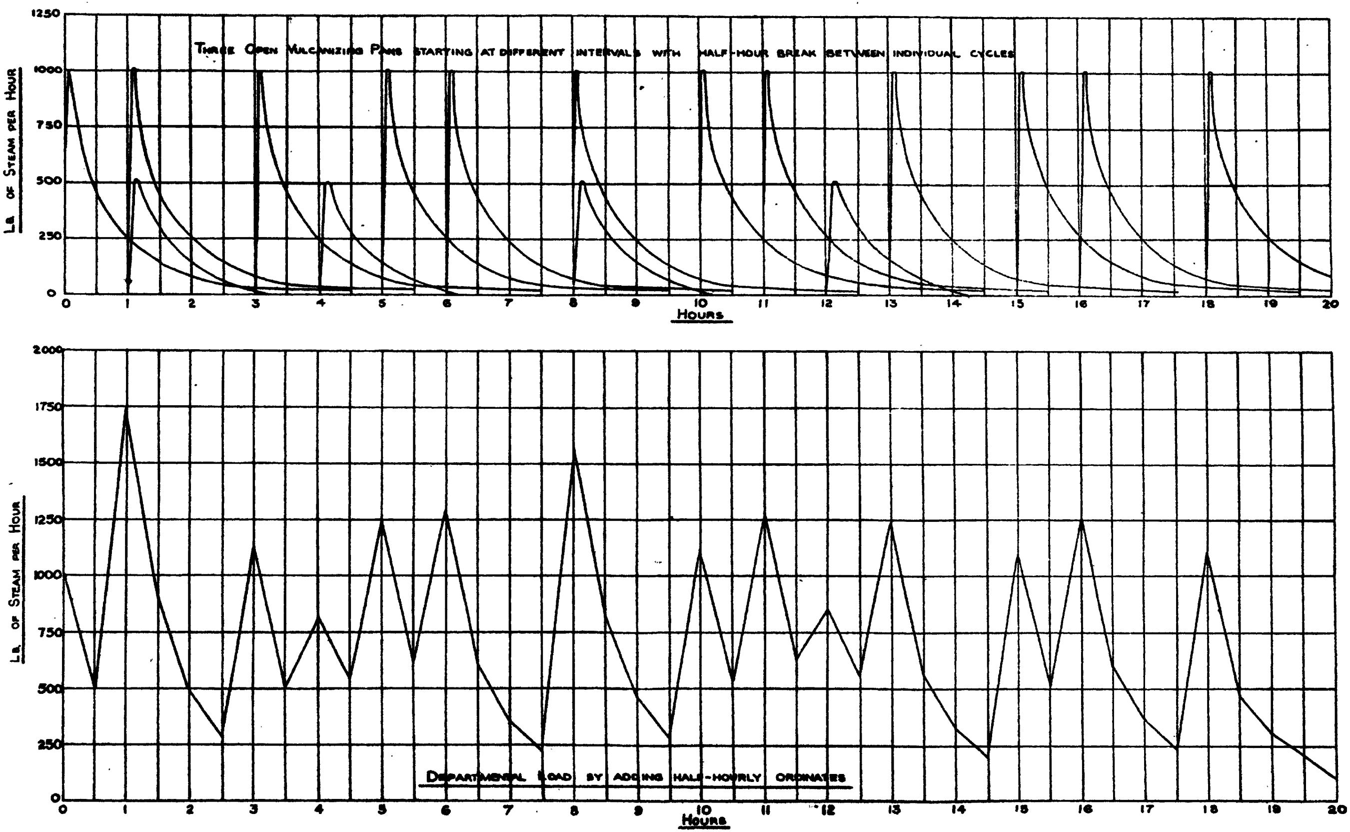

FIG. 1. FACTORY DEPARTMENTAL STEAM FLOW/TIME GRAPHS

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CHAPTER I

CHOICE OF BOILER

IF full and continuous efficiency is to be obtained from steam generating equipment, it is essential that correct selection should be made in the first place. To this end, all possible information should be gathered with regard to the expected load, both as to magnitude and character. It is not only necessary to know the possible maximum steam flow, some idea of which can be obtained from a simple survey of the apparatus which it is required to supply, but also as close an estimation as possible must be made, of the *incidence* of that flow.

Graphical Representation

Processes vary so widely that only a detailed study of the individual case can yield the information required. Methods for estimating the steam absorbed by various classes of apparatus are given in a later chapter, and when these are applied, the results should be plotted on a time/steam flow base, and the graphs for each machine superimposed to form a departmental graph. The departmental graphs, all drawn to the same scale on transparent paper, should then be imposed upon each other (or the ordinates added) to form the final diagram for the factory. Fig. 1 shows the method and the resulting diagram. This work should be carried out in as much detail as possible, especially where violent fluctuations of demand are known to be prevalent. Only in this way is it possible to obtain a sufficiently accurate picture of the probable load on the steam plant, and since the final result is perhaps the most important of all the factors which influence the selection of the boiler itself, care and accuracy at this early stage will be well repaid.

Load Estimation and Capacity

Where the boiler is required to supply steam for prime movers, study of the load is of equal importance but easier to obtain.

The makers' steam consumption curves should be studied with care, and the steam load characteristic assessed from the electrical or mechanical loads on the machine. Naturally, the capacity of the boiler will be determined by the size and steam consumption of the machine it is required to serve, but the type of load will have some influence on the design. Therefore some knowledge of the steam characteristics of the prime mover at all loads is of advantage, even when the boiler is in

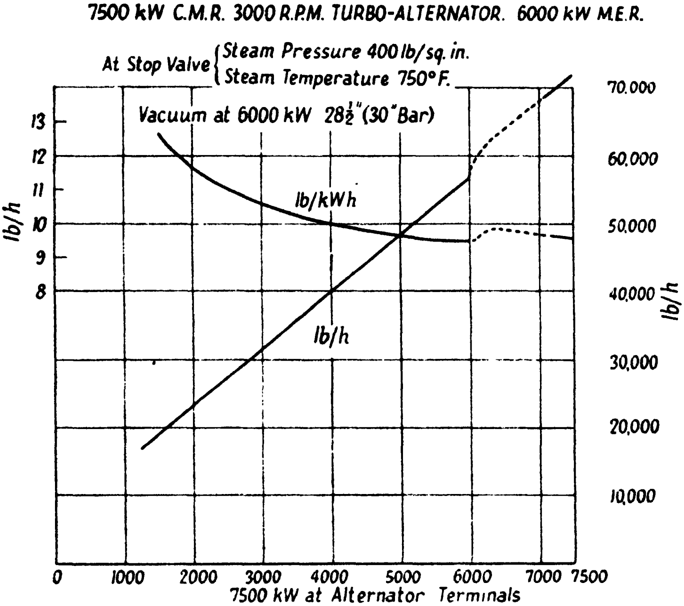


FIG. 2. CONDENSING TURBINE STEAM CONSUMPTION CURVES
(*Fraser & Chalmers, Erith, Kent*)

the project stage. Figs. 2 and 3 show typical steam consumption curves of turbines and reciprocating engines running fully condensing.

Where the whole factory is in the project stage, the matter of load estimation is frequently one of some difficulty, if only for the reason that the steam engineer is at the mercy of the production specialists, who possibly may have no very clear idea of what they really want until too late in the day. At the best only a very approximate estimation can be made, based on similar lay-outs elsewhere, but it is obviously best to plan for ultimate development if such development is likely to approach completion in a reasonable length of time, and if financial considerations will permit.

Pressure

Magnitude and character of load being determined, pressure is the next factor to be considered. If this is finally determined at above 300 lb per sq in. certain types of boiler are eliminated at once, and the field of choice is narrowed. Where the steam is required for process work only, the decision as to pressure may be fairly straightforward, involving as it does only the determination of the maximum pressure required by the process. This in its turn is influenced, or even determined by

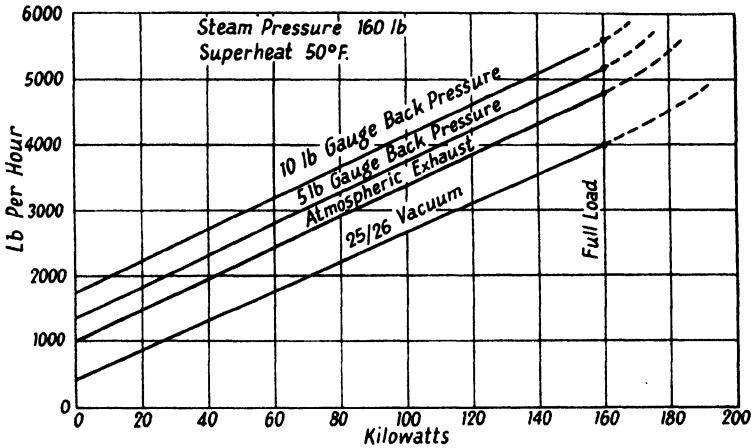


FIG. 3. RECIPROCATING ENGINE STEAM CONSUMPTION CURVES
(Bellis & Morcom, Birmingham)

such critical temperatures as the process may demand. If the temperatures are not critical there is greater latitude of choice, but careful consideration must still be given, and for the following reason.

For process purposes low pressures are advantageous because of the greater latent heat available from each pound of steam. But lower pressures mean larger process pipes for a given heat capacity, and larger capital expenditure on this account. It does not follow, therefore, that the most desirable pressure from the process standpoint is the most economical. Again, the sizing of the pipes themselves is affected by the character of the process, and in certain cases it may be possible to allow considerable pressure drops, and so reduce expenditure. Pipe sizing is dealt with in more detail in a later chapter.

Combined power and process schemes present problems of much greater complexity, and full consideration must be given to such factors as steam quantity and state at the process supply point of the prime mover, whether engine or turbine, and the balance between the power and process loads. Where completely new plant is visualized the design of the prime mover will be taken in conjunction with that of the boiler, for the selection of the correct pressure and steam state at inlet is vital to the subsequent economy of the scheme. These considerations are discussed in the chapter on prime movers.

Space Factor

After capacity, type of load, and pressure, the next greatest influence on boiler selection may well be available space. This factor is naturally of primary importance where the new plant has to be accommodated in existing buildings, or is taking the place of boilers which have become obsolescent. Not only the boiler itself but all its auxiliary plant is affected, and the space required by a large shell-type boiler together with economizer, etc., may militate against an otherwise probable choice. Much depends also on the possible disposition of the plant, and at this stage yet another factor obtrudes. The quality of the feed water, and the necessity or otherwise of softening plant, not only influences the selection of the boiler directly but is also of importance when space is considered.

The class of coal to be burned may not affect choice of boiler, at least so far as any one main type is concerned, but details of design, of stokers, grates, and flue ducts cannot be fully drafted until an analysis is available. This, however, is a matter for the stoker designer rather than for the engineer responsible for the selection of the boiler.

Maintenance

Lastly, before the final decision is made, the maintenance resources of the factory concerned and the possible effect of the proposed boiler should be considered. This point is not always fully appreciated. The term "Boiler" in these days embraces far more than has been the case in the past. Even that ubiquitous maid of all work, the Lancashire boiler, is often appearing in new guise, fitted with balanced draught, economizers, and air preheaters—a far different conception from the too common idea of latter years, of the plain unvarnished shell with flue tubes, whose greatest virtue was its simplicity, but whose efficiency frequently did not approach 50 per cent.

Such results cannot be tolerated nowadays, and in the quest for efficiency, accessories which not so long ago were regarded as luxuries, and even frowned on by the more conservative, are now provided as a matter of course. But each additional item of equipment, necessary though it may be, throws more responsibility on the maintenance section, who more often than not are sufficiently hard pressed already.

Certain modern boilers, while capable of very high efficiencies, can only be kept in optimum condition by skilled personnel, backed by suitable equipment. Not every factory, however, is in a position to provide either the staff or the tools necessary, and the choice of a highly sensitive boiler in such cases would be of doubtful wisdom, even if a particular design is attractive from other considerations.

After studying the various points, pressure, type, and magnitude of load, feed water, space available, and maintenance resources, the actual choice of the boiler can be proceeded with systematically.

Selection of Boiler

For pressures below about 180 lb per sq in. and loads below about 3000 lb steam per hour, either a Cornish, or one of the many reputable vertical boilers would receive consideration. If the load is fluctuating the Cornish type would have preference by virtue of its greater water capacity. Should floor space be limited the choice of the vertical boiler might well be obligatory, but if the load surges are sufficiently violent a larger size than is demanded by the evaporation should be put in. The vertical boiler also has the advantage that the foundations are simple, and no brick flues need be built. On this score, the total expenditure may be less than that for the Cornish. There are, however, many types of vertical boiler, and in differentiating between them the quality of the feed water must be taken into account. The vertical cross tube boiler allows a good deal of latitude in this respect, but the thermal efficiency is low, and for this reason should be considered only for the smallest outputs. Both the vertical cross tube boiler and the Cornish boiler may be operated at need by unskilled or semi-skilled labour. For outputs of round about 2000 lb of steam per hour the vertical smoke tube boiler or the vertical water tube boiler would probably be preferred. Both these types will operate at quite reasonable efficiencies (about 70 per cent), but both need good feed water. The vertical boiler with vertical smoke tubes suffers from the disadvantage that the ends of the tubes are exposed to the radiant heat of the fire, and if served with a scale-forming water, trouble with tube ends is bound to occur. The type of boiler with horizontal smoke tubes does not suffer from this particular defect, but scale-forming waters will form harmful deposits on the furnace crown. Moreover, with this design the space factor is not so essential because some floor space must be allowed for tube cleaning.

With the vertical water tube type less damage will be done by poor feed, but tubes may be hogged, and in any case, a scale-forming or sedimentary water makes more frequent cleaning an absolute necessity.

All types of tubular boiler require a higher degree of skill for operation and maintenance.

Pressures up to 250 lb per sq in. and outputs between, say, 3000 lb per hour and 12,000 lb per hour could be met by Lancashire, water tube, or one of the many boilers of the Economic type. Above about 8000 lb per hour there is a special type of high efficiency, self-contained boiler which combines some of the advantages of the Lancashire boiler, with the responsiveness of the water tube. This is the Super Lancashire boiler, which is treated in detail with other boilers in a subsequent chapter.

Again, for fluctuating loads, the boiler with the largest water capacity would be the first choice, and this would naturally fall upon the Lancashire. But for the outputs now under consideration economizers, and probably induced draught, would be obligatory, so that apart from the consideration of capital cost of settings, buildings, etc., the question of space becomes of first importance. It may be said at once that from the standpoint of load fluctuation only, the Lancashire boiler has all the advantages, and where space is of little moment might well be installed without further discussion. This boiler is still a great favourite for industrial work, having much to recommend it on the ground of simplicity, but the large space demanded by the complete plant, including economizers, is one great objection to its use to-day. For steady loads, where the advantage of its large water capacity is of small account, first consideration would be given to plant of a more self-contained type and of greater inherent efficiency.

There are many forms of horizontal tubular boiler, each with some special point of design peculiar to itself, but all definable under the general heading of Economic Boiler. These boilers are built for pressures up to 300 lb per sq in. and for capacities as high as 30,000 lb per hour. They combine a fairly large water space with high efficiency, without the use of economizers. Figures of 73 to 75 per cent are quite usual, and under good conditions and with skilled firing may be even higher. While not so good as the Lancashire boiler for fluctuating loads, the Economic boiler is frequently resorted to as a solution to the space problem. In common with all tubular boilers, they must be fed with good water, and negligence in this respect must inevitably lead to troubles and high maintenance. Foundations are simple; in many cases the boiler is put down on cradles on a level floor. The brick set Economic boiler is seldom seen nowadays, preference being given to the more easily installed self-contained internally fired type.

Owing to their high capital cost, it would probably be difficult to justify the use of water tube boilers for the particular conditions under discussion, although there are very many instances of their use, particularly in conjunction with prime movers. .

Above 300 lb per sq in. and for the highest evaporations, the water tube boiler comes into its own. There are three main types: the straight tube, the bent tube, and the forced circulation. In all types, great flexibility of design is possible, and almost any condition can be met. Economizers, air preheaters, and other auxiliary plant can be disposed to take up the minimum floor space, but considerable headroom is usually necessary. The water content is small in proportion to the evaporative capacity; in fact, present day designs are approaching closely to the conception of the true "flash" boiler.

Good quality feed water is necessary, and softening plant or water treatment is usually obligatory. Skilled operation and maintenance must be available if the best results are to be obtained. Very high efficiencies are designed for, and for large units under skilled supervision figures as high as 88 per cent have been attained.

Table I is a summing up in convenient form of the characteristics of the main classes of boiler likely to be found in industry, and the duties to which they can best be applied. It will be

TABLE I

Type of Boiler	Pressures, Lb/sq in.	Evaporation, Lb Steam per Hour	Space/Capacity Factor	Suitability for Fluctuating Loads	Feed Water	Efficiency, per cent	Maintenance
Vertical cross tube	30–120	200–2500	Occupies small space but inherently low capacity	Fair if designed with good water space	Some latitude allowable	Usually below 60	Reasonable care
Orthodox vertical boiler with vertical smoke tubes	Not normally over 150; Special Designs to 250	300–6000	Good	Fair only; should be ordered larger for this duty	Good feed essential	65–67	Routine care and regular cleaning
Vertical water tube	120	370–6000	Good	Fluctuation should not exceed 20 per cent of full capacity	Good feed essential	68–70	Routine care and regular cleaning
Normal single pass economic	300	Up to 4000 single flue; 30,000 double flue	Good	As above	Treatment usually essential	70–75	Skilled attention
Lancashire	260	12,000	Very bad; large area required	Very good owing to large water space	Some latitude allowable, but good feed desirable	63, boiler only, up to 75 with economizers	Routine care and cleaning
Cornish	160	4000	See Lancashire	See Lancashire	See Lancashire	60–63	See Lancashire
Water tube	All commercial pressures	Any, up to 1,000,000	Very high; some headroom usually necessary	According to design	Treatment essential	78–88 or more	Skilled supervision essential

This table is intended as a guide to rough selection only. Special types of boiler are not dealt with. Fuller details of most commercial types are to be found in the text.

realized that no hard and fast rule can be established, and that each problem must of necessity be dealt with on the spot, taking into account all the circumstances peculiar to the particular industry. The table is helpful as a guide only. The most vital work in connection with boiler selection is not so much the choice itself but the accurate assessment of the work the plant will be called upon to do. When selecting a boiler, circumstances more often than not compel a compromise, but if the preliminary survey has been well carried out the engineer can at least be certain of ordering a boiler which is well up to its job, even if non-technical considerations (space or finance) force him to a choice against his first inclinations.

CHAPTER II

BOILER TYPES

THIS chapter is devoted to brief descriptions of some of the more representative types of each main class of boiler.

Vertical Boilers

These may be sub-divided into the following categories:

The Vertical Cross Tube Boiler

Fig. 4 shows a typical boiler of this class. The capacities range from about 200 lb of steam per hour to 2500 lb per hour, and standard pressures are about 30 to 120 lb per sq in. The boilers are designed for natural draught, and under these conditions they will burn from 15 to 20 lb of coal per sq ft of grate area per hour. The evaporation rate is based on 5 to 8 lb of water per lb of coal, from and at 212° F. The total heating surface may vary in different designs and sizes, from eight to ten times the grate area. The height of the firebox is influenced by the class of fuel burned, but is usually from 1·4 to 1·85 times the diameter. It will be seen from Fig. 4 that the grate area is limited by the diameter of the firebox; from this dimension a rough estimation of the duty of a boiler may be made.

EXAMPLE.—Assume a vertical cross tube boiler with a grate diameter of 4 ft. The area is approximately 12 sq ft. If, say, 15 lb of coal per sq ft per hour are burnt, the coal consumed per hour is $15 \times 12 = 180$ lb. Assume 6 lb of steam per lb of coal (from and at 212° F), then the duty of the boiler is $180 \times 6 = 1080$ lb per hour.

The overall height varies with different designs, but is usually such that the steam storage space is approximately 2·5 cu ft for each 100 lb per hour of rated capacity.

The crown plates of the firebox and shell may be either flat or dished, the latter being preferred for the larger diameters. Where flat plates are used for diameters over about 3 ft 6 in., care must be taken to see that they are well stayed.

The number of cross tubes fitted is arbitrary, and is usually chosen by the designer according to the diameter of the boiler.

The gases from the furnace pass through the region of the cross tubes with but little turbulence, so that the heat transfer to the tubes is low. The flue gas exit temperature is therefore

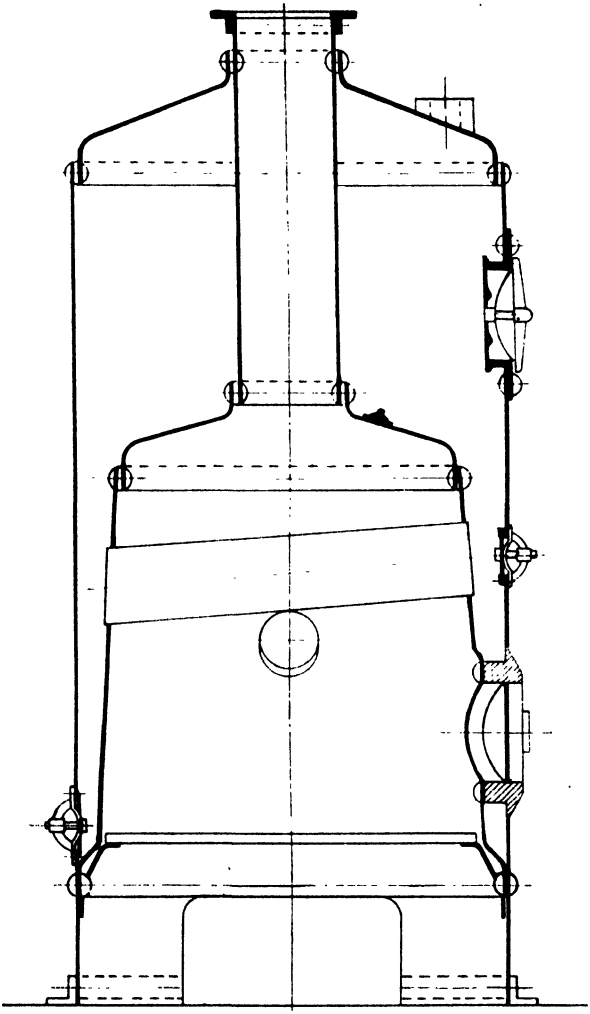


FIG. 4. VERTICAL CROSS TUBE BOILER

high, so that the uptake is subjected to very severe conditions. It is essential that the uptake should be of ample thickness, usually not less than $\frac{3}{8}$ in. The larger sizes, where the uptake diameter is greater than about 8in., are sometimes fitted with a cast-iron internal liner. Particular attention should be given, during examinations, to the water line at the uptake, where corrosion is liable to take place with great rapidity.

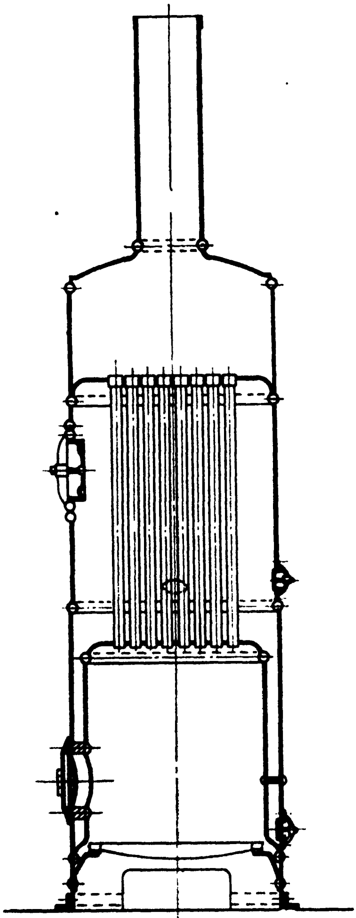


FIG. 5. VERTICAL SMOKE TUBE BOILER

When well designed and constructed, these boilers will stand a certain amount of rough treatment. They are not efficient. About 50 per cent is the most that can be expected, but for small temporary supplies of steam their simplicity has much to recommend it. No special foundations are needed; all that is required is a level floor capable of bearing the weight. The smaller sizes are readily portable, and on this score alone they have been the means of solving many problems of supply, where efficiency is not a major consideration.

The Vertical Boiler with Vertical Smoke Tubes

The low efficiency of the vertical cross tube boiler led to the development shown in Fig. 5. These boilers are usually designed for pressures not greater than 150 lb per sq in. but for special purposes higher pressures up to 250 lb per sq in. can be provided for. Capacities range from 300 to 6000 lb of water per hour. The design provides for a number of tubes which are expanded into the flat furnace crown at their lower end, and into a tube plate at the other. The top tube plate is riveted to the boiler shell, which extends beyond the plate to form the smoke box. The stack may be taken out from an elbow attached to the shell plate, in which case a baffle is provided to protect the top crown plate, and to divert the gases in the desired direction. It is, however, more usual to pierce the crown plate and fit the stack in a central position.

Heat transfer is greatly improved, because the gases are split up into thin streams, and the velocity through individual tubes is higher. In general, a lower firebox is provided so that the tubes may be of effective length. In the conventional design the ratio of firebox height to diameter is of the order of 0·8 to 1·0, as compared with 1·4 to 1·85 for the cross tube boiler. Because of the improved heat transfer higher combustion rates are permissible, and with suitable draught conditions 25 lb of coal per hour per sq ft of grate area may be burned. The total heating surface ranges from 12 to 25 times the grate area, differing with size and maker. Efficiencies of 65 to 67 per cent are realized under normal factory conditions, becoming higher with good fuel and clean heating surfaces. Provision for superheating can, if desired, be provided by a coil type superheater installed in the smoke box. About 80° to 100° of superheat may be added in this way.

Reference to Fig. 5 will make it clear that the lower ends of the smoke tubes are exposed to the radiant heat of the furnace. For this reason, if for no other, it is essential that these boilers should be supplied with good feed water. Fillets of scale forming at the expansion of the tubes into the bottom tube plate (which in the conventional design is extremely difficult to clean properly) must inevitably lead to overheating and consequent damage. In an effort to overcome this difficulty certain makes have the tubes so grouped that cleaning is made easier. Not only must feed water be good from the scale-forming standpoint but also corrosive tendencies must be watched. The water line at the smoke tubes is particularly liable to be attacked in this way, unless steps can be taken to control the quality of the feed.

The smaller sizes of this boiler are as easily portable as the cross tube boiler, and the same remarks with regard to the simplicity of foundations may be made.

Vertical Boilers with Horizontal Smoke Tubes

A well-known example of this type of vertical boiler is shown in Fig. 6. This is a particularly fine example of its kind. It is built for normal capacities, when firing coal, of from 250 to 6000 lb of water per hour, and can be obtained for pressures of 250 lb per sq in. The firebox is a single steel pressing, and therefore seamless; it is of the strongest possible construction for the stresses it is called upon to bear, and its shape is particularly advantageous for the absorption of the radiant heat from the furnace. The convection heat surfaces are provided by the nest of tubes through which the gases must pass on exit from the firebox. The shape and construction of

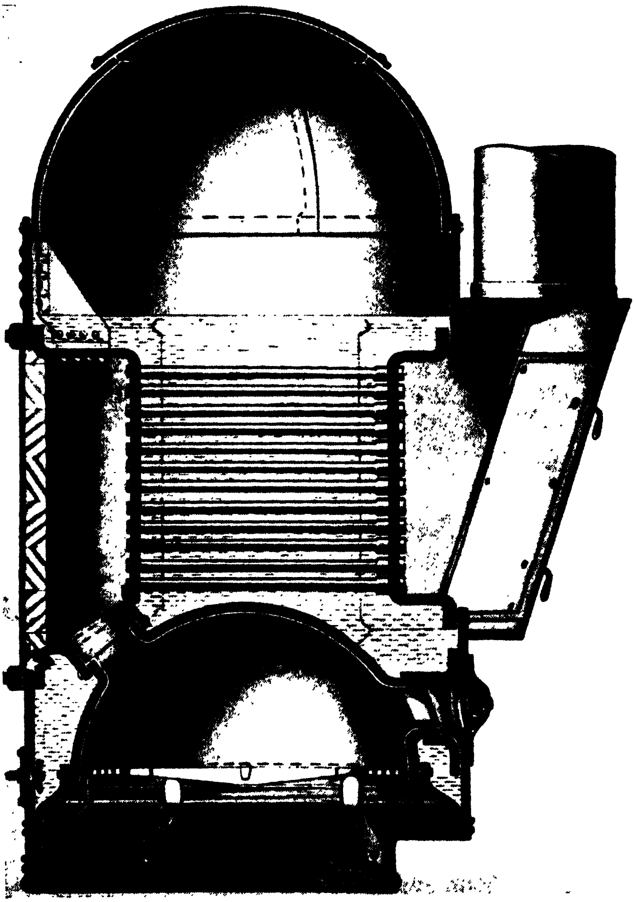


FIG. 6. COCHRAN BOILER
(*Cochran (Annan) Ltd.*)

the firebox make the boiler peculiarly adaptable to various forms of fuel such as wood or oil. For wood, a special well-grate can be fitted, but for this fuel it is recommended that a boiler about 30 per cent larger than the size which would be chosen for coal should be installed.

The severe conditions imposed by oil firing are quite safely sustained by this boiler, and in this connection it should be noted that the one-piece construction of the firebox permits a design in which no riveted seams are exposed to the furnace. All types of oil burner may be fitted, and there is a good deal of latitude with regard to the choice of position. With oil firing, greater outputs are obtainable than with coal, but the increase should not be pushed to more than about 20 per cent.

Stay tubes are fitted in the tube nest, the number depending upon the size of the boiler and the pressure for which it is built. A feature of the tubes is that they are made slightly taper for greater ease of fitting. The water line is normally kept about 3 in. above the gusset stays which strengthen the boiler at the horizontal portion above the tube plate.

On leaving the firebox, the gases impinge upon the refractory lined baffle plate, and are directed thence into the smoke tubes. This refractory plate, forming the back of the combustion chamber, has sufficient heat storage capacity to assist the complete combustion of the gases, and is conveniently arranged for removal for tube cleaning. In addition to this, the larger sizes are fitted with a smaller door which is useful for periodic inspection. Should superheat be desired the superheater is fitted in the smoke box, immediately beneath the stack, for low degrees of superheat, or in a specially fitted combustion chamber at the firebox exit where high temperatures are required.

With coal firing, efficiencies in the neighbourhood of 70 per cent may be expected, and with oil, under good conditions, 72 to 75 per cent.

For normal coal or oil firing the foundations are simple. For wood, also, a straightforward type of well-grate is all that is usually needed, but the larger bulk fuels must be accommodated by a larger combustion chamber than can be provided in the boiler itself. In such cases an external refractory lined grate of large area is constructed, and the boiler so placed that the hot gases may impinge upon the hemispherical firebox. In this arrangement some of the advantages of the firebox, such as radiant heat absorption surface are lost, so that the efficiency suffers slightly. But with fuels which cost little or nothing, this is of small consequence.

When installing, it is necessary to provide space external to the boiler for tube cleaning, and this should be at least equal to the diameter of the shell. The horizontal disposition of the tubes, and the fact that the ends protruding through the tube plate are protected from the radiant heat, lessen the possibility of the tube ends burning off as in the case of the boiler with vertical tubes, but the temperatures at the back tube plate above the firebox exit are high enough at full combustion rates to inflict damage unless the plate is kept reasonably free of scale. Also, deposits on the firebox crown will cause not only loss of efficiency, but if severe enough, damage by overheating. Feed water must therefore be as free as practicable from scale-forming and sedimentary matter.

Vertical Water Tube Boilers

The relative effectiveness for heat transfer of water tubes as compared with smoke tubes is well known, but may be mentioned here before describing one of the many examples of water tube vertical boiler.

When hot gases are passed through a tube which is arranged so that the heat in the gases is transferred to a medium external to the tube, the heat so transferred is a function of the diameter of the tube, and the velocity of the gases. In a large diameter tube the highest velocity is established towards the centre, becoming progressively less towards the circumference of the tube where the gas stream is subjected to "drag" by the frictional resistance of the tube walls. There is therefore a tendency for a cooler film to form at the outer borders of the gas stream because the heat in the outermost layer is already given up to the tube, and is not being replaced, first because of the slower movement of the stream at this point, and secondly because of the poor thermal conductivity of the gases. When it is realized that a gas film only $\frac{1}{4}$ in. in depth offers the same resistance to heat flow as $1\frac{1}{4}$ in. of good insulating brick, the point is immediately apparent. The effect is minimized in practice by providing small diameter tubes, or by employing means to increase the turbulence of the gases and so continually break up the outer film.

The total resistance of the tube system, however, imposes limits to the extent to which either of these practices can be carried out. Where the cool medium is inside the tube, and the gases outside, as in the water tube boiler, this particular problem no longer exists because the tube nests can be readily arranged to produce the necessary turbulence, without unduly increasing the resistance. Moreover, there is a scrubbing effect induced by the shape of the tubes themselves, tending to prevent the formation of a stagnant film.

An example of the vertical water tube boiler is illustrated in Fig. 7. This boiler is normally made for pressures up to 120 lb per sq in., and the capacities range from 378 lb water per hour in the smallest size to 4500 lb per hour in the largest. Both these figures are reckoned from and at 212° F, and are given as normal steaming rates. Maximum rates are 500 and 6000 lb per hour respectively. The grate areas vary from 4 sq ft to 33 sq ft. Heating surfaces are 12·75 times the grate area in the smallest size to 15·7 times in the largest. Therefore, in the smallest boiler the evaporation is rated on 7·4 lb of steam per hour per sq ft of heating surface and in the largest 8·7 lb. Referred to grate area, the figures are 94·5 and 136 respectively. All the above are reckoned for normal evaporations, from and at 212° F.

In general conception, the design of the boiler is not unlike the more conventional cross tube boiler but the tubes are smaller in diameter, and scientifically disposed to give the maximum effect. Some portion of all the tubes absorbs heat by radiation, but only the lowest row is

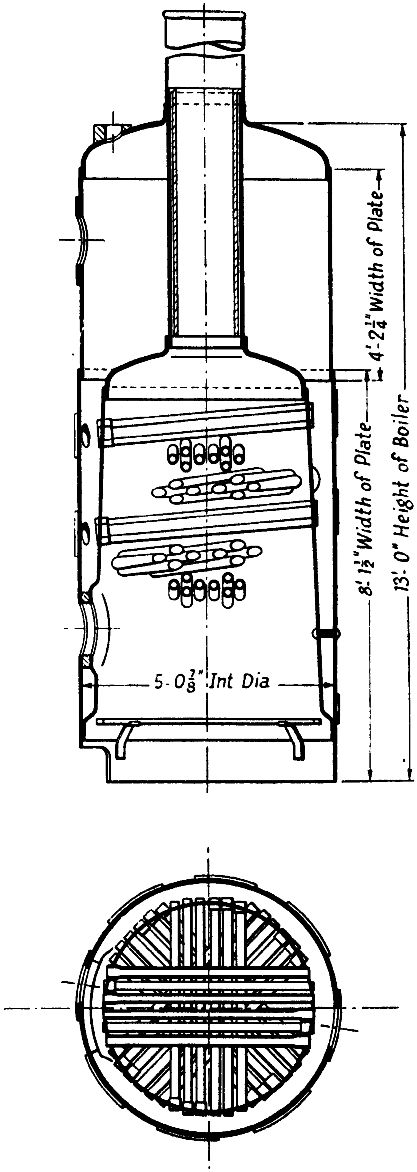


Fig. 7. Vertical Thermax Boiler
(*Ruston & Hornsby, Lincoln*)

directly exposed along the whole of its length. It is therefore at this point that the maximum heat transfer takes place. The number of tubes in each nest varies with the size of the boiler, and the nests are so arranged that the gases are compelled to a swirling motion as they pass through the tube region. The tubes themselves are set at a slight slope, a feature which tends to establish a positive circulation. The water surface is fairly large in proportion to the size of the boiler, so that steam release is unrestricted. Facilities are provided for tube cleaning by well-proportioned hand holes, opposite to each nest. The usual access doors and mudholes are in convenient positions for inspection and sludging. The efficiency obtainable with this boiler under good conditions is 68 to 70 per cent.

Cornish and Lancashire Boilers

These two types of boiler are perhaps the commonest to be found in industry to-day. For evaporations up to 4000 lb per hour, the single-flued Cornish boiler is adequate, but beyond this, and up to about 12,000 lb per hour, the two-flued Lancashire is used. Cornish boilers are normally made for pressures up to 160 lb per sq in., but there is a type of single-flue boiler, fitted with a shortened flue tube and smoke tubes, which can be built for pressures up to 200 lb per sq in., and is capable of evaporations of 6000 lb per hour. Lancashire boilers are in service at pressures up to 260 lb per sq in. but the standard practice does not as a rule allow for more than 200 lb. Tables II and III give typical main dimensions for a range of Cornish and Lancashire boilers respectively.

It will be seen that the ratio of grate area to total heating surface varies in the Cornish boiler from 1 : 21·4 to 1 : 26·4, and in the Lancashire the figures are 1 : 17·7, and 1 : 24. Taking the heating surface as given, the evaporation rates are based on average figures of 8·9 lb of steam per sq ft per hour for the Lancashire, and 5·75 for the Cornish.

While these figures are good enough for rough estimations of duty, they do not at all represent a true picture of what is taking place. Actually, it may be said that about 50 to 60 per cent of the total evaporation is carried out by that portion of the furnace tube immediately over the fire, and which is directly exposed to radiant heat. Since this is only about 15 per cent it follows that the remaining 40 per cent of the evaporation is spread over 75 per cent of the heating surface.

TABLE II

STANDARD SIZES OF LANCASHIRE BOILERS

Size	Diameter of Flues	Grate Area Length	Grate Area Area	Heating Surface	Lb of Water Evaporated per Hour	Indicated Horse Power at 20 lb of Water
ft in. ft in.	ft in.	ft in.	sq ft	sq ft		
20 0 × 6 0	2 3	5 0	22½	400	2,600	130
20 0 × 6 6	2 6	5 0	25	461	3,000	150
22 0 × 6 6	2 6	5 6	27½	514	3,400	170
24 0 × 7 0	2 9	6 0	33	636	4,200	210
28 0 × 7 0	2 9	6 0	33	766	5,000	250
28 0 × 7 6	3 0	6 0	36	838	5,500	275
30 0 × 7 6	3 0	7 0	42	905	6,000	300
30 0 × 8 0	3 2	7 0	44	983	6,700	335
30 0 × 8 6	3 6	7 0	49	1080	8,000	400
30 0 × 9 0	3 9	7 0	52½	1235	10,000	500
30 0 × 9 6	4 0	7 0	56	1350	12,000	600

If the boiler feed water is cold, the evaporation will be reduced by one-seventh
(*Courtesy Messrs. E. Danks*)

TABLE III

STANDARD SIZES OF CORNISH BOILERS

Size	Diameter of Flue	Grate Length	Grate Area	Heating Surface	Lb of Water Evaporated per Hour	Indicated Horse Power at 20 lb of Water	Nominal Horse Power
ft in. ft in.	ft in.	ft in.	sq ft	sq ft			
12 0 × 4 0	2 3	4 0	9	192	960	48	10
12 0 × 4 6	2 6	4 0	10	220	1100	55	11
14 0 × 4 6	2 6	4 0	10	254	1270	63	13
16 0 × 5 0	2 9	5 0	13¾	326	1630	81	16
18 0 × 5 0	2 9	5 0	13¾	354	1770	88	18
20 0 × 5 0	2 9	5 0	13¾	408	2040	102	20
16 0 × 5 6	3 0	5 0	15	342	1710	85	17
18 0 × 5 6	3 0	5 0	15	382	1912	95	19
20 0 × 5 6	3 0	5 0	15	428	2140	107	21
18 0 × 6 0	3 3	5 0	16¼	432	2160	108	21
20 0 × 6 0	3 3	5 0	16¼	484	2420	121	24
20 0 × 6 6	3 6	5 0	18	529	2645	131	26
24 0 × 6 6	3 6	6 0	21	552	3120	156	31

If the boiler feed water is cold, the evaporation will be reduced by one-seventh
(*Courtesy Messrs. E. Danks*)

EXAMPLE.—Consider a Lancashire boiler with a total heating surface of 1350 sq ft and evaporating 12,000 lb of steam per hour. 15 per cent of 1350 is 200 sq ft and this is responsible for 60 per cent of 12,000, which equals 7200. Then this portion which is over the fire is evaporating at the rate of 36 lb of steam per sq ft per hour. The remaining 1150 sq ft of heating surface are therefore evaporating 5000 lb per hour, an average rate of 4·35 lb per sq ft per hour. This last figure will naturally vary, being greatest in the portion of the furnace tubes after the bridge, and least in the side flues.

These figures will be referred to later. In themselves, they are not important at the moment, except that they indicate the type of information necessary for the prediction of flue gas temperatures in various passes of the boiler—a calculation which is to be the subject of further discussion.

The sizing of the grates for both Cornish and Lancashire boilers is conventional rather than scientific. The grate width is determined by the flue tube diameter, which is itself limited by the size of the shell. The length of the grates seems to have been chosen in the past with regard to the distance a man might be expected to throw a shovelful of coal, rather than from any desire to obtain a specific grate area. As a result, it is found that for many conditions the grates of these boilers are far too long, and that better results are obtained by shortening them. This is due less to the fact that it is very much easier to keep the back of the grate well covered than to the beneficial effect of the resulting increase of furnace temperature.

The smaller sizes of boiler suffer too, in that the external flues must be designed not primarily from the standpoint of gas velocity and maximum heat transfer, but must be made large enough for the flue cleaner to crawl along their whole length.

Fig. 8 is a sectional view of the usual form of Lancashire boiler setting. Although this perfectly plain arrangement is to be found in many works in this country, a moment's consideration will show that, in this form, the efficiency must necessarily be low. It has been shown that the greater portion of the evaporation is carried out by a small proportion of the total heating surface, but the gas temperature in the bottom flue may still be of the order of 1000° F. How much of the heat in the gases at this point and in the side flues is given up to useful purpose is problematical, especially when it is remembered that some portion of the available heat is given up to the brickwork. Flue gas temperatures of 700° F might quite easily be attained, and even assuming reasonably good combustion conditions, this represents a loss to the stack of nearly 30 per cent. If for a rough computation the remaining losses are assessed at 12 per cent, the efficiency will be 58 per cent—a figure which cannot be tolerated for a moment for large evaporations and continuous working. It is absolutely

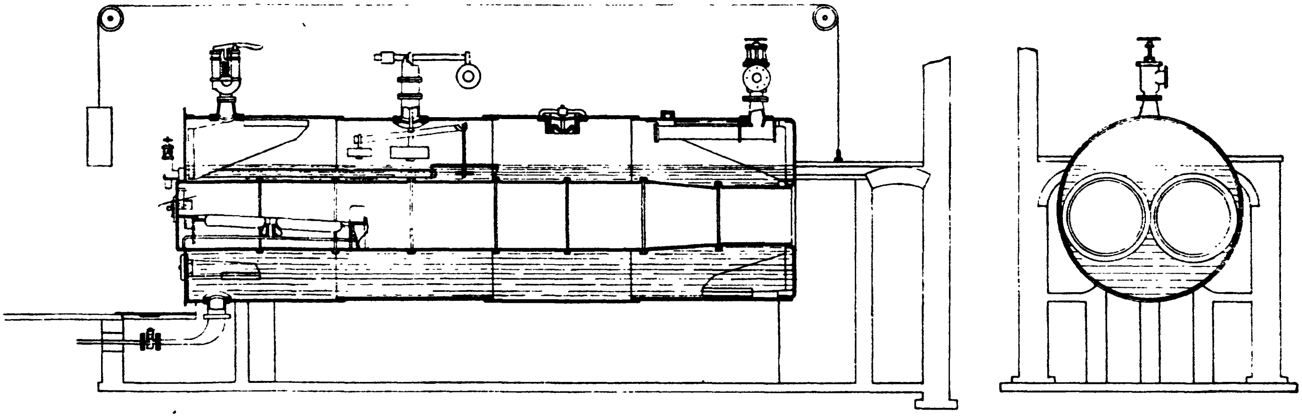


Fig. 8. Section of Simple Setting for Lancashire Boiler

essential, therefore, if reasonable efficiencies are to be realized, that some means must be employed to recover some of the heat in the flue gases. Economizers are the usual solution to the problem, and in recent years air preheaters, either alone or in combination with the economizer have been used. Superheaters are easily fitted to the Lancashire boiler, usually in

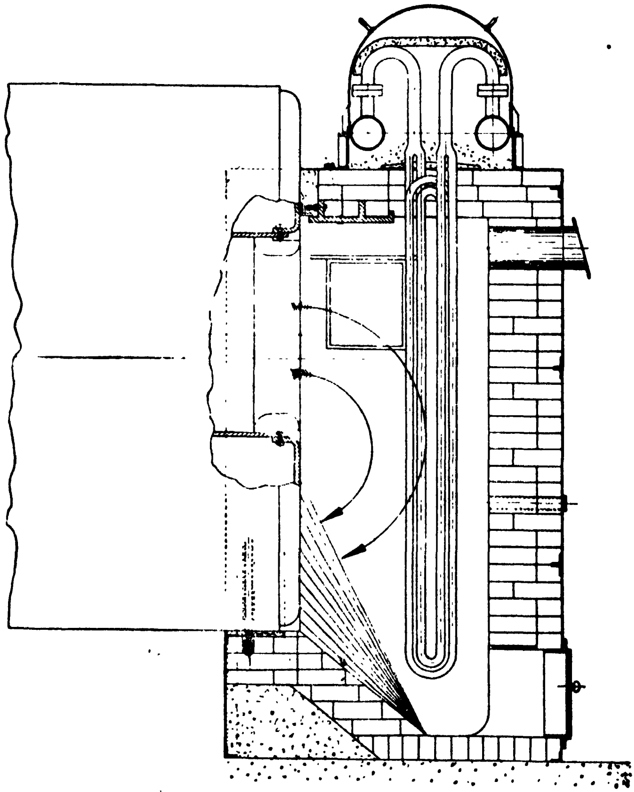


FIG. 9. SUPERHEATER FOR LANCASHIRE BOILER
(*Daniel Adamsons, Hyde*)

the downtake, in which position about 250° F of superheat can be added. Still higher temperatures can be arranged for by fitting the superheater elements inside the flue tubes where they are subjected to a proportion of radiant heat. Fig. 9 shows the more usual form of superheater.

Although in recent years the basic design of the Lancashire boiler has remained unaltered, various manufacturers have developed improvements in detail. Among the most important of these is the dish end. An example of this is shown in Fig. 10. The design enables the gusset stays, which in the flat ended boiler are essential, to be omitted. The adoption of this design overcomes the weaknesses of the flat end, but the plate is very rigid so that special provision must be made to mitigate

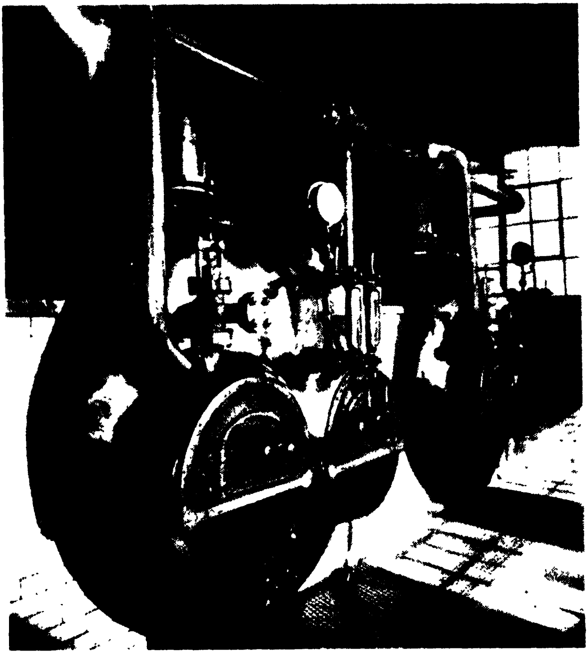


FIG. 10. DISH END LANCASHIRE BOILER
(*John Thompsons, Wolverhampton*)

the resultant stresses set up in the flue tubes. When the conditions under which the flue tubes must work are considered, it becomes evident that the stresses are both severe and complex. Besides the pressure load, which is easily calculated, shear stresses are set up at the point of greatest temperature difference between the upper and lower portions of the tube. The fact that the upper segment, immediately above the fire, expands much more rapidly than the portion below the grate bars gives rise to a tendency for the tube to deform. If deformation actually takes place, bending stresses of considerable magnitude will be established. In addition to all the above

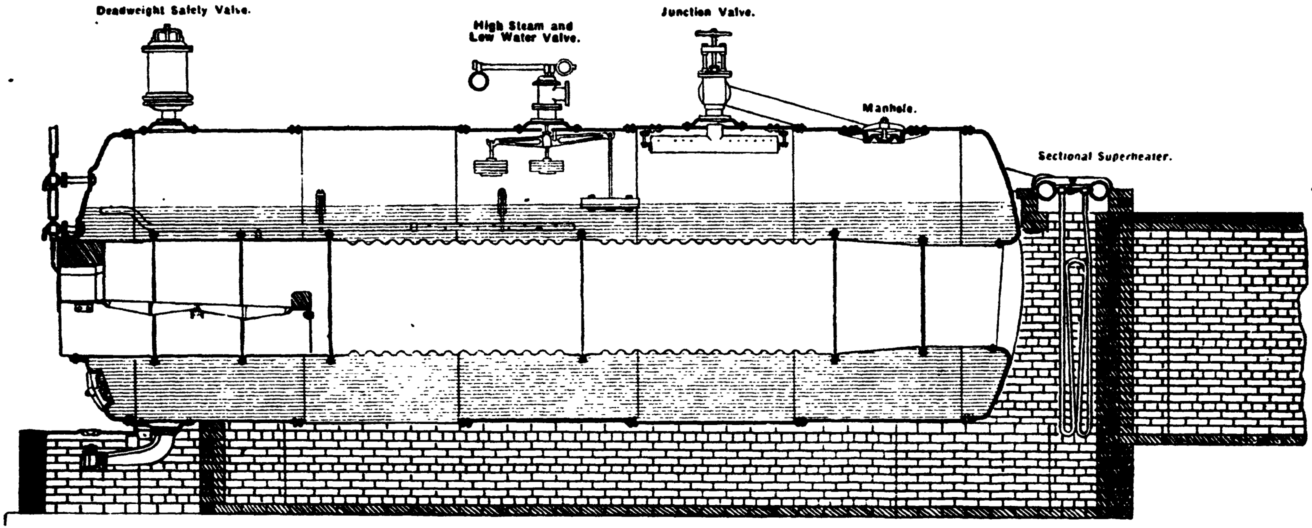


FIG. 11. DISH END LANCASHIRE BOILER WITH CORRUGATED FLUE TUBES
(*John Thompsons, Wolverhampton*)

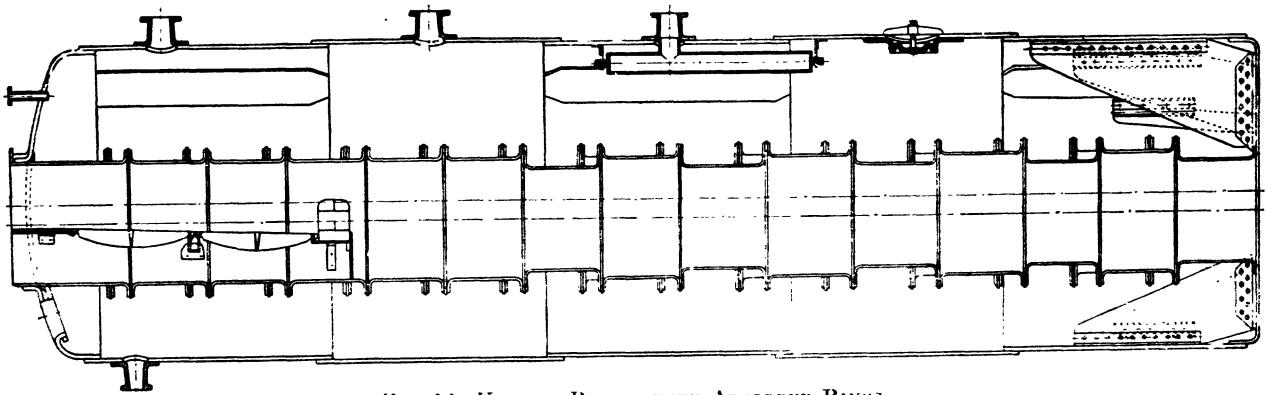


FIG. 12. UNIDISH BOILER WITH ABSORBER RINGS
(*Daniel Adamsons, Hyde*)

are the stresses brought about by the different expansion rates of the tubes as compared with the shell.

Unidish Boiler

In order to minimize these stresses as much as possible, various devices are adopted with the view to increasing the flexibility of the flue tubes. Some makers achieve this object by means of special designs of joint ring, while others insert one or more lengths of corrugated flue. Examples of both methods are shown in Figs. 11 and 12.

Fig. 12 also shows an interesting example of shell construction which combines some of the advantages of both the flat end and dish end boilers. This is the "Unidish" boiler made by Messrs. Daniel Adamsons of Dukinfield. The design, among other advantages, overcomes the difficulty which is experienced with the full dish end boiler—that of making a good job of the brickwork at the back end.

The importance of the construction and condition of the setting cannot be overestimated. It is absolutely essential, if efficiency is to be maintained, that air infiltration should be kept to the minimum. The difficulty of achieving this desirable result will be appreciated, when it is realized that a large Lancashire boiler expands nearly one inch in length from cold to working temperature. Proportional movements take place laterally, and the boiler is continually "breathing." The extent and severity of these movements depend in some measure both on the details of construction of the particular boiler and the way it is worked; but even with the steadiest of loads, and the most scientific construction and workmanship, movement in some degree cannot be avoided. The effect of this movement on the setting is obvious, and much ingenuity has been devoted to the problem.

The Super Lancashire Boiler

The difficulties which are inseparable from external brick settings are entirely avoided in the boiler now to be described. This is a development by a well-known British maker, which results in a boiler possessing some of the characteristics of both the Lancashire and Water Tube types. See Fig. 13.

The boiler is entirely self-contained, and consists essentially of a shell fitted with two flue tubes, which are rather smaller and set a little higher than in the conventional Lancashire.

This arrangement allows space beneath the flue tubes for two scientifically designed nests of smoke tubes. A substantial sheet-steel, brick-lined smoke box or combustion chamber is provided at the back of the boiler, and in this a superheater may be fitted if desired. The chamber is usually provided

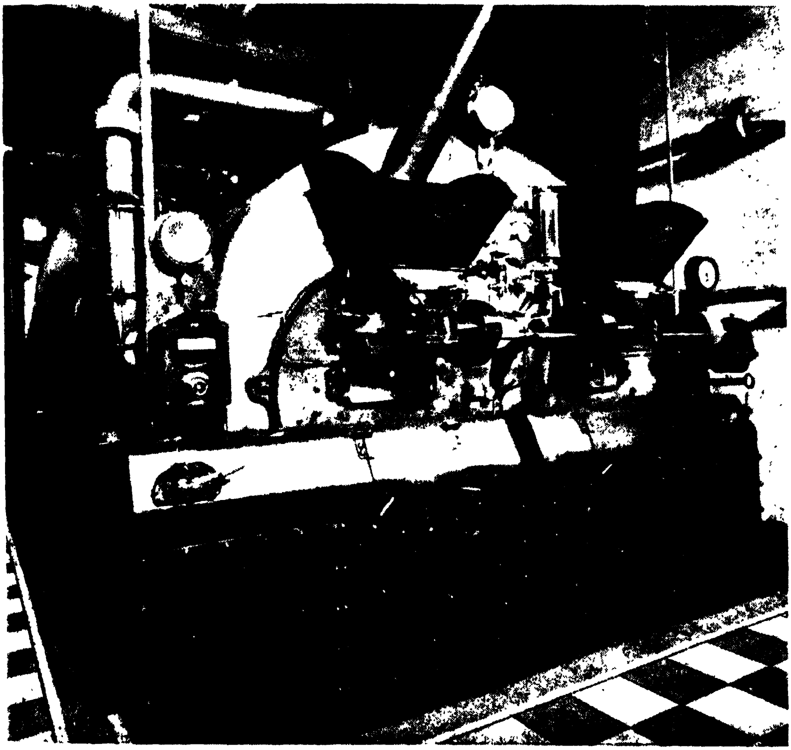


FIG. 13. SUPER LANCASHIRE BOILER
(*Daniel Adamsons, Hyde*)

with a suspended arch of the Liptak type. Ample cleaning and access doors are formed in this downtake. The gases from the flue tubes discharge into this chamber, and are directed thence to the front of the boiler, through the smoke tubes. At the boiler front each furnace is fitted with a separate steel casing, amply provided with doors. On exit from the smoke tubes the gases are drawn through air preheaters, one on each side of the boiler at floor level, and thence to the stack. Balanced draught is used, the standard equipment being an

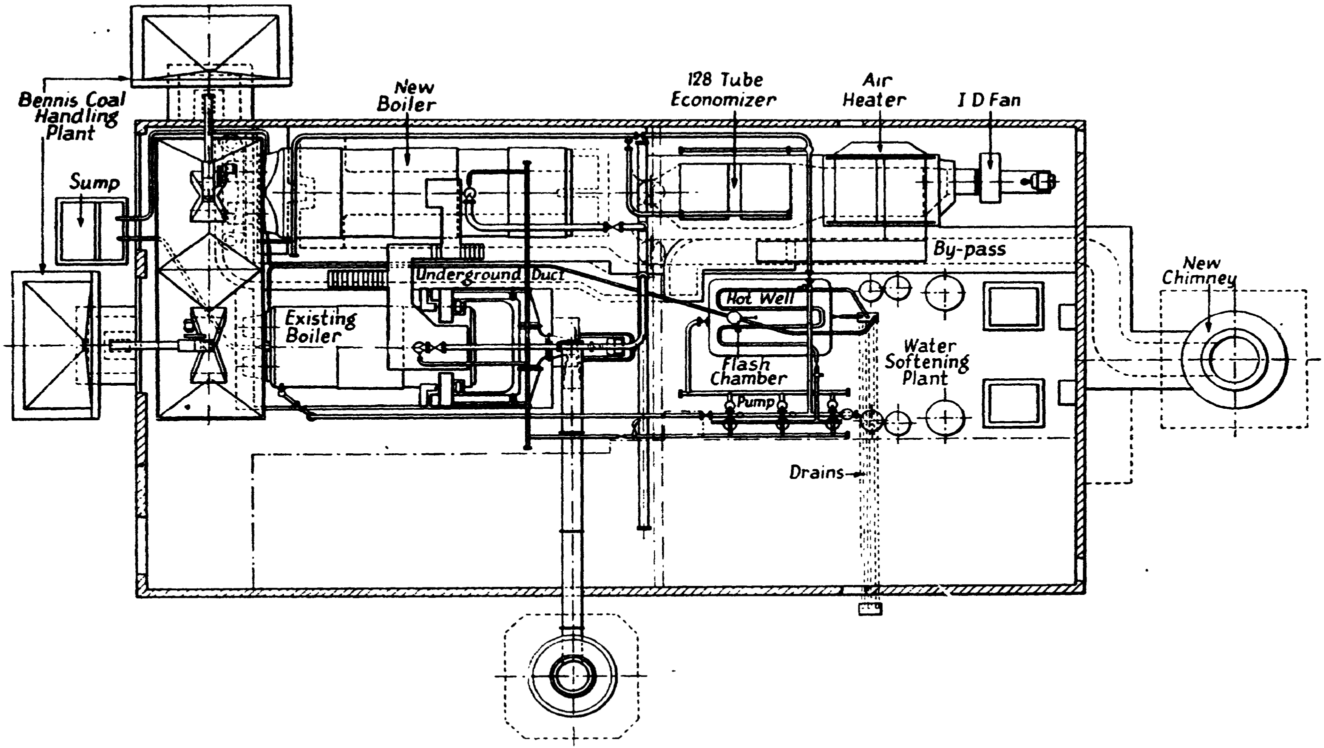


FIG. 14. ACTUAL LAY-OUT, SHOWING SPACE COMPARISON BETWEEN SUPER LANCASHIRE BOILER AND ORDINARY LANCASHIRE BOILER WITH ECONOMIZER AND AIR PREHEATER

induced draught fan fitted in a breeches pipe connected to the gas outlet from the preheaters, and one forced draught fan to each furnace. Gas and air dampers are provided in convenient positions at the firing floor. An ingenious and simple by-pass device directs a small proportion of heated air to the fresh air inlet to the forced draught fans, and thus the air temperature is kept well above the dew point. By this means, possible trouble from corrosion of the air preheater tube ends is avoided. The boiler is a quick steamer, and is, under good conditions, capable of continuous efficiencies of over 80 per cent.

Good feed water is absolutely essential, and comparatively light scale formations are apt to lead to trouble. Provided that good water treatment and careful supervision are available this boiler forms a highly satisfactory unit. Being self-contained and independent of brickwork, its efficiency does not deteriorate at the same rate as the more conventional Lancashire boiler.

The plant possesses a very high steaming capacity per unit of floor space, and Fig. 14 shows an actual installation in which the floor occupied may be compared with a Lancashire boiler of equal capacity, with economizer and air preheater.

Economic Boiler

The more usual form of Economic boiler is shown in Fig. 15. This shows the self-contained internally-fired type, which is very common in this country. Evaporations of from about 1000 to 30,000 lb of steam per hour at pressures up to 300 lb per sq in. can be obtained. Under the best conditions, with adequate draught, efficiencies of 70 to 75 per cent may be expected. The smaller sizes of boiler are fitted with a single flue, but for evaporations exceeding about 4000 lb per hour two flues are more usual. The limiting figure depends on the class of fuel it is proposed to burn, and also on the views of the designer. Whether single or double flue, the basic design is the same. The gases, on emerging from the flue tubes, enter a combustion chamber at the back of the boiler, and are diverted through the smoke tubes to the smoke box at the boiler front, and thence to the stack. Economizers are not usual, but a small and compact design is occasionally mounted between the smoke box and the stack. The number and size of the smoke tubes depends upon the desired gas velocity, and the fact that allowance must be made for sooting up. There are also certain arbitrary dimensions which must be allowed for tube spacing, and breathing space round the flue tubes, so that the whole structure shall not be too rigid.

The volume of the combustion chamber too is subject to a certain latitude, being influenced by the volume of gases

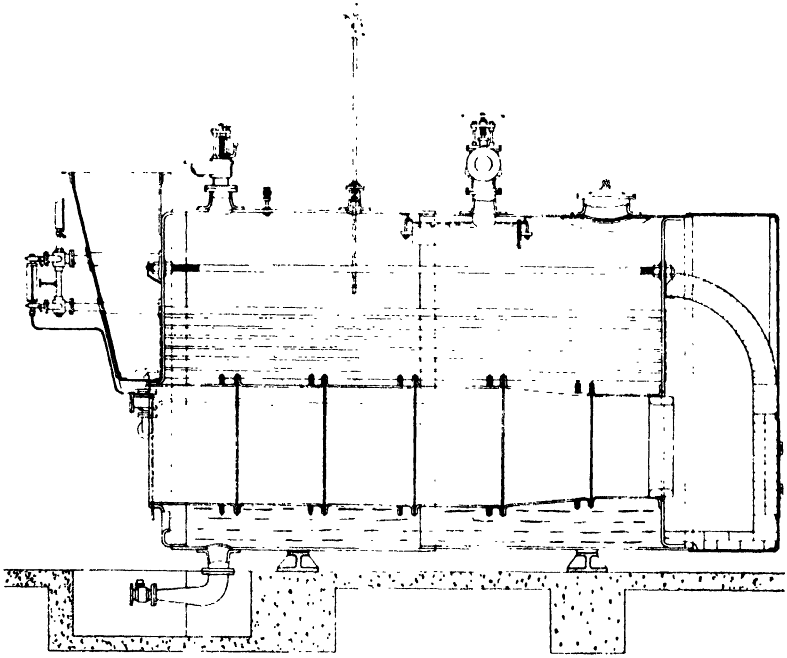


FIG. 15. ECONOMIC BOILER
(*E. Danks & Co. (Oldbury) Ltd.*)

passing—a quantity which depends upon the fuel and the efficiency of combustion.

The boilers are economical in floor space and, roughly speaking, it may be said that they occupy about one-half the space required by a Lancashire boiler of equal capacity. Some additional space is necessary for tube cleaning, but on the other hand a Lancashire boiler of equal capacity and efficiency would at least require the addition of an economizer, which itself may take up another two-thirds of the space occupied by the plain shell. The boilers are quick steamers, and are quite suitable for fluctuating loads provided the peaks are not greater than 25 per cent of the full-load capacity.

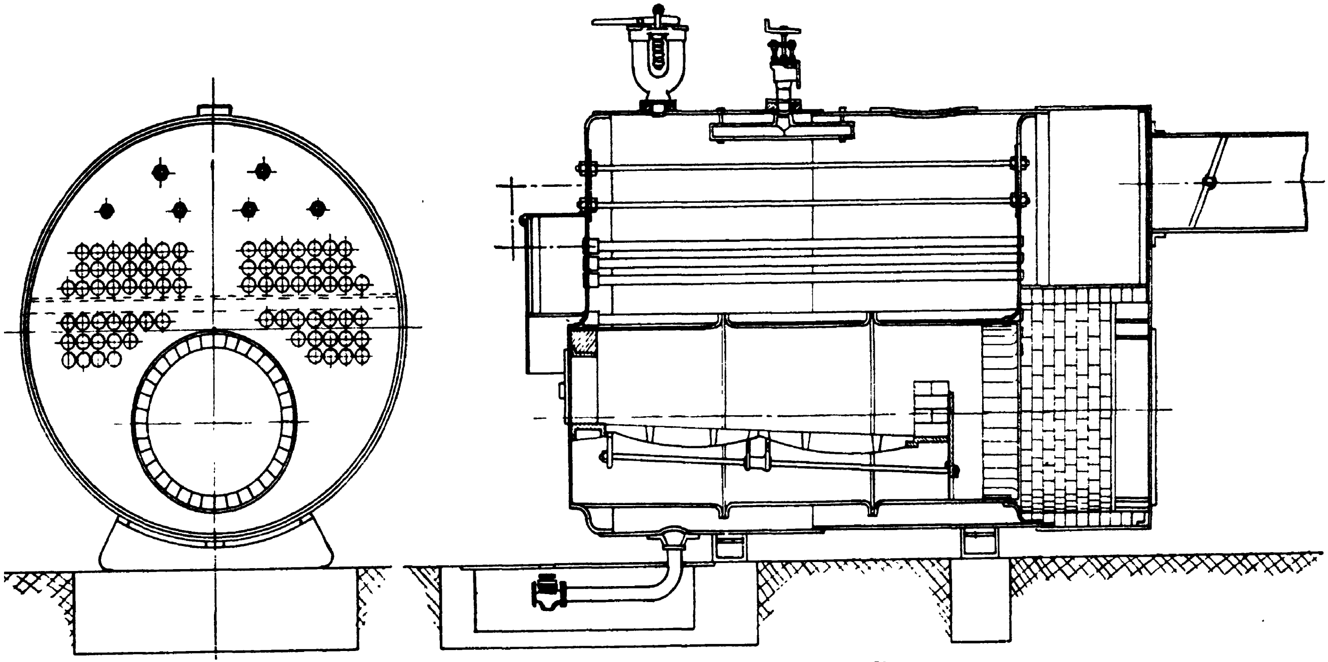


Fig. 16. Robey Double-return Economic Boiler
(*Robey & Co., Lincoln*)

Feed water must be of good quality if good efficiencies are to be maintained and high maintenance avoided.

The Double-return Economic Boiler

An example of this type is shown in Fig. 16. This boiler is made for capacities ranging from 900 to about 7500 lb of steam per hour. It is relatively shorter than the previously described single-return type. There are four nests of tubes, arranged in two pairs, the lower pair forming the first return. On emerging from the flue tubes the gases enter a brick-lined combustion chamber, and flow through the bottom nests to a steel casing at the boiler front. Thence they are directed through the upper tubes to an unlined smoke box, mounted on the top of the combustion chamber. The stack pipe is led away from this smoke box.

Another type of double-return boiler is shown in Fig. 17. In this case the smoke tubes, instead of traversing the shell longitudinally above the level of the flue tubes, are so fitted that the highest tube is just below the level of the flue top. Four nests are arranged in pairs, and the first return from the combustion chamber to the boiler front is via the upper pair of nests. The return (third pass) to the smoke box at the back is by way of the two lower nests. It will be seen from the illustration that the smoke tubes are admirably disposed to transfer the maximum heat to the water, a feature of the arrangement being that the last pass, in which the gas temperature is at its lowest point, is situated in what is normally the coldest region of an internally-fired boiler. The higher and therefore hotter regions of the water space are catered for by the hotter tubes of the first return. Above the level of the fire tubes, still more heat is transmitted by the elevated temperature of the upper half of the flue tubes. It is thus obvious that at all points maximum temperature differences are established between the heated parts of the boiler and the water. Further, the double-return design enables the boiler to be so reduced in length that a larger proportion of the flue tube length is exposed to radiant heat than is possible with the single-return type. This feature is, however, common to the majority of double-return designs.

The boiler illustrated is made for capacities up to 24,500 lb of steam per hour, and for pressures up to 260 lb per sq in. It may be noted that the largest size of this boiler (capacity 24,500), is 12 ft in diameter, and 23 ft 9 in. long, so that the capacity per unit of floor space is very high indeed.

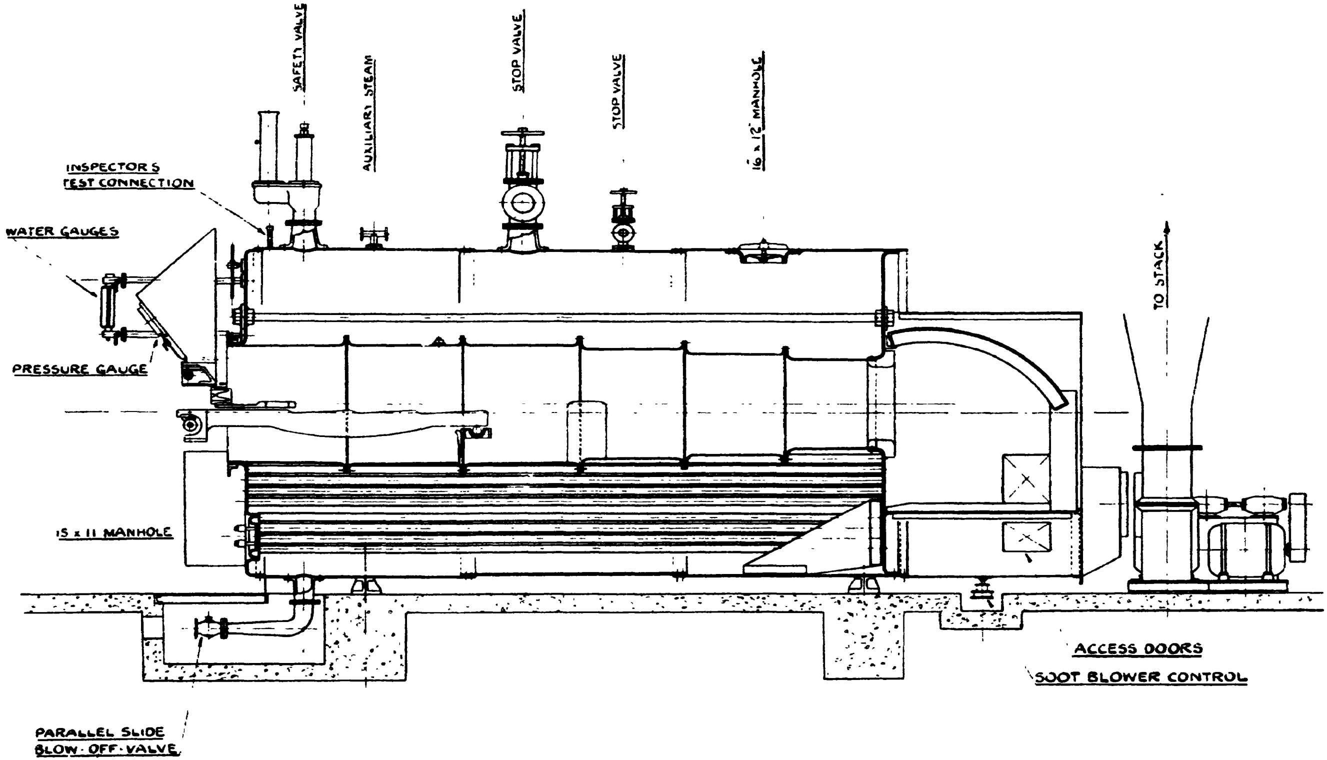


FIG. 17. SUPER ECONOMIC BOILER
(H. & T. Danks, Netherton)

Yet another type of double-return boiler is illustrated in Fig. 18. An important feature of this design is that the flue tubes, smoke tubes, and combustion chamber are completely submerged. Since all the directly heated surfaces are so immersed, it follows that radiation losses are reduced to a minimum. The boiler is normally designed for induced draught, and the velocity of the gases through the smoke tubes is so calculated that maximum heat transfer takes place. Pressures up to 260 lb per sq in. are designed for, and the largest capacity is some 14,000 lb of steam per hour. Very high efficiencies, approaching 85 per cent, are obtained. As will be seen, the plant is entirely self-contained, and is set down on simple cradles. The method by which the flexibility of the combustion chamber is used to alleviate internal stresses is of great interest. With separate anchorages for the second and third passes, the difference in temperature between the two sets of tubes can have no evil effects.

Water Tube Boilers

For pressures above 300 lb per sq in. the use of the Water Tube Boiler is obligatory. The difficulties of large pressure stressed areas which beset the designer of the shell type boilers are overcome by the use of small diameter drums, which form a very small proportion of the total heating surface. The greater part of the work of heat transfer is done by a number of water tubes determined by the required evaporation. The tubes are small in volume, and may be disposed in an almost infinite variety of arrangements.

It is now becoming usual for the larger evaporations to devote some portion of the tube surface to water-cooled walls. By this means it is possible to design for much higher furnace temperatures than would be permissible with simple refractory walls, but the method must be used with care, for with certain classes of coal, combustion troubles may develop if furnace wall temperatures are reduced too far.

The restriction of furnace space which is a feature of the shell-type boiler, and which imposes some limit on the permissible combustion rate, is contrasted in the water tube setting by an almost limitless flexibility of arrangement and construction. With such freedom at his command, it is possible for the designer to provide for any desired steam condition at the boiler stop valve, and for the highest capacities, temperatures,

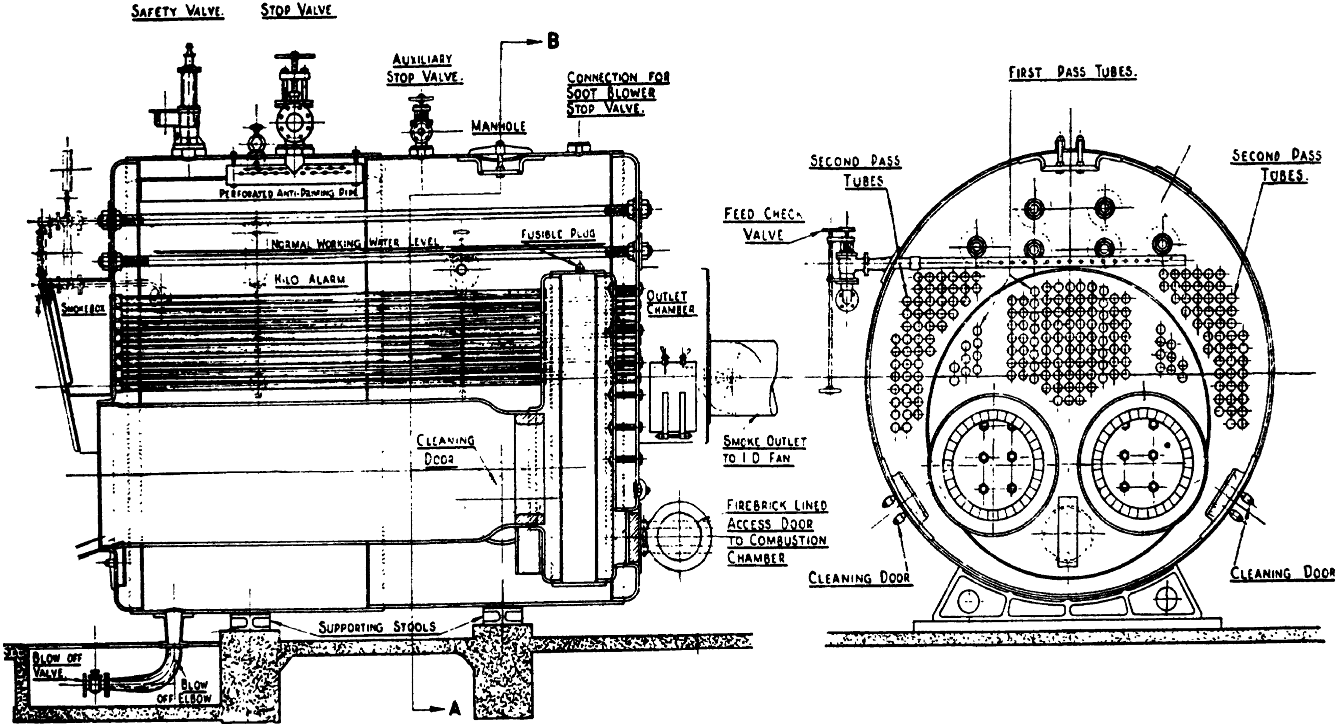


FIG. 18. HORIZONTAL THERMAX
Longitudinal section and front elevation on section line *AB*.
(*Ruston & Hornsby, Lincoln*)

and pressures. Evaporations of 1,000,000 lb of steam per hour have already been realized, and units have been constructed in which steam is produced at the critical pressure of 3200 lb per sq in. Temperatures well in excess of 800° F at the stop valve are not uncommon in power station practice.

The possible steam conditions seem to be imposed rather by the requirements of the turbine designer than by any

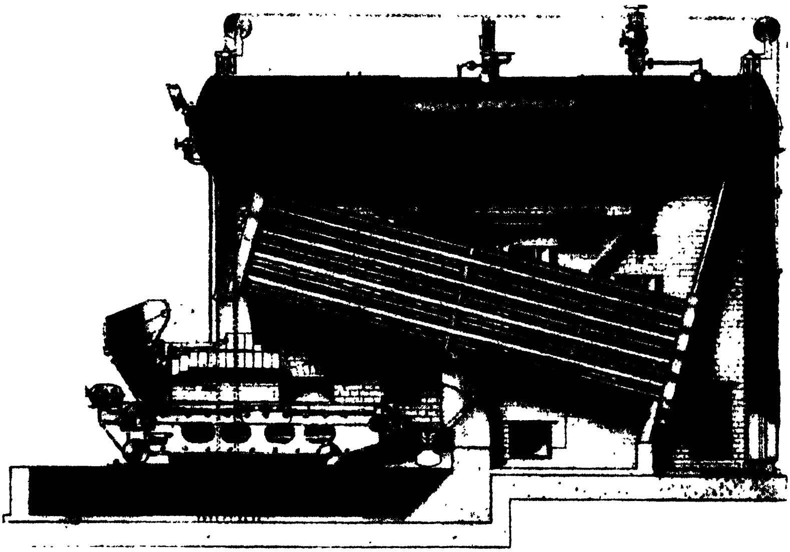


FIG. 19. EARLY TYPE BABCOCK BOILER
(*Babcock & Wilcox, Ltd.*)

limits set by material or constructional difficulties. The modern trend in design is well illustrated by Figs. 19 and 20, in which a modern water tube boiler is contrasted with an older design. Note the enormous volume of the combustion chamber in the modern type as compared with the older model.

Referring for a moment to the internally-fired shell boiler, it is interesting to examine the essential differences in design technique. The differences in form are too obvious to need comment, but the important point is that in the shell boiler the dimensions of the heating surfaces are fixed by the size of the boiler, which in itself is determined by certain empirical considerations. The designer is therefore faced with the problem of producing efficient combustion in a space which is already

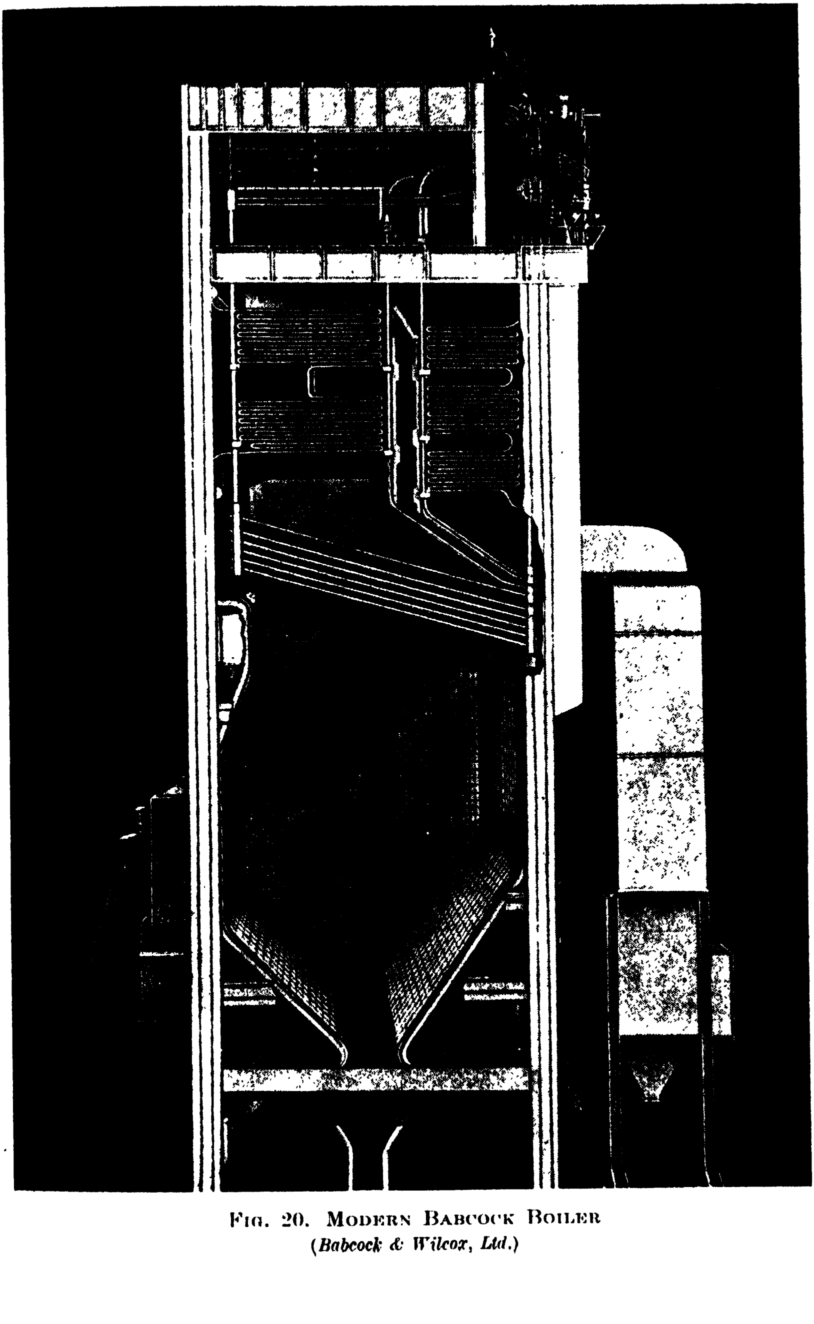


FIG. 20. MODERN BABCOCK BOILER
(*Babcock & Wilcox, Ltd.*)

determined, not however from considerations of adequate heat release, but forced upon him by the exigencies of construction. Not only is the furnace volume limited, but the position is further aggravated by the fact that the coal must be burned in a space completely surrounded by cool surfaces which, should certain defined combustion rates be exceeded, will reduce the temperature of the volatiles below the ignition point, with resulting black smoke and inefficiency. True, it is possible to produce good results with internally-fired boilers, to which statement many plant engineers in this country will bear witness, but the fundamental fact remains that the boiler constructor produces a shell with certain definite limitations, in face of which the combustion engineer must do the best he can.

From a consideration of Fig. 19 it would appear that the possibilities of the new freedom from combustion chamber limitations were not at first realized by the early designers of water tube settings, for the illustration shows a setting which possesses many of the disadvantages of the restricted fire tube. The tubes, it will be noted, are set low down on the grate, so that highly volatile coals must inevitably have given rise to sooting and kindred troubles. But note also that there is little fundamental difference in boiler design between Fig. 19 and Fig. 20; the real contrast is in the setting. Here is the crux of the matter!

Modern Boilers

The modern water tube boiler designer is able to give primary consideration to combustion conditions unfettered by limitations of space imposed by predetermined dimensions, so that the combustion chamber volume may be decided upon from the desired heat release. This factor settled, the heating surfaces may then be disposed in the appropriate positions in the gas stream to give the required heat transfer and steam condition at the stop valve. The heat content of the gases may be calculated at every point, and such items as economizer, superheater, air preheater, etc., arranged to give optimum performance. Surfaces may be exposed to, or screened from, radiant heat as desired.

In general, water tube boilers may be divided into three groups: the straight tube, the bent tube, and more recently, the forced circulation boiler. Examples of the first two groups are shown in Figs. 21 and 22.

Fig. 21 illustrates a usual type of Babcock WIF boiler fitted with chain grate stoker. The tubes are expanded into headers, each of which is connected front and back to the drum horn plates by riser tubes. When assembled, the tubes are thus suspended from the drum, making a very flexible arrangement. Opposite each tube the headers are pierced for hand holes.

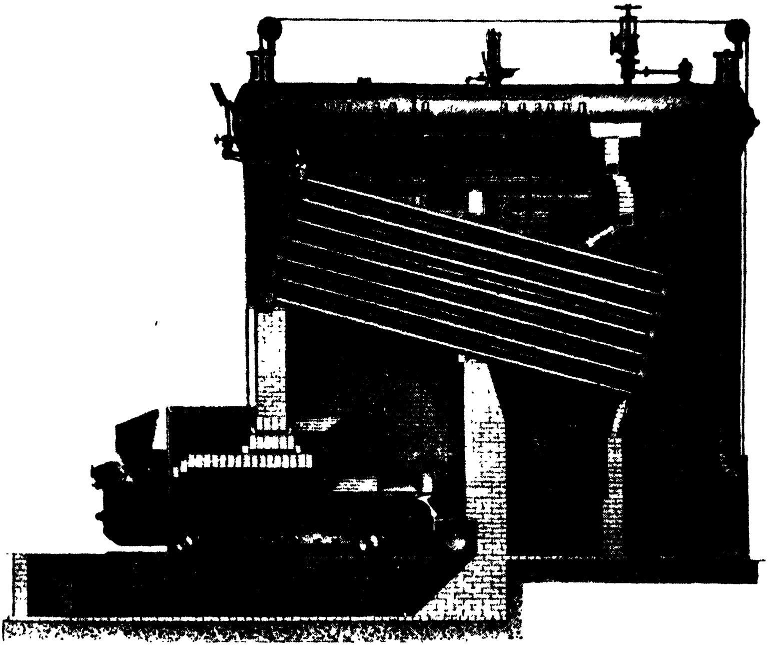


FIG. 21. MODERN TYPE BABCOCK WIF BOILER
(*Babcock & Wilcox, Ltd.*)

They are provided with caps which must be removed for tube cleaning. In this case the superheater is located in the space between the tube bank and the drum.

The Stirling boiler shown in Fig. 22 is of the three-drum type, and the great flexibility of the design is obvious. Access to all tubes is obtained by opening the manholes in the drum ends. Three, or even two drum arrangements may be provided as dictated by the steam requirements.

The forced circulation boiler is shown in diagrammatic form in Fig. 23. The tube surface is large in proportion to the

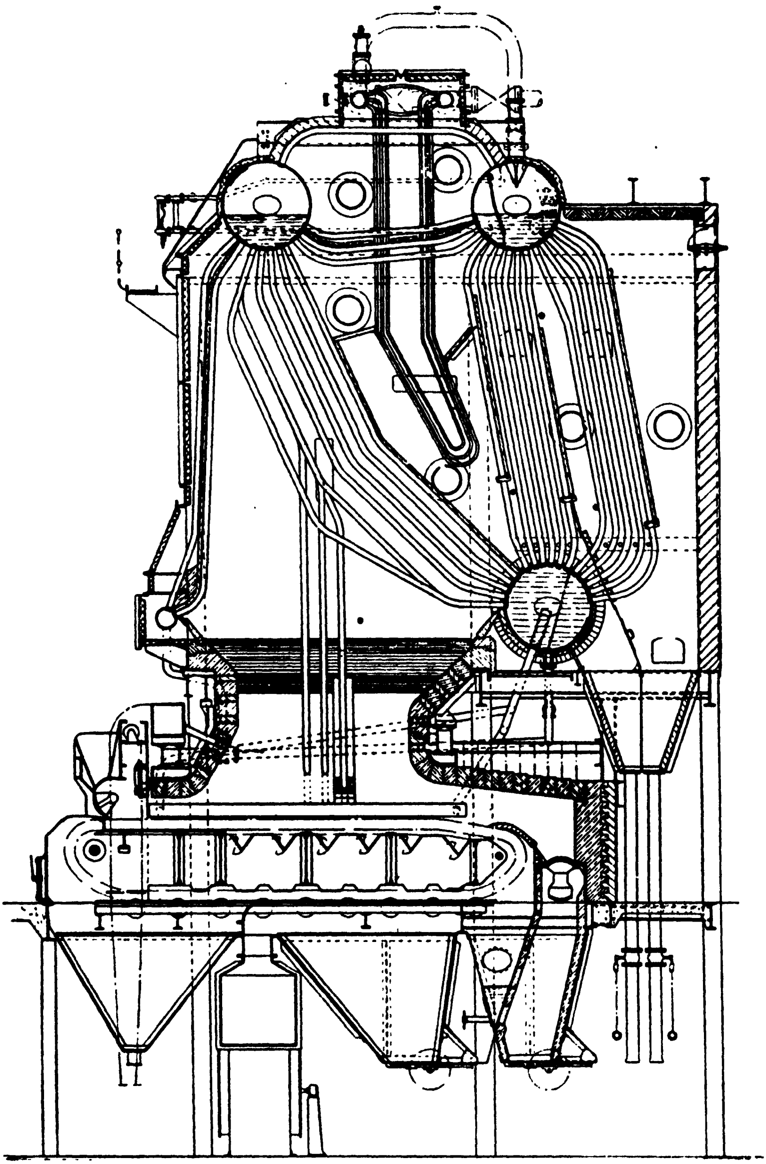


FIG. 22. STIRLING BOILER
(*Stirling Boiler Co.*)

water content, and it will be noted that the drum plays no part in evaporation, but is mounted external to the setting. Rapid circulation is produced by the pump shown, which takes

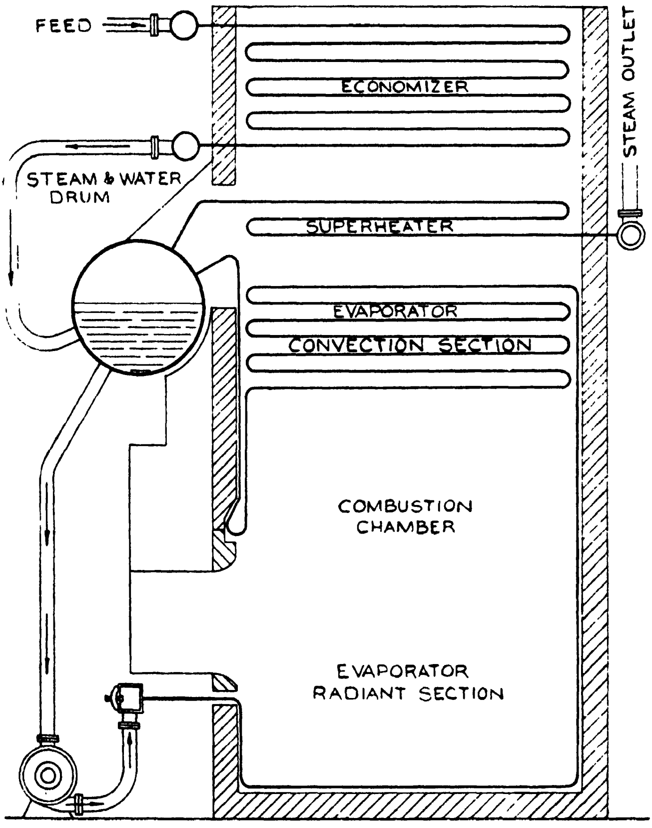


FIG. 23. DIAGRAM OF FORCED CIRCULATION BOILER

water from the drum and forces it through the radiant heat section of the evaporator, which forms a water screen for the combustion chamber. Thence the water passes through the convection section, and so to the drum steam space.

CHAPTER III

ECONOMIZERS AND AIR PREHEATERS

In the last chapter reference was made to the low inherent efficiency of the Lancashire boiler, and to the necessity of some form of heat recovery from the flue gases. At one time the economizer was designed as a means of performing this function alone, but of recent years the development of the high pressure steaming type has led to a somewhat changed viewpoint.

The steaming economizer is not subject to the same limitations of temperature as the more orthodox straight cast-tube construction, and in effect may be regarded as an extension of the boiler heating surface. As its name implies, steam is allowed to form as the water passes through, so that it is quite legitimate to assess the heat absorbed as partly given up to evaporation. As a contrast, the vertical cast-iron tube economizer, which is most commonly to be found in industry, performs the simple function of raising the temperature of the feed water on its way to the boiler by abstracting heat from the flue gases. The amount of heat which can be abstracted in this way is definitely limited by the temperatures which may be permitted at the water inlet and outlet points, and, of course, by the heat available in the gases.

Since, in the cast iron construction, steam must not be allowed to form in the economizer, it follows that the upper limit must be some function of the boiler pressure, and is in fact fixed in practice at about 50° F below the temperature at which steam will form under working conditions. Thus, with a boiler working at 150 lb per sq in. the steam temperature is 366° F, and the limiting temperature for the economizer outlet 316° F.

The lower temperature at the inlet is more or less defined by the dew point of the gases, and is generally established at 110° F. With certain coals, however, it may be necessary to increase this to 130° F if rapid corrosion is to be prevented. Where cold feed is normally supplied, the circulator shown in Fig. 24 is a convenient means of maintaining correct conditions. The feed water flows through the simple injector shown, and in so doing draws some of the water from the hot

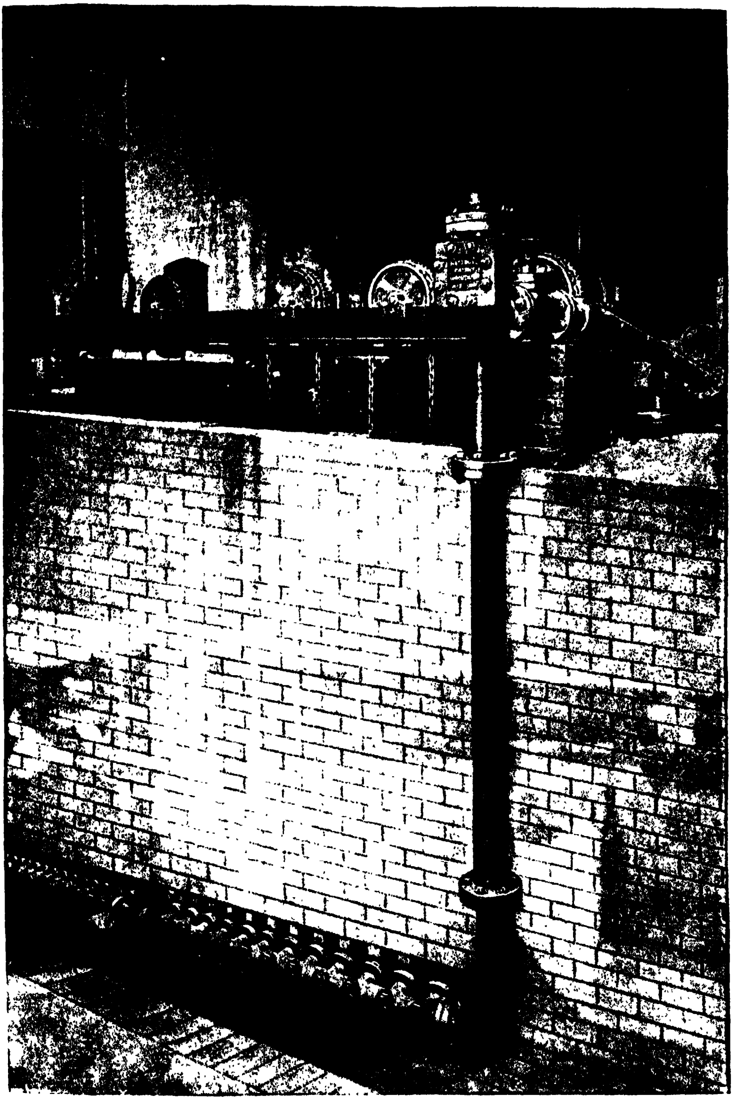


FIG. 24. CIRCULATOR ARRANGEMENT ON VERTICAL TUBE ECONOMIZER
(*National Boiler & General Insurance Co.*)

end of the system and mixes it with the feed. Some measure of control is given by the non-return regulating valve.

Limitations of Temperature

Dangerous conditions may arise if the limiting temperatures at the water outlet are exceeded, so that it becomes the task of the designer so to calculate the installed heating surface that the danger point is not approached. At the same time it is necessary to arrange for the maximum heat to be recovered from the flue gases.

Economizers which are too large can be a constant source of trouble from the operational standpoint. It is quite possible for the plant to be so sensitive that any change of load may tend to establish dangerous conditions, so that the boiler attendant must perforce operate the by-pass damper in order that the economizer may cool down. At the same time, no designer can be held responsible for troubles arising from plant worked under conditions different from those for which it was intended. Instances are not unknown of economizers, originally designed for a pressure of, say, 170 lb per sq in. being expected to work continuously at some much reduced pressure, say 20 lb per sq in., because of some revision of the processes served by the boiler.

In this case, the heating surfaces are proportioned for a final water temperature of 325° F, while the highest permissible temperature when working at 20 lb per sq in. is about 211° F. The difficulties of working the plant under these conditions are manifest.

EXAMPLE.—Consider a Lancashire boiler steaming at 12,000 lb per hour, at a pressure of 150 lb per sq in. The temperature of steam at 150 lb per sq in. is 366° F, so that the limiting temperature at the economizer outlet is 316° F. The feed water is supplied to the inlet at 120° F.

The temperature rise of the water can therefore be

$$316 - 120 = 196° \text{ F} \qquad . \quad . \quad . \ (1)$$

Considerations of heat transfer limit the efficiency of the boiler shell alone to 63 per cent, so that we get

Heat required by the steam per hour,

$$12{,}000 \times [1195 - (316 - 32)]$$
$$= 10{,}900{,}000 \text{ B.Th.U. per hour} \qquad . \quad . \ (2)$$

But at 63 per cent efficiency, the heat required from the coal is

$$17{,}400{,}000 \text{ B.Th.U. per hour} \qquad . \quad . \ (3)$$

Subtracting (2) from (3), the figure obtained represents the losses which are 6,500,000 B.Th.U. per hour. But 10 per cent of these may be allocated to combustion losses and radiation, so that 5,850,000 B.Th.U. per hour represents the heat carried away in the flue gases, and available for use in the economizer.

The weight of the gases must now be calculated.

Referring to (3), 17,400,000 B.Th.U. per hour are required from the coal, so that for coal of 12,000 B.Th.U., 1440 lb per hour are burnt.

Assume the combustion conditions are such that 11 per cent CO_2 is obtained in the flue gases, then reference to the graph in Fig. 26 gives 18·4 lb of flue gas emitted for each lb of coal burned.

Then the weight of gas per hour is

$$18{\cdot}4 \times 1440 = 26{,}600 \text{ lb.}$$

But the heat carried in the flue gases is 5,850,000 B.Th.U., so that if T is the temperature of the gases at the economizer inlet, and 0·24 the specific heat of the gases, we get

$$T = \frac{5{,}850{,}000}{0{\cdot}24 \times 26{,}600}$$
$$= \underline{\underline{910^\circ \text{ F.}}}$$

At this stage the figure may be checked up by the well-known formula for flue loss, which is

$$\text{Flue loss percentage} = \frac{0{\cdot}35\,(T - t)}{CO_2 \text{ percentage}}$$

where T = the flue gas temperature
t = the temperature of the air entering the furnace
0·35 = a constant derived from the specific heats of the gas constituents.

In the example

$$\text{Flue loss percentage} = \frac{0{\cdot}35(910 - 60)}{11}$$
$$= 27 \text{ per cent.}$$

Assessing the remaining losses at 10 per cent, we arrive at the original figure of 63 per cent as the efficiency of the shell alone.

But the temperature rise of the water through the economizer has already been assessed at 196° F (1), and the weight of water passing is 12,000 lb per hour. But to produce this temperature rise, 26,600 lb of gas are passed, whose specific heat is 0·24. Then the drop of temperature in the gases, to produce the known temperature rise in the water, will depend upon the weight of the gases as above, and the specific heat.

$$\text{Then temperature drop} = \frac{196 \times 12{,}000}{0{\cdot}24 \times 26{,}600} = 367^\circ \text{ F.}$$

The final flue gas temperature will therefore be

$$910 - 367 = \underline{\underline{543^\circ \text{ F.}}}$$

Referring to the formula given above, and assuming that the CO_2 at the chimney base will have dropped to 9·5 per cent by infiltration of air, the efficiency of the whole plant is 72·2 per cent so that the economizer may be credited with 9·2 per cent.

It has been shown that the upper limit of temperature at the economizer outlet is determined by the working conditions. In any specific instance, therefore, there is a figure beyond which the *inlet* temperature cannot be raised with economy, because with the outlet temperature already fixed, any elevation of the former must still further limit the range in which the plant must work.

Take the example above, and assume that with the same steam and combustion conditions the temperature of the feed water is raised to 180° F. Then the work done by the economizer must be between the temperatures of 180 and 316° F so that the temperature rise is 136° F. With the same weight of flue gas as before, the corresponding drop in the gas temperature is 255° F, and the final flue temperature is 655° F.

The efficiency in this case, with 9·5 per cent CO_2 as before, is 68 per cent.

If by reason of condensate return or the utilization of waste heat by other methods, the feed is normally returned to the boiler at high temperatures, it is obviously uneconomical to dispose of this heat in other directions in order to cool the boiler feed. At the same time, the limitations of heat recovery with the economizer are clearly defined, so that it becomes necessary to consider further means for dealing with the heat remaining in the gases.

Process Water Heating

Some factories use large volumes of water, which is hot but not boiling. This water can often be provided by using an economizer installed in the flue, after the ordinary feed water heating economizer. The quantity of "process water" which can be pumped or circulated through this second economizer

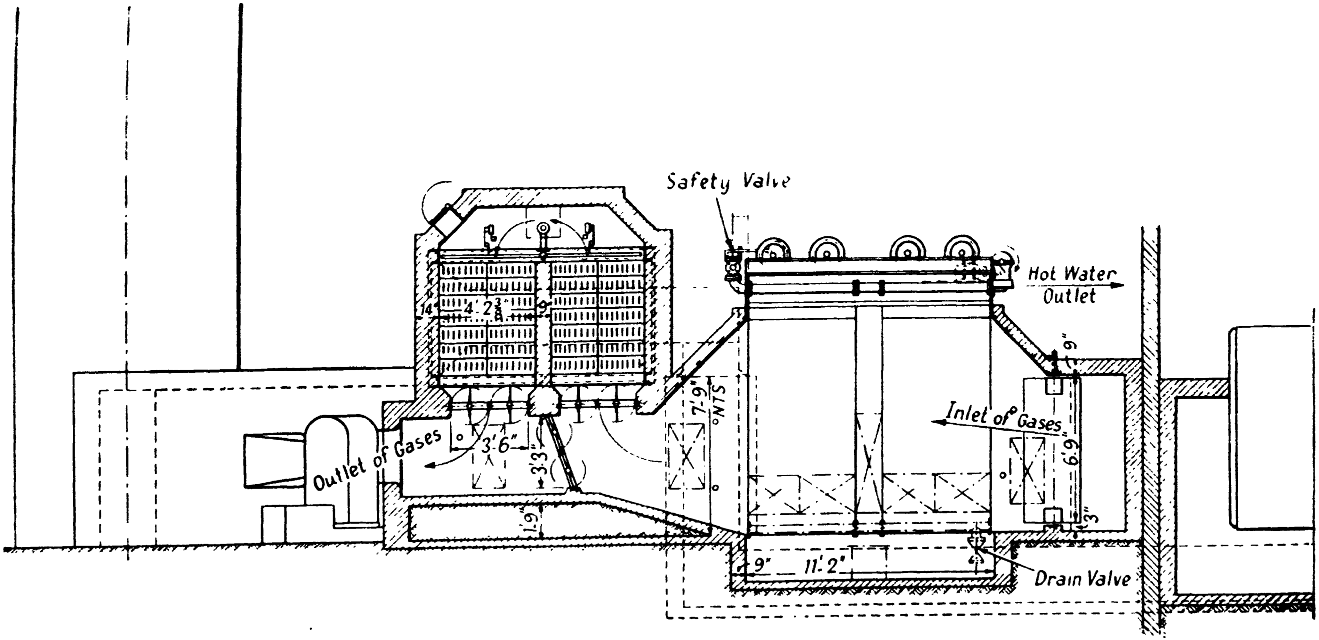


FIG. 25. ARRANGEMENT OF AIR PREHEATER AND ECONOMIZER
(*E. Green & Son. Wakefield*)

will depend on the amount of heat to be recovered. A large volume of water may be raised in temperature by 10 or 20° F, or a small volume may be heated nearly to boiling point. In either case, the temperature of the water entering the second economizer should not be lower than 110° F.

Where process water heating is not feasible the air preheater merits consideration, and in fact may well prove the only adequate answer to the problem.

Air Preheaters

Where preheaters are installed it is necessary for the boiler to be equipped with forced or balanced draught. In the simplest form shown in combination with an economizer in Fig. 25, the air for combustion may be drawn through the preheater by fans situated at the furnace fronts, or forced through by a single fan installed close to the casing of the preheater itself.

In the example shown, the boiler is arranged for balanced draught, and each furnace is equipped with a forced draught fan, which draws its supply through the preheater down the long duct at the side of the setting. The induced draught fan is also shown in a quite normal position on the boiler house floor, adjacent to the preheater, and between it and the stack. It will be noted that in this case the air heater is arranged for parallel flow, that is to say, the air flow is in the same direction as the flue gases. Thus, the maximum heat transfer takes place in the first bank of tubes. This arrangement is sometimes adopted when the temperatures will allow, and when there may be risk of the flue gas temperature dropping below the dew point if cold air were admitted to the cooler end of the heater. The contra-flow arrangement is more usual than that illustrated, as it is possible to realize higher average heat transmission per unit area.

Refer now to the previous example of the Lancashire boiler with economizer, fed with water at a temperature of 180° F. With balanced draught it may be said that practically the whole of the air for combustion is passed through the forced draught fans, and therefore through the preheater. With the CO_2 at 9·5 per cent at the chimney base, it may be reckoned that at the inlet to the preheater the figure will be 10 per cent. Then by reference to Fig. 26, 20 lb of flue gas per lb of coal will pass through the heater, and with 1440 lb of coal per hour burned, the weight of gas is 28,800 lb per hour. The temperature of this gas has already been calculated at 655° F. Now for the air.

The combustion conditions are such that 11 per cent CO_2 is obtained in the boiler passes at exit, so that by referring again

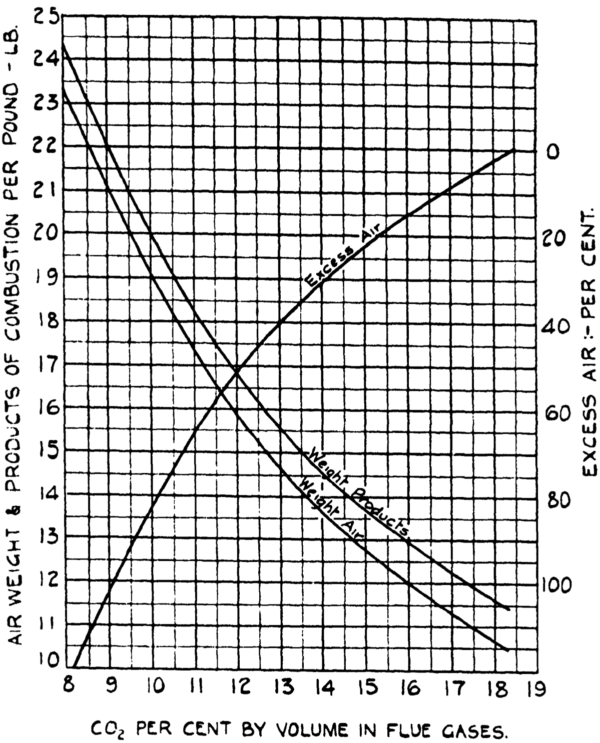


FIG. 26. GRAPH OF AIR AND GAS WEIGHTS PER LB COAL FOR VARIOUS CO_2 CONTENTS IN THE FLUE GASES

to Fig. 26 it is seen that the weight of air for combustion is 17 lb per lb of coal, and therefore the weight passed per hour is 20,800 lb. The temperature at which the air is admitted to the heater is 70° F.

Limits of Temperature

The upper limit to which the air may be raised depends upon the combustion appliances in which it is to be used. For sprinkler or chain-grate stokers 300 to 320° F is a usual figure, but for moving grate bars, a maximum of 240 or 250° F is safer.

Then the temperature rise in the example is

$$250 - 70 = 180° \text{ F}.$$

From the weights and specific heats (Air 0·2, Gas 0·24)

$$\frac{180 \times 20{,}800 \times 0{\cdot}24}{28{,}800 \times 0{\cdot}20} = 156° \text{ F, which is the}$$

temperature drop of the flue gases.

But the gas temperature at the inlet to the preheater is 655° F, so that at the entrance to the induced draught fan the temperature is

$$655 - 156 = 499° \text{ F}.$$

From the formula already given for the flue loss by the dry gases,

$$\text{Flue loss} = \frac{0{\cdot}35(499 - 70)}{9{\cdot}5} = 15{\cdot}8 \text{ per cent.}$$

Reckoning the remaining losses as before at 10 per cent. the total becomes 25·8 per cent, and by subtraction from 100 the efficiency is seen to be 74·2 per cent.

At first sight it would seem to be very doubtful that the air preheater is an economical proposition, because the increase in efficiency as compared with the first example given, with feed at 120° F, is only a matter of 2 per cent. On the figures as given, the capital expenditure would certainly not be worth while, as the most optimistic estimate, with coal at £2 per ton. 50 weeks in the year, and 120 steaming hours per week, gives an annual saving of only £160.

But the true facts are far different when the feed temperatures are taken into account. The criterion is not of course a paper efficiency, but the one figure which is of moment to engineer and manager alike: the comparison of the lb of steam per lb of coal obtainable in each case. From this standpoint, we get,

Case 1. Efficiency 72·2.

Feed temperature 120° F. Steam pressure 150 lb/sq in.

Heat required per lb of steam from feed at 120° F $= 1195 - (120 - 32) = 1107$ B.Th.U.

Calorific value of coal, 12,000 B.Th.U./lb.

$$\text{Lb steam/lb coal} = \frac{12{,}000 \times 0{\cdot}722}{1107} = 7{\cdot}85.$$

Case 2. Efficiency 74·2.

Feed temperature 180° F. Steam pressure 150 lb/sq in.

Heat required per lb of steam from feed at 180° F

$$= 1195 - (180 - 32) = 1047 \text{ B.Th.U.}$$

$$\text{Lb steam/lb coal} = \frac{12{,}000 \times 0{\cdot}742}{1047} = 8{\cdot}5$$

On the same basis as the preceding paragraph, the annual gains amount to £640, which is a far more attractive proposition.

Types of Economizer

Rele

The foregoing illustrates the limitations of the non-steaming economizer, and demonstrates the manner in which the air preheater may be used as an adjunct, to recover heat which must otherwise be lost because of the necessarily restricted temperature range imposed by the formation of steam as the upper limit, and the dew point of the flue gases as the lower.

Generalization is dangerous and frequently inaccurate, but as a rough guide it may be said that whenever the conditions are such that the possible temperature rise through the economizer is 100° F or less, with the flue exit temperature at 500° F or more, some form of air preheater should be considered as a matter of course.

When used as described, in combination with an economizer, the air heater cost is to some extent set off by a possible reduction in the number of economizer tubes. The actual amount of reduction is entirely governed by the particular circumstances. In the calculated examples the 160-tube economizer which would be installed for the conditions quoted for Case 1 would be replaced by one of about 128 tubes, when followed by the preheater.

The type of apparatus shown in Fig. 25 is given in greater detail in Fig. 27, which shows the economizer only.

As illustrated, this very well-known plant is made for pressures up to about 200 lb per sq in. Higher pressures are designed for, but the ring stay tube takes the place of the simpler pressed construction which is used for the lower pressures. The tubes and headers are high-grade cast iron, and the economizer is built up in sections of four, six, eight, or twelve tubes. When assembled, the sections are connected by cross headers, which form the inlet and outlet manifolds.

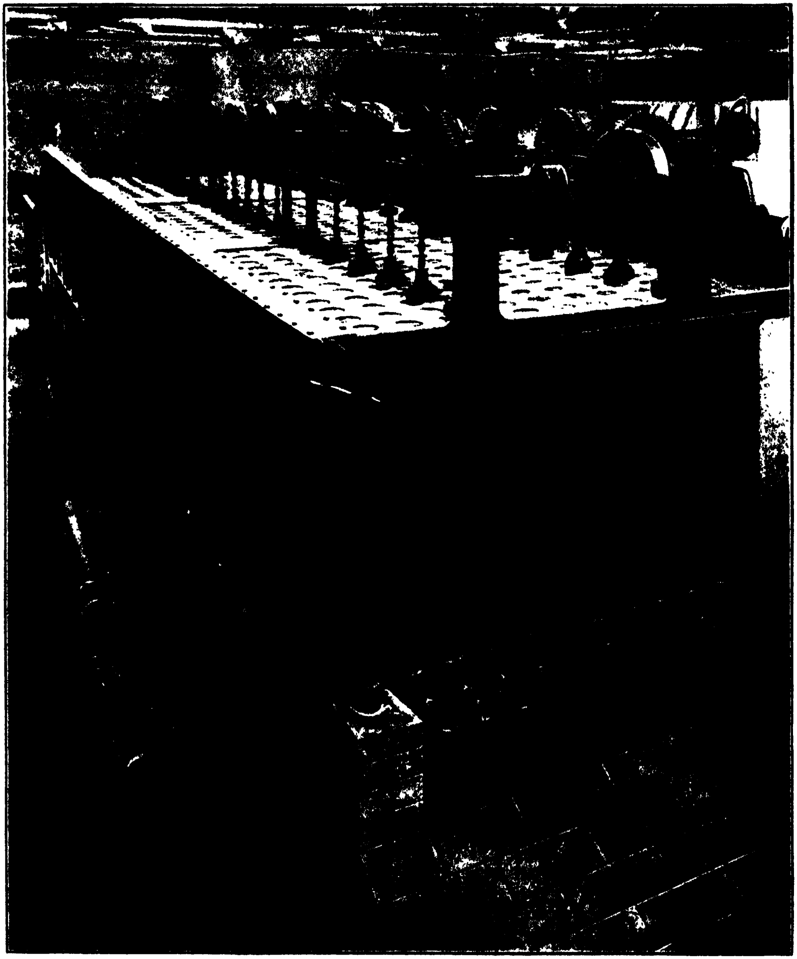


FIG. 27. GREEN'S VERTICAL TUBE ECONOMIZER
(E. Green & Son, Wakefield)

Externally, the tubes are smooth and straight, and the cast construction renders them resistant to the very severe corrosive effects of the flue gases. The scrapers for external tube cleaning now to be described are clearly shown in Fig. 27.

Scrapers

A simple and effective device is adopted to keep the tubes clean externally. Briefly, this is a system of continuous scraping, whereby carrier bars, bearing sets of scrapers, are pulled up and down between each pair of tubes, the chains on which the carrier bars are suspended being driven by a motor or small engine. An automatic reversing gear limits the travel of the chains, and reverses the motion at the correct time, i.e. as each set of scrapers reaches the limit of its travel. The scrapers themselves are suspended quite loosely on the carrier bars, and cannot damage the tubes. A guard bar is suspended from the same central pin as the carrier bar, and the two are lightly bolted together at the extreme ends.

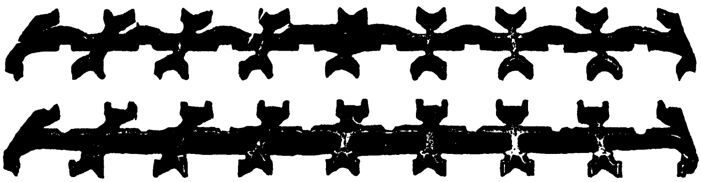


FIG. 28. EXPLODED VIEW OF GUARD BAR AND CARRIER BAR FOR VERTICAL TUBE ECONOMIZER
(*E. Green & Son, Wakefield*)

The scrapers are provided with specially shaped lugs, which drop into semi-circular recesses in the carrier bars, and once in position are prevented from falling out by the guard bars. With certain classes of low-grade coals having high sulphur content, heavy tarry deposits may form on the tubes to such an extent that the scrapers are hampered, and may even break. The replacement is a simple matter, which may quite easily be carried out by the maintenance staff of the factory. Even under normal circumstances with good coals the scrapers will wear, slowly it is true, but in any event it is as well for the maintenance staff to be familiar with the method of renewal.

The general arrangement is shown in Fig. 28. When in working position, the guard bar is retained by the end bolts already mentioned, and also by a taper pin, which is driven through the central suspension rod, above the bar. To replace the scrapers it is necessary first to remove the end bolts, and withdraw the taper pin so that the guard bar can be raised an inch or so above the carrier bar. The taper pin is then

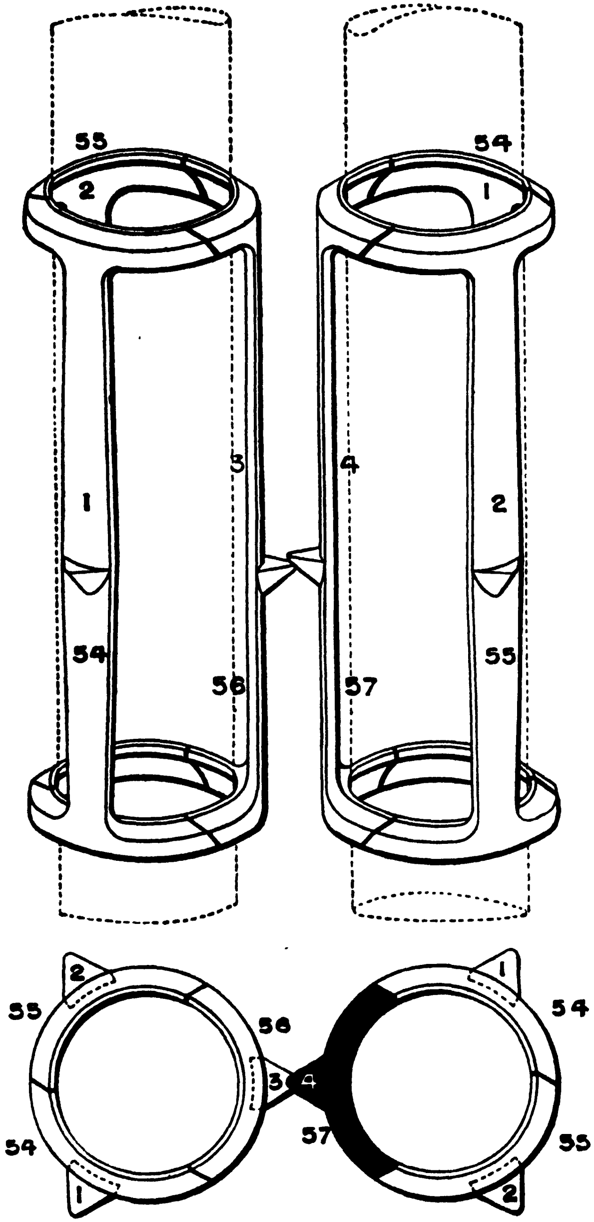


FIG. 29. CORRECT ORDER AND ARRANGEMENT OF SCRAPERS
(*E. Green & Son, Wakefield*)

inserted below the guard bar, so as to support it in the raised position. It is then an easy matter to slip the scrapers into place. There are four patterns of scraper to each set, and these

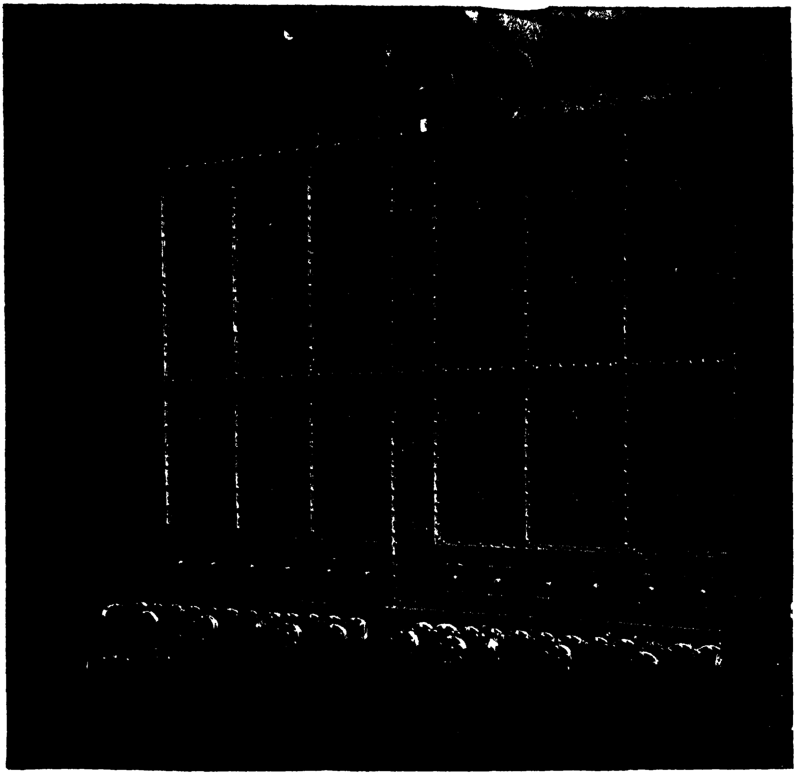


FIG. 30. ARRANGEMENT OF STEEL SIDE PANELS
(*E. Green & Son, Wakefield*)

must be inserted in the correct order as given in Fig. 29, which also shows the correct placing in relation to the tube pairs.

Even in inaccessible positions in the centre of the tube groups, the scrapers can be "fished" into position with the aid of a loop of string on the end of a light stick. The renewal, not only of scrapers, but of the guard and carrier bars if necessary, and in fact all internal maintenance and inspection, are greatly facilitated by the fitting of the insulated steel panels shown in Fig. 30. The advantages of the arrangement are obvious. The removal of a few small bolts frees the panels, and exposes the whole side of the economizer tubes throughout the whole of their length. No brickwork is disturbed in the process, and the panels are just as easily replaced. The framework is so designed that, if required, a row of smaller inspection doors which give access to the lower portion of the tubes, may be removed without disturbing the main panels. For periodic inspection these doors are quite adequate, and minor cleaning and adjustments may also be carried out with no further dismantling.

More serious repairs, such as the replacement of individual tubes, or even section headers, are dealt with by means of special repair tubes or headers which the makers will supply. The defective tube is broken away from the header concerned, and the special repair tube is inserted, either from the top, through the lid cap or, in difficult positions, from underneath. The socket in the bottom header is carefully cleaned, and the tube inserted and driven home. Care is necessary to avoid cracking the header. The upper end of the tube is provided with a collar for caulking, and the whole is formed into a rust joint with cast-iron borings and sal ammoniac. Full instructions are given by the makers for this class of repair, and also for the replacement of both top and bottom headers.

The scraper driving gear, together with the arrangements for reversing, are of very robust construction; all parts are easy of access, and beyond routine maintenance work they demand no special attention.

In operating, care should be taken at all times to see that the inlet water temperature is kept well above the point at which the flue gases will condense on the tubes, as discussed earlier in this chapter, and when shutting down, the gases should be by-passed as soon as possible. Keep the scrapers working at all times. It is as well to start them going directly the fires are put in, and they should not be stopped until the grates have been run off, or, in the case of hand-firing, drawn.

Provision is made in the design of the driving gear to hand-operate the scrapers in order to free them should sticking occur, as it may if the plant has been neglected or incorrectly operated. Where the deposits have been allowed to accumulate it may be necessary to resort to burning before normal scraper working can be resumed. This method will clear the tubes even in the worst cases, and where bad coals are used it may be well to use it periodically, in order to avoid broken scrapers. The procedure is to lift the safety valve of the economizer, remove one or two of the access lids, empty down and light a wood fire in the setting, on the boiler side of the economizer. The heaviest deposits will dry up and scale off. If the scrapers can be worked at all, it is as well to keep them going during the process to assist the removal. If the scrapers cannot be operated, the flue should be entered as soon as possible and the tubes brushed off with wire brushes. This is difficult to perform satisfactorily, and it is better to work the scrapers if it is at all possible. In any case, much of the deposit will come away from the tubes during cooling, and can be disposed of through the access doors in the lower part of the setting.

Steam should never be allowed to form in the tubes under any circumstances. Watch must be kept on the outlet thermometer at all times, and the by-pass dampers must be used at the first indication of a dangerous rise in temperature. This applies to all non-steaming economizers, and is a most important factor in operation. Should an involuntary stoppage occur to the feed pumps, the gases must be by-passed at once, even if it is known that the stoppage will be of only a few minutes' duration. The potential energy possessed by the water content will be realized when the volume of steam flashed off by a sudden release of pressure, such as would follow the rupturing of a tube, is appreciated. Consider this in conjunction with the stresses produced by the admission of water to a tube in which steam has formed, and the necessity for care in operation needs no further emphasis.

It will be realized that for any given condition the heat available from the flue gases is a fixed quantity, and that in order to extract the maximum amount of this available heat between the limits imposed by the factors discussed above, the designer has but two courses open to him. He may endeavour to modify either the temperature difference between the gases and the water or the heat transfer through the tube walls. The first course can be followed only to a strictly limited extent, except in the case of the steaming or process water economizers. Here, although the lower temperature is still defined by the dew point of the gases, there is much greater latitude at the upper end of the temperature range. But the non-steaming, feed heating type of plant is subject to no such latitude, so that the only means the designer has at his disposal to affect the temperature difference alone, lies in the manner in which he may be able to manipulate the direction of flow of the one medium relative to the other. In other words, it may or may not be possible to arrange for contra flow.

The second modification, that is to say, the heat transfer through the tube walls, is in itself affected by two other main factors. The first of these is the rate of heat transfer from the flue gases to the wall of the tube, and that depends on the speed of the gases over the tube surface. The second factor is the amount of heating surface *per foot of tube exposed to the gas flow.* It may be said here that, between wide limits, variations in the rate of water flow have no appreciable influence on the rate of heat transfer.

The velocity of the gases must be carefully considered in

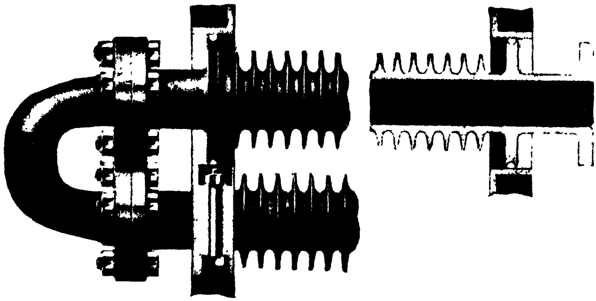


FIG. 31. GILLED TUBE FOR ECONOMIZER
(*E. Green & Son, Wakefield*)

relation to its effect upon the power requirements of the draught-producing equipment. It is obvious that there is an economic boundary beyond which the designer may not venture.

Consideration of the second factor led to the introduction of the gilled or finned tube, in which the heating surface per unit length of tube is considerably increased, and enables larger heating surfaces to be provided in restricted spaces. Advantage can also be taken of higher gas velocities.

An early but still widely used form of "extended surface" economizer tube having circular gills, is shown in Fig. 31. The gills are cast integrally with the tube, but it is also common practice to shrink a cast-iron gilled sheath or sleeve on to a steel tube. This arrangement can be constructed for the highest pressures, and also possesses the advantage that the cast-iron sleeves are resistant to the corrosive action of the gases.

The H-tube Economizer

A later design of economizer is shown in Fig. 32. This is the H-tube variation of the gilled tube, and it is claimed that it combines some of the advantages of both smooth and gilled economizers. The tubes, it will be noted, are horizontal and are furnished with fins, so shaped that when assembled, vertical gas passages are formed by the spaces between them. These passages are quite straight, and it is claimed that the gas velocity permissible through them is so high that little dust deposition can take place. A simple form of soot blower is, however, provided.

The system of connections for the water tubes is such that almost any desired effect can be obtained. The tube ends, when in position, are equidistant from each other, so that the simple U-connector bends may be fitted in a variety of ways, providing series flow, parallel flow or any desired combination. Alterations could be made, even after the completion of the installation.

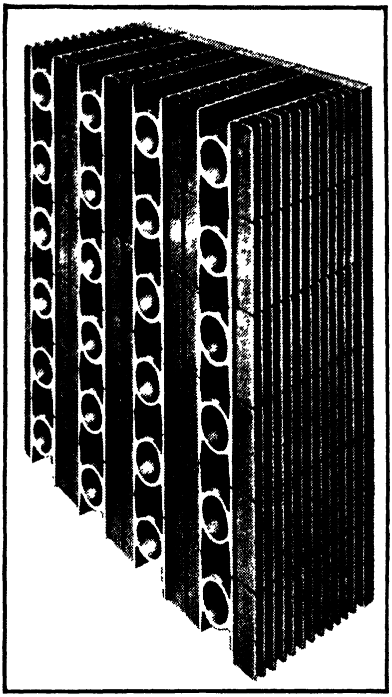


FIG. 32. SENIOR H-TUBE ECONOMIZER
(*Senior Economizers, Ltd.*)

"Premier Diamond" Economizer

A modern design of gilled tube economizer is shown in Fig. 33. This is the "Premier Diamond" economizer, which is claimed to combine the advantages of straight gas passages with ease of inspection of all external surfaces, there being no inaccessible pockets between the tubes. The gills themselves are rectangular, and are formed upon a diamond-shaped tube body. When assembled, the tubes take up a staggered arrangement, ensuring fullest contact with the gases and at the same time the unique shaping of the tube encourages an approximation to streamline gas flow. This reduces to a minimum the dead space effect which is inseparable from tubes of a more conventional shape.

The method of assembly is clearly shown in the illustration, from which the easy access for inspection, and general cleanliness of design are evident. Another point to be noted is that in this construction the jointing flanges are not the same as the tube supporting flanges, thus allowing bolts to be used instead of studs.

Cast-iron gilled tubes are generally made for pressures up

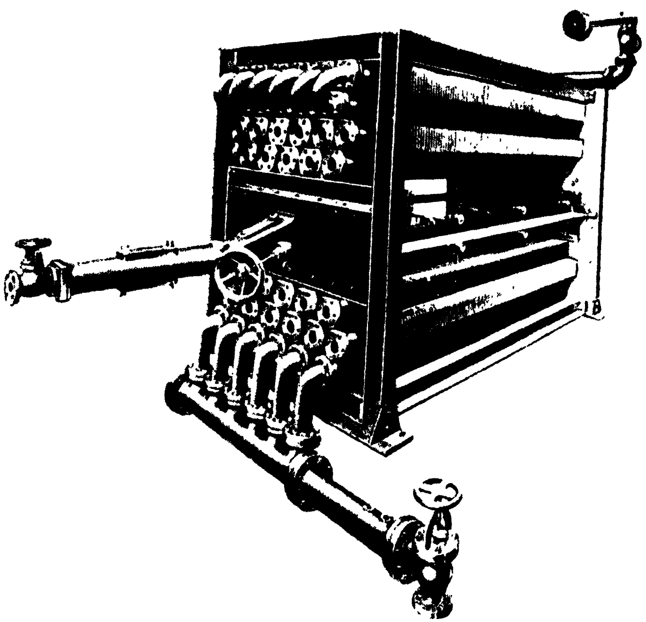


FIG. 33. "PREMIER DIAMOND" ECONOMIZER
(*E. Green & Son, Wakefield*)

to 300 lb per sq in. though the modern designs are suitable for somewhat higher pressures. Both the H-tube and the "Premier Diamond" economizers are available in the high pressure form, which has cast-iron fins shrunk on to steel tubes. Standard designs are available for pressures up to 2000 lb per sq in., and special constructions are produced for the very highest commercial pressures. Gilled tubes in general have certain limitations, and care is needed in studying a plant before deciding to use this type of economizer. They should never be used where scale-forming untreated feed water has to be employed, because the internal surface of a gilled tube economizer is very small in relation to the volume of water with which it has to deal. For this reason gilled tubes choke rapidly with scale-forming water. Apart from this, the

FIG. 34. GREEN'S OVAL TUBE AIR PREHEATER
(*E. Green & Son, Wakefield*)

water content of this type of economizer is very low, and its thermal storage is small in comparison with the older vertical tube type. The point is not of great importance where the load on the boilers is very steady, but if there are sudden peak demands the advantage of a large volume of hot feed water stored in the economizer is very great.

Types of Preheater

Another difficulty with the gilled tube economizer is its liability to collect flue dust. Mechanical cleaning is hardly a practicable proposition, and it is by no means easy to devise and arrange a form of soot blower which will keep such extended surfaces clean and free from dust lodgment.

Various designs of soot blower are available, some using steam, and others compressed air, but it must be admitted

FIG. 35. END VIEW OF ELEMENTS SHOWING OFFSET
(*E. Green & Son, Wakefield*)

that the cleaning problem is very real, and, with some coals, presents major difficulties.

Reverting to air preheaters, the type shown in Fig. 25 is given in greater detail in Fig. 34, which illustrates an assembled section. The tubes, which are oval in section, and are of cast-iron, $\frac{1}{4}$ in. thick, are mounted in cast end plates. One pair of end plates carries eight tubes, and the method of mounting is such that each individual tube is free to expand, so that the end plates themselves are not subject to stresses. Specially-shaped recesses are cast in the plates, and the tubes are caulked with specially-prepared asbestos cord. The peculiar shape of the recess makes it impossible for the packing to work out in use.

From Fig. 35, which shows the end view of an element, it will be noticed that an extremely simple method is adopted to facilitate a staggered tube assembly. The tubes are not mounted symmetrically in the end plates, being set one-half tube width "off" the shorter centre line.

The sides of the end plates are pierced with assembly slots, so that by simply fitting each section the opposite way up to its fellow a complete staggered tube arrangement is achieved, with a single pattern of section.

The oval form of the tubes offers little resistance to the gas flow and at the same time a very large proportion of the total tube surface is effective for heat transfer. The sections may be either supported by a steel framework or, for small jobs, built direct into the brickwork of the setting. Soot blowers are provided for cleaning.

Before proceeding to other forms of preheater, it will be well to examine an example of the use of this appliance alone, without an economizer.

The method in which a tubular form of preheater is incorporated in the design of the Super Lancashire boiler is a suitable instance. (Some details of this plant have been given in Chapter 2.) The air heaters consist of nests of tubes, circular in section, and of perfectly straightforward design, fitted into what may be described as rectangular ducts, on either side of the setting. The flue gases flow through the tubes after passing through the convection tubes of the boiler itself. Deflectors, which consist of long strips of twisted metal, are placed in the tubes, and the swirling motion which is thus imparted to the gases tends to break up, and prevent the formation of a heat-resistant gas film at the walls.

The air for combustion traverses the outside of the tube nest on its way from the forced draught fans to the furnace. At the cold air inlet to the fans a controllable by-pass device is provided, whereby some portion of warmed air is allowed to mix with the incoming air, raising the temperature, so that the temperature of the mixture never falls below the dew-point. By this means difficulties with corrosion of the tube ends are avoided. At the same time, the temperature at the inlets to the forced draught fans is not so high that any special design is called for, and for the same reason the volume to be handled is such that the fans are kept within reasonable dimensions. The actual temperature at inlet is 115 to 120° F, depending on the operating conditions and the position of the by-pass damper control, and the air under the furnaces is at about 320° F. But the temperature at inlet, as has been shown, is the temperature of a mixture, some part of which has already passed the preheater, so that in estimating the actual work performed by the tubes the rise must be calculated from the ambient temperature of the boiler house. Assuming this to be 70° F, the rise through the preheater of the air temperature is seen to be 250° F.

The efficacy of the arrangement is such that under normal operating conditions the flue gas exit temperature is frequently below 350° F, and if we apply the methods already discussed, the temperature of the gas at the entrance of the preheater tubes, i.e. after the normal convection surfaces have been traversed, is seen to be of the order of 610° F.

The cased-in arrangement of the whole plant enables the CO_2 figures of 12 per cent to 13 per cent to be carried quite easily, without the formation of CO, so that by the use of the usual formula the dry gas loss at the inlet to the stack may be estimated at 7·55 per cent to 8·17 per cent. The radiation is less than in the usual Lancashire boiler, and actual tests have shown that the aggregate of the remaining losses is not more than about 9 per cent. The efficiency is therefore 83·45 per cent to 82·83 per cent, according to the combustion conditions. At the entry to the preheater, where the gas temperature is 610° F, the figures on the same basis are 76·5 per cent, and 75·2 per cent respectively. In this case, therefore, the preheater is responsible for 6·95 per cent of the work done.

For boiler work, this is the normal range in which the preheater can operate. It has been shown that the lower temperature is determined by considerations of dew-point, and mention has also been made of an upper limit of about 320° F at the grates, a figure which is influenced by the fusing point of the ash in the fuel used.

An interesting type of preheater, much in demand for power station work, is illustrated in Fig. 36. This is a rotary heater, the moving element of which consists of a rotor built up of specially corrugated mild steel plates, arranged to form passages for the gas and air.

The preheater may be described as a regenerative device, in which one-half of the slowly moving element is always being heated by the flue gases, and the other half cooled by the incoming air for combustion. Heat is thus conveyed from the gases to the air by reason of the heat storage capacity of the elements, which are alternately in actual contact with the hot and cold gas and air streams. The two streams are kept apart in practice by the provision of sealing strips and sealing

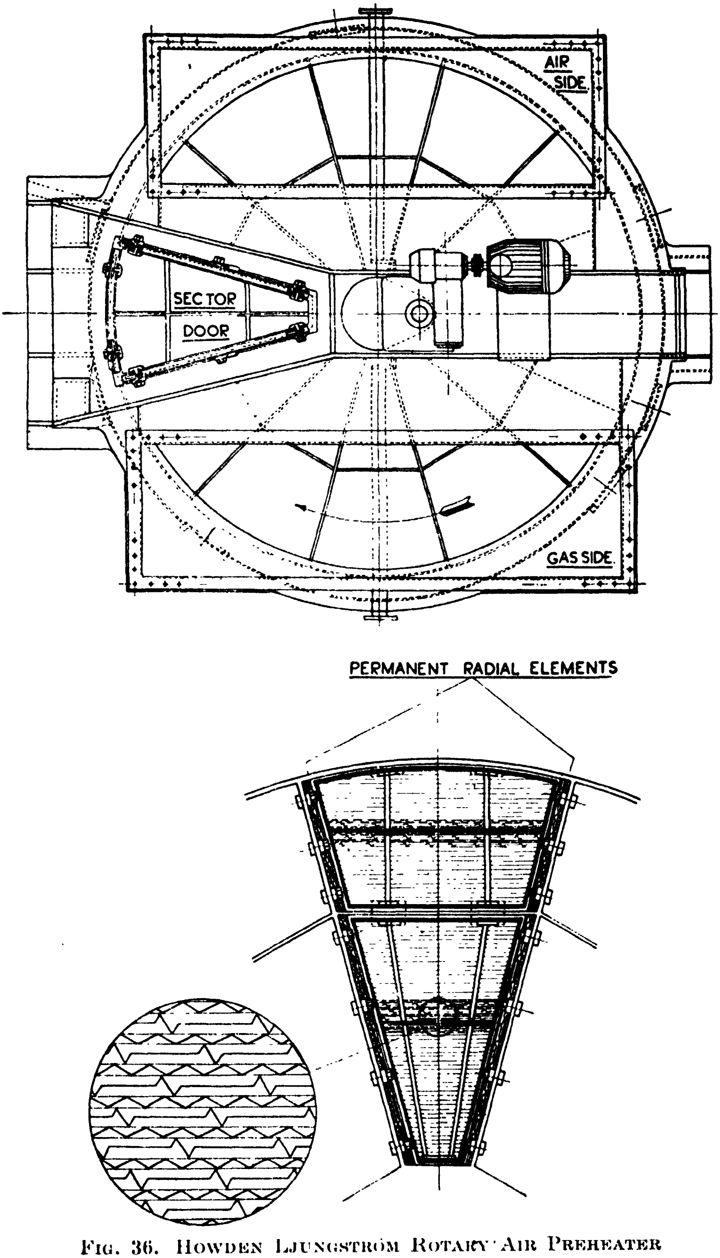


FIG. 36. HOWDEN LJUNGSTRÖM ROTARY AIR PREHEATER

(*James Howden, Glasgow*)

plates, so arranged that one of the strips must always be in contact with a sealing plate, so that any interchange of air and gas is prevented.

Still another type of preheater is that which embodies

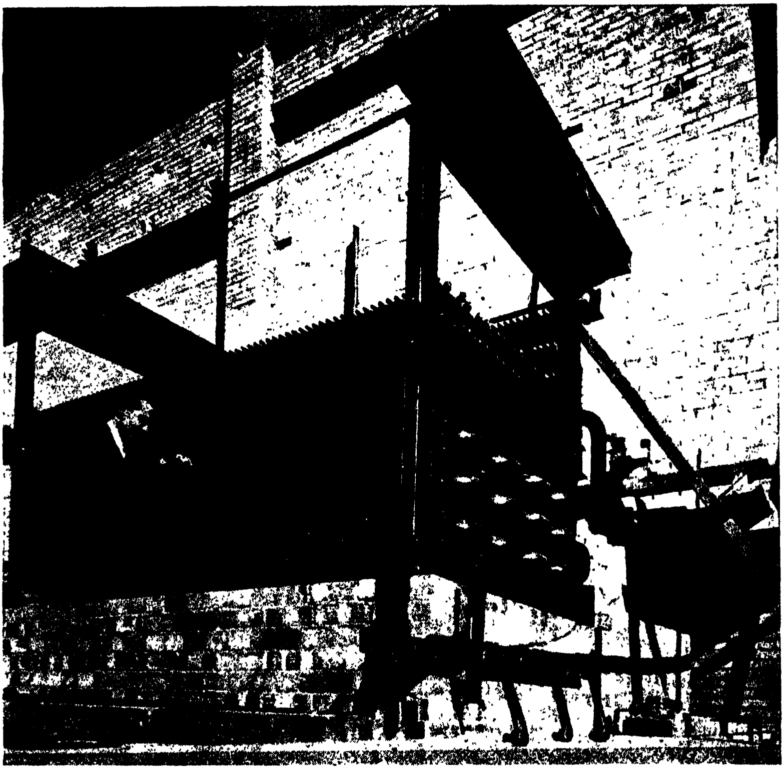


FIG. 37. NEWTON NEEDLE ECONOMIZER

The air preheater tubes are similar in construction, and are made the same length, so that the completed preheater may be superimposed upon the economizer if desired.

(*Newton Chambers Thorncliffe, Sheffield*)

needle tubes. In the well-known type similar to the economizer (Fig. 37) needles are provided both on the gas and air sides of the elements. The construction is of cast iron, which, as in the case of the economizer, is chosen for its corrosion-resistant properties. Also, as in the case of the gilled tube economizer, the object of the design is to provide large heating surface in small space, and in fact a needle type preheater occupies only about half the space taken up by some of the more usual designs. In this instance, and as a contrast to the preheater used in conjunction with the Super Lancashire boiler, the air flows through the interior of the tube elements, while the flue gases traverse the outside. Soot blowers are provided to clean the gas passages, and these may be operated by steam or compressed air.

CHAPTER IV

MECHANICAL STOKERS

For shell type boilers there are four types of mechanical stoker, three of which may be said to be in general use. Of these three, two have their counterpart in hand-firing methods, that is to say, they feed the fuel to the furnace in the same way as the shovel in the hands of a skilful fireman. But whereas the skilled fireman can adapt his methods both to changes of fuel and to load conditions, and can therefore handle a very wide range of fuels upon the same grate (always providing that adequate draught is available), the method of firing for any given stoker cannot be altered, and there is some limit to the class of coal it is possible to handle upon it. Of recent years there have been many improvements in detail, largely concerned with the supply and control of air, and the range of coals which can be burnt upon individual machines has been considerably widened. At the same time, careful consideration must be given to the class of fuel most likely to be available, and to the characteristics of the load to be developed, before the class of stoker is decided upon.

Before proceeding to the details of construction and working, it will be well to make a general comparison of the two classes of stoker, and to discuss those factors which influence the selection of either type.

Comparison of Coking and Sprinkler Stokers

The coking stoker possesses a feeder mechanism by which the fuel is pushed at a controlled rate from the hopper on to a dead plate at the furnace front, where it is ignited, and from this point a moving grate transports the burning fuel slowly towards the back. The rate of movement and air supply should be such that the coal is just burnt out upon its arrival at the end of the grate bars, so that only ash is allowed to drop over into the ashpit.

The fuel drops from the hopper on to the dead plate, in a mass from 8 to 12 in. thick, so that little air can pass through the fuel bed at this point. Distillation commences, and the volatiles are ignited during their passage over the incandescent fuel bed. The partially coked coal is gradually pushed, or rather carried, by the moving grate bars, and as it clears from the coking zone, and the firebed becomes thinner, primary air is able to pass through to combust the carbon. Quite obviously it is necessary to provide for a precise control of air, in order that the amounts passing both over and through the grate may be correctly proportioned. It is equally obvious that the principle upon which the stoker operates imposes very definite limits upon the fuel which can be burned.

Also from a consideration of those principles, and particularly of the fact that there must at all times be a mass of unburned or partially burned fuel upon the front of the grate, it is clear that the stoker cannot be as responsive to sudden changes of load as a machine which embodies the sprinkler principle.

The ideal fuel for the coking stoker is a coking coal, without dust or large lumps, and sufficiently rich in volatiles for the coking principle to be fully exploited. The coking stoker will *not* burn anthracite or coke breeze. Certain semi-coking coals may be burned, providing they are sized so that excessive riddling does not take place. Within these limitations very efficient combustion is obtainable, and it is possible to burn, smokelessly, very difficult fuels, which would not burn efficiently and cleanly by any other method.

The sprinkler stoker, as its name implies, does not employ the coking principle, but is provided with a controllable feeder gear, and a throwing shovel which projects the fuel in small quantities to each portion of the grate in turn. It is the embodiment of the "little and often" maxim, so well known in boiler houses, and so seldom carried out.

Either fixed or moving grates can be provided. A rather wider range of coals can be burnt on this stoker, because non-coking coals are included in the category of fuels which can be handled successfully, but the machine is rather more sensitive to sizing. Unless the coal is so graded that it may be properly projected by the thrower, trouble may be experienced. It is rather more difficult, too, to burn highly volatile coals which have a natural tendency to produce black smoke, especially in the restricted combustion space of the shell boiler. On the other hand, low volatile fuels, even anthracites and semi-cokes or coal and coke mixtures, which could not be burnt on the coking stoker, can be handled quite successfully by the sprinkler. Also, because the volume of unburned or partially burned fuel upon the grate at any given instant is much smaller, the sprinkler stoker is much more flexible in its response to load changes. The stoker is readily adaptable for either natural, induced, forced, or balanced draught. Maintenance is generally higher on the sprinkler stoker, due to the rather larger number of working parts, and to the more violent action with resultant stresses generated in the throwing

FIG. 38. COKING STOKER APPLIED TO SHELL BOILER
(*Bennis Combustion Co., Little Hulton*)

mechanism which must propel the fuel to the extreme end of the grate.

From the foregoing, it will be seen that there are certain conditions to which either class of stoker could be applied almost equally well, and that in such cases the selection is not easy. If the range of available fuels comes within the capabilities of either stoker, then the deciding factor might well be the characteristics of the load likely to be developed. For steady loads or for loads whose fluctuations can be predicted with reasonable certainty, and which follow a definite cycle, the coking stoker would be the first choice for its inherent simplicity, low maintenance, and for the fact that the coking principle allows smokeless combustion to be attained more easily. Rough slacks with coking properties are more easily handled by the coking stoker, which has no throwing mechanism to be affected by variation in size. Definitely non-coking fuels *cannot* be burnt successfully, and the sprinkler stoker becomes necessary where such fuels are alone available.

A difficult proposition is presented when highly volatile fuels, which would normally burn well on the coking stoker, must be used for violently varying loads. Where a range of boilers is installed, consideration might be given to installing some of each type of stoker, the numbers being so proportioned that the base load is taken by the coking stokers. The responsive characteristic of the sprinkler machines is utilized to take the peaks.

A well-known type of coking stoker is shown in Fig. 38, as applied to a shell boiler. The design illustrated is intended for natural or induced draught. The section given in Fig. 39 clearly shows the principle of working. The hopper (3) contains about 2 cwt of fuel, which drops into a feed box containing a ram. This ram, the stroke of which may be adjusted to suit particular conditions, has a reciprocating motion which pushes the fuel over a shaped distributing nose on to the coking plate at the furnace front. The grate bars, the stepped construction of which is well shown in the section, are actuated by cams mounted upon a camshaft which is mounted across the boiler front just below the grate level. The cams are so arranged that all the bars move towards the back of the grate carrying the coal with them. Certain of the bars then remain in that position, while the others are drawn forward again. The stationary bars serve to restrain any forward movement of the fuel during this withdrawal. The withdrawn bars then remain stationary, while the remaining bars are brought back to their original position. All the bars then move towards the back of the boiler, and the cycle is repeated. The result is a steady movement of the fuel bed towards the back of the boiler.

The speed of the movement is adjustable in relation to the rate of coal feed, so that it is possible to arrange for the best conditions, with the fuel just burned off as it arrives at the extremity of the back end of the bars. The ash falls over into the ashpit, from whence it is removed by hand through the special ash-door whose hinge may be seen immediately beneath the back sealing device shown at (8) in Fig. 39.

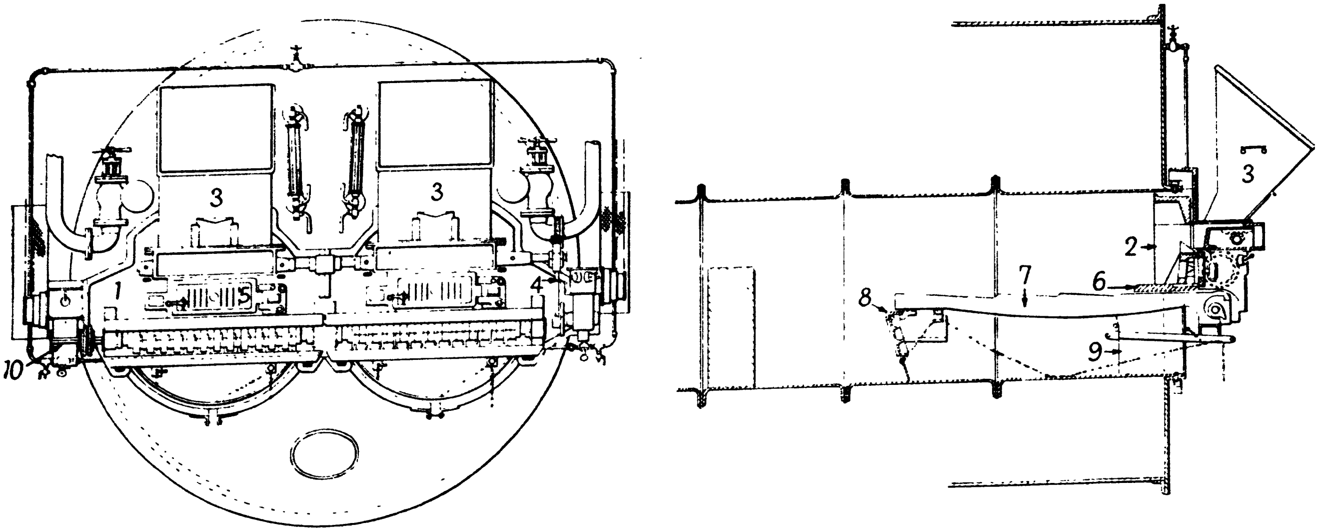


FIG. 39. SECTION OF BENNIS COKING STOKER

(*Bennis Combustion Co., Little Hulton*)

Arrangement of "Bennis" natural draught coking stoker applied to 9 ft burning Lancashire boiler:

1. Stoker front.
2. Baffle plates and front protection.
3. Hopper.
4. Stoker feed mechanism.
5. Fire doors.
6. Coking plate.
7. Self-cleaning grates.
8. Back sealing device.
9. Patent air regulating damper.
10. Stoker driving gear.

Air Control

The necessity for correct and precise air control has already been emphasized. With natural or induced draught the correct total amount of air is determined by the amount and characteristics of the coal which is to be burnt, and the draught is produced by the chimney stack or fan, or a combination of both. It is controlled by the adjustment of fan speed or

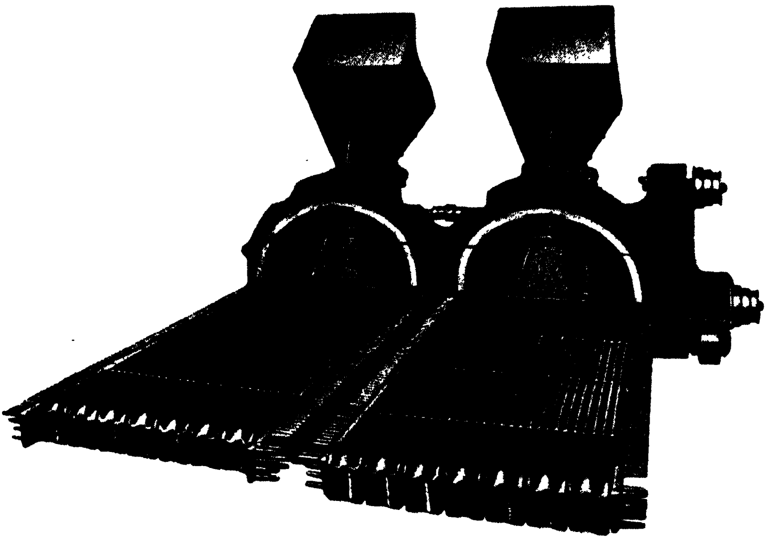


FIG. 40. BENNIS SPRINKLER STOKER
(*Bennis Combustion Co., Little Hulton*)

main dampers, or again by both. But this control only affects the total air supply, and does not exert any influence upon the distribution over or under the grate, which, unless special means are provided, is some function of the resistance of the fuel bed. Since this resistance is in itself affected both by the thickness of the bed, which in its turn is affected by the load, and by the class of fuel burned, it is obvious that unless the air supply can be controlled in the matter of distribution as well as in total quantity, not only will severe limitations be imposed in the choice of fuel, but efficient combustion could not be expected throughout the whole range of load. It follows from this that, in the consideration of stoker design, the more efficient the arrangements for air distribution the wider will be the range of fuels which can be handled.

A very simple arrangement by which the desired air control may be carried out at will by the operative, is shown at (9) in Fig. 39. The device is self-explanatory. At a point which, in normal working, corresponds to the thickest portion of the

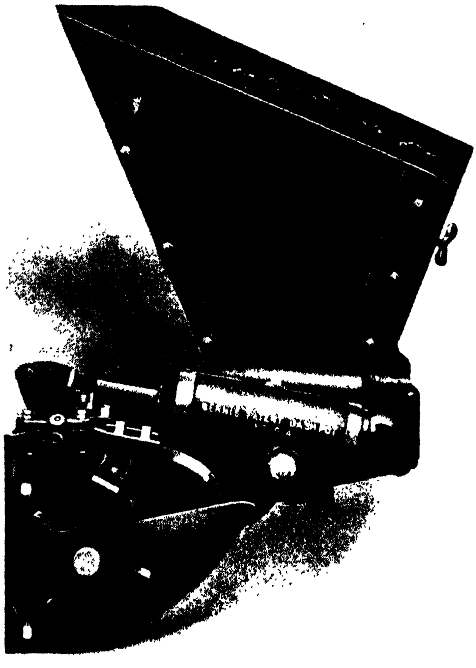


FIG. 41. THROWING MECHANISM OF SPRINKLER STOKER
(*Bennis Combustion Co., Little Hulton*)

firebed, and just to the rear of the coking plate, a hinged well-fitting damper is suspended below the grate bars. This damper, which is conveniently operated by the handle shown, is in effect an air proportioning valve, whose function is to admit a greater or less porportion of the total air to the underside of the fire. The intelligent operation of this damper, in conjunction with the slots in the fire door, which permit some control of the secondary air, greatly enhances the range of fuels which may be confidently handled by the stoker illustrated. With suitable draught, very high rates of combustion are obtained with these stokers, with consequent high fuel bed temperatures and good heat transfer. Rates as high as 55 lb fuel per sq ft of grate area per hour can be realized.

Bennis Sprinkler Stoker

A type of sprinkler stoker is shown in Fig. 40. The coal is thrown or sprinkled over the whole area of the firebed. The

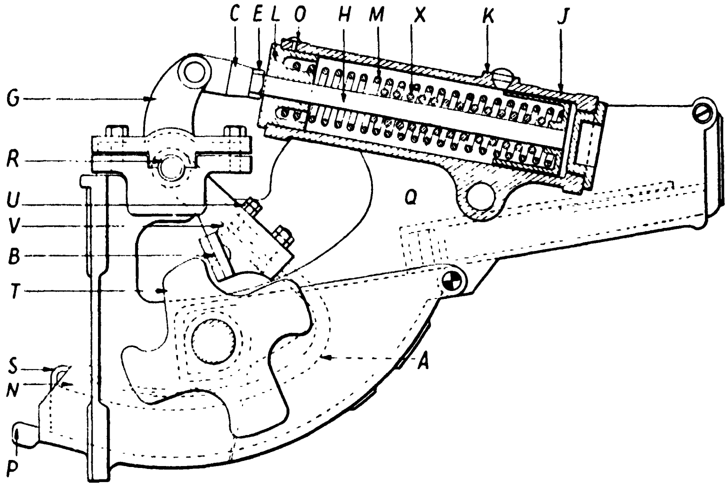


FIG. 42. DETAIL OF THROWING MECHANISM
(*Bennis Combustion Co., Little Hulton*)

P	Lug	*C*	Crosshead
N	Shovel arm	*E*	Crosshead nut
S	Shovel	*L*	Adjusting nut
T	Cam	*O*	Lock screw
B	Tripper	*H*	Piston rod
V	Tripper arm	*M*	Spring
U	Tripper locking bolts	*X*	Internal spring
R	Rocking shaft	*K*	Oil hole
G	Lever arm	*J*	Piston

motion is continual, and the quantities of fuel are so adjusted and distributed that the firebed surface is never deadened.

The throwing mechanism is shown in Fig. 41, which illustrates the four-armed throwing cam and the fittings related to it. The varying width of the arms will be noted. The effect of this width difference is to vary the throw of the spring-actuated shovel, so that for each revolution of the cam four sections of the grate, each about 18 in. long, are supplied in turn. The distance the fuel is thrown may be adjusted by altering the tension on the throwing spring by the nut on the pneumatic cylinder. Very slight movements of this nut make a considerable difference to the throw, and it is essential that care is exercised when this operation is carried out for the first time. It should always be done under working conditions because of the effect of the draught on the carry-over of the coal. Referring to Fig. 42, the procedure is as follows—

Loosen the set-screw *O* and adjust the nut *L*. If the coal is thrown too near the furnace door the tension must be increased by screwing the nut up, that is to say in a clockwise direction. The tension may be increased in this way, until there is a slight carry-over of unburned coal at the back of the grate, at which point the nut is slackened back anti-clockwise one-half turn. The set-screw is then tightened, and the adjustment should be correct.

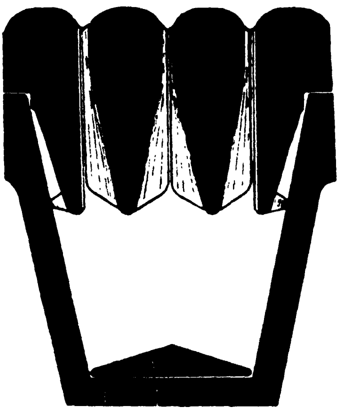


FIG. 43. SECTION OF TROUGH BARS FOR AIR DRAUGHT STOKER, SHOWING ROUND-TOPPED GRIDS AND AUTOMATIC ASH CLEANING SLIDE

(*Bennis Combustion Co., Little Hulton*)

The quantity of the fuel is regulated by a screw, by which the movement of the feed mechanism may be adjusted while the machine is in motion. The air supply to the furnace may be either natural or induced draught, in which case the principles just described for the coking stoker are adopted. Either forced or balanced draught may be provided for. The methods used for either of the two are similar.

The particular stoker shown is fitted with what is known as the compressed air furnace, with self-cleaning bars, forced draught being supplied in this case by means of superheated steam injected into the trough bars. A section of one of these bars is given in Fig. 43. Each trough holds four rows of firebars, which are interlocking, and about 2 ft long. The rounded top of the firebars tends to prevent the adhesion of clinker to individual bars, an object which is further assisted by the self-cleaning movement of the grate. The troughs themselves are laid closely side by side, and are free to slide longitudinally, and independently of each other. A camshaft as previously described for the coking stoker is mounted across the boiler front, and a similar motion takes place.

All the troughs carrying the bars move forward together for a distance of about 2½ in., carrying the mass of the firebed with them, after which they are individually withdrawn to their original position. There are no steps in the bars as in

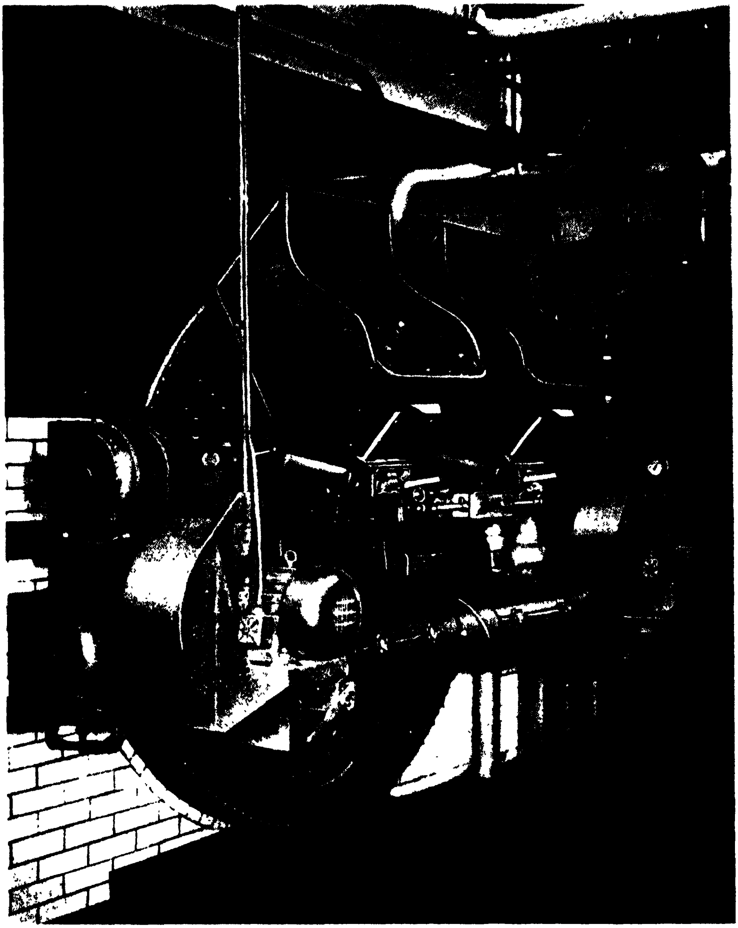


FIG. 44. AIR DRAUGHT SPRINKLER STOKER WITH INDEPENDENT DRIVES
(*Bennis Combustion Co., Little Hulton*)

the coking stoker, so that the movement of the bed is not so positive, but a "shuffling" action takes place, which is most effective for the purpose of cleaning, and at the same time there is a general tendency for the bed to move towards the back.

Fig. 44 illustrates a modern type of air draught sprinkler stoker. The general design does not differ greatly from the compressed air pattern, except that the draught under the grate is supplied by fans instead of steam jets. An air duct is provided along the front of the boiler, from which specially proportioned nozzles project into the troughs. The nozzles do not move with the troughs but remain stationary. The

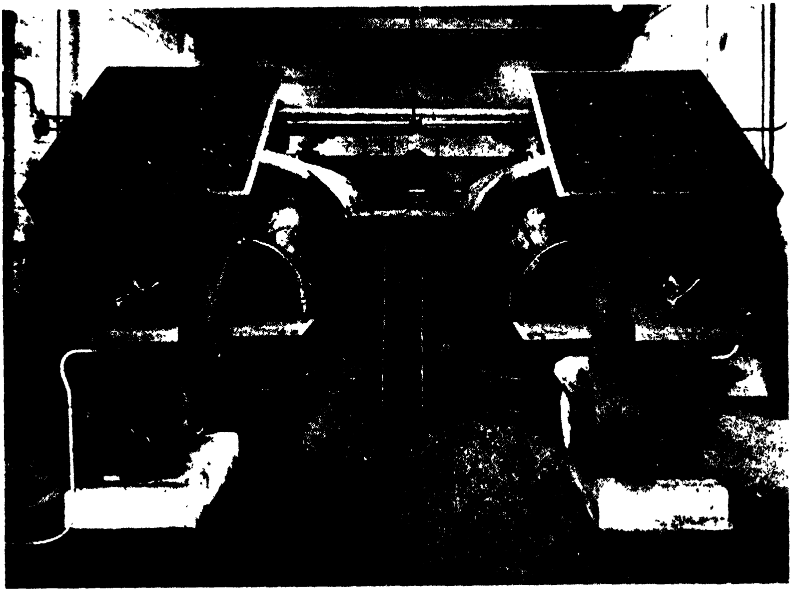


Fig. 45. Hope Underfeed Stoker
(Hope Heating & Lighting Co., Smethwick)

resistance against which the fans are required to work is usually of the order of 4 to 6 in. water gauge. Adaptations of the same design are used for balanced draught or preheated air.

Underfeed Stokers

The third class of stoker mentioned at the beginning of the present chapter as suitable for shell boilers is the underfeed stoker. There are two main types—the pot and the retort, the latter being the more usual for the larger sizes. There is no method of hand-firing analagous to the principle on which either of these two designs operates. The coal is fed from a hopper and pushed, either by means of an auger or by rams, into a firepot or retort, in which a hot zone is maintained by suitably placed air blasts. Thus, the green coal is constantly being moved upwards into proximity with the hot zone at the surface of the fire. Distillation commences at some point below the fire surface, and the volatiles pass through the hotbed, where they are ignited. Continuing its upward movement, the glowing coke comes under the influence of the transverse air blast, in which combustion is completed.

A straightforward and well-designed type of stoker is shown in Fig. 45. This fundamental design has many applications. The smaller sizes are put to such diverse uses as the heating of bakers' ovens, lead-melting pots, metallurgical furnaces, etc., which are not the concern of the present work. The type illustrated is suitable for steam boilers, or hot water boilers for central heating, and kindred appliances. The simplicity of the design makes it very suitable for the application of thermostatic control, of which there are very many ingenious examples in industry. Essentially, the machine consists of a hopper fitted to a coal tube, which is directly connected to the interior of the firepot. In the coal tube is an auger, so mounted that it lies along the bottom of the hopper, and protrudes into the firepot at its remote end. When the auger is rotated the fuel is screwed along the coal tube into the firepot, and the rate of feed obviously depends on the rate of movement of the auger. A small motor performs the dual duty of actuating the auger and driving the fan, which is an essential part of the equipment.

A variety of arrangements is used, but in principle there is much similarity between them. The motor is suitably mounted to drive the high speed element of a worm gear, either directly or by means of a short, compact Vee belt drive or chain. The fan may be mounted either on the motor shaft, or on the high speed shaft of the gear-box. The worm driven by the motor engages in a worm wheel, which is so arranged that it gives a reciprocating motion to a pawl, which may be adjusted while the machine is in motion. The pawl in its turn actuates a ratchet, which is directly connected to the coal screw. By the reciprocating motion of the pawl the coal screw is rotated by a series of impulses, with a pause between each. The length of the impulses, or in some cases the period of time between them, is adjustable, so that the rate of fuel supply may be varied between fairly wide limits. Some makers fit three-speed pulleys to the smaller sizes of stoker, so that the coal may be fed at three fixed rates. The stoker must be stopped whenever it is desired to vary the feed, which for many purposes might well be a serious disadvantage, but for small sizes only and for steady loads, the increased simplicity of the gear-box compensates for the lack of flexibility which such a design imposes.

Surrounding the firepot is an air box which is connected to the fan outlet by a suitable duct, and mounted on the air box are hollow tuyère irons, so arranged that their openings direct the air blast across the pot just below the working surface of the fuel. Openings are also provided at the outer circumference of the tuyères, to ensure the complete combustion of fuel pushed over the sides of the pot. The fan being driven from the motor or gear-box, runs at constant speed, and the air quantity is controlled by a simple damper plate, so arranged that it may be screwed towards or away from the fan inlet, and secured in any desired position by means of a lock-nut. A very exact adjustment of the air/coal ratio is possible.

The ideal coal for these stokers should be graded, and of the free-burning bituminous class. Semi-anthracites can also be handled successfully. Sizes should not normally be below washed peas, but during the War, when graded fuels of any class could be supplied only for special purposes, many stokers of the underfeed type were run on fuels far inferior to those mentioned as ideal, not, however, without some initial difficulty. In one instance at least, in the author's knowledge, certain melting processes were kept going for many months on a poor quality open-cast coal, sized down to fine slack, and up to ¾ in. This extremely unlikely fuel gave reasonably good results, provided that the utmost care was taken to feed it to the hoppers as dry as possible. If this precaution was not taken, trouble was experienced with the finer portions binding in the coal screw so tightly as to cause stoppages. Some measure of success was also achieved by the use of preheated air, with which mixtures of open-cast dry slack and coke dust were burnt. Apart from the binding mentioned, no abnormal mechanical or other trouble developed, but with the coke dust, the brickwork of the furnaces deteriorated more rapidly.

When supplied with suitable fuel, however, the performance of these stokers can be highly satisfactory. With the aid of a CO_2 indicator or recorder, or by routine tests with the Orsat apparatus, optimum conditions can be established and maintained with certainty, but even without the help of these invaluable instruments, the basic principle of the firing method compels the attention of the operator to the vital air/coal ratio.

If the fan damper be opened too wide for the prevailing

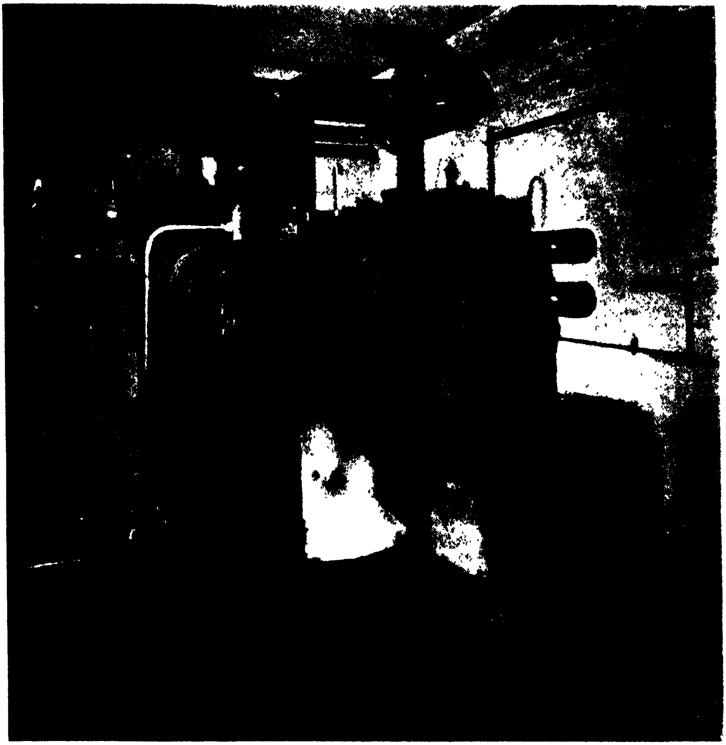


FIG. 46. APPEARANCE OF FIRE WITH UNDERFEED STOKER
(*Hope Heating & Lighting Co., Smethwick*)

rate of coal feed, so that large quantities of excess primary air are admitted, the coal tends to burn away too rapidly, and the hot zone assumes too low a position in the firepot, with the consequent possibility of damage to the tuyères or even the end of the coal screw. On the other hand, if too little air is supplied, the coal builds up on the firepot, and with certain strongly-caking coals a black mass forms in the centre, so dense that the air cannot penetrate. The net result is that a quantity of half-burned combustible is pushed over the side of the firepot, where much of its value may well be lost because it is no longer under the direct influence of the central air blast. Either of these conditions is at once apparent to the operator, and as they are easily corrected by simple adjustments there is little excuse for their continuance. The appearance of the fire when correctly operated is shown in Fig. 46.

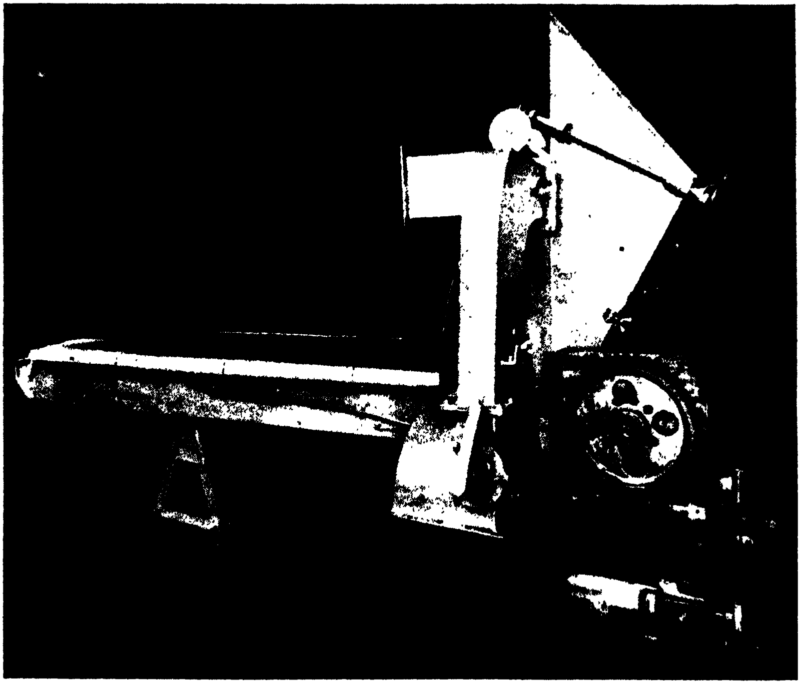


FIG. 47. "OLDBURY" STOKER
(*E. Danks & Co. (Oldbury) Ltd.*)

The "Oldbury" Stoker

The fourth type of stoker which may be applied to shell boilers is rather more unusual but is making its appearance at the time of writing. This is the "Oldbury" chain grate stoker illustrated in Figs. 47 and 48. In many respects the stoker follows the design of the larger and more conventional patterns applied to water tube boilers, but there are some features which are peculiar to this design alone. The motion of the grate is continuous. The air supply ducts are of steel, and form the chassis of the stoker. Forced draught is normally supplied, and a portion of the air is tapped off the main duct to supply secondary air over the fire.

The outlets for this are seen in Fig. 47. The arrangements for control of fire thickness are more or less conventional in design, consisting of a refractory protected guillotine door, operated by hand through a worm and worm wheel. The

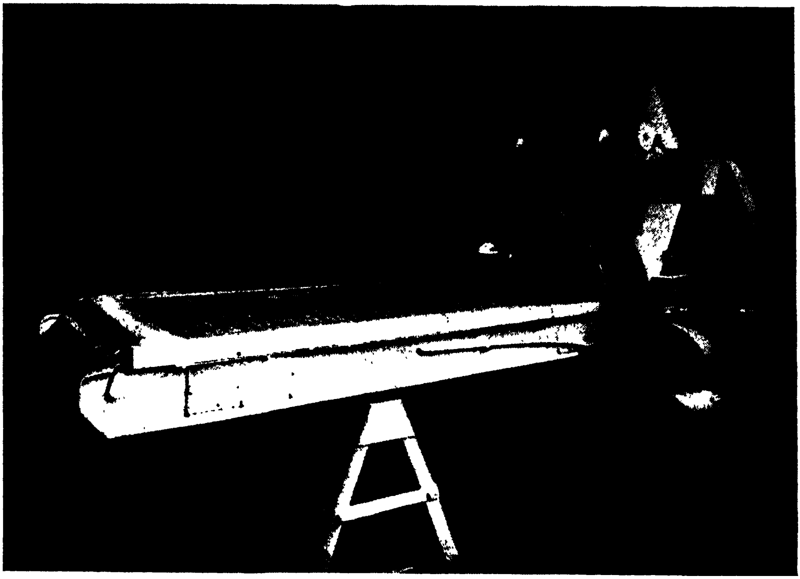


FIG. 48. "OLDBURY" STOKER, END VIEW
(*E. Danks & Co. (Oldbury) Ltd.*)

whole unit is compact and self-contained, and it is possible to withdraw the whole machine from the flue tube if desired. It is claimed that a very wide range of coals can be burned on this stoker.

The Spreader Stoker

Although some of the stokers already discussed are occasionally applied to water tube boilers in the smaller ranges, i.e. up to about 20,000 lb of steam per hour, it is far more usual to find the more orthodox chain grate or retort designs so applied. Before dealing with them, however, it will be well to mention a type of stoker which possesses some of the features of the sprinkler machine, and also of the chain grate. This is the spreader stoker, some designs of which have been developed in America, and are now making their appearance in this country.

The machine consists of a hopper, at the outlet of which is a powerful throwing mechanism, which projects the coal over

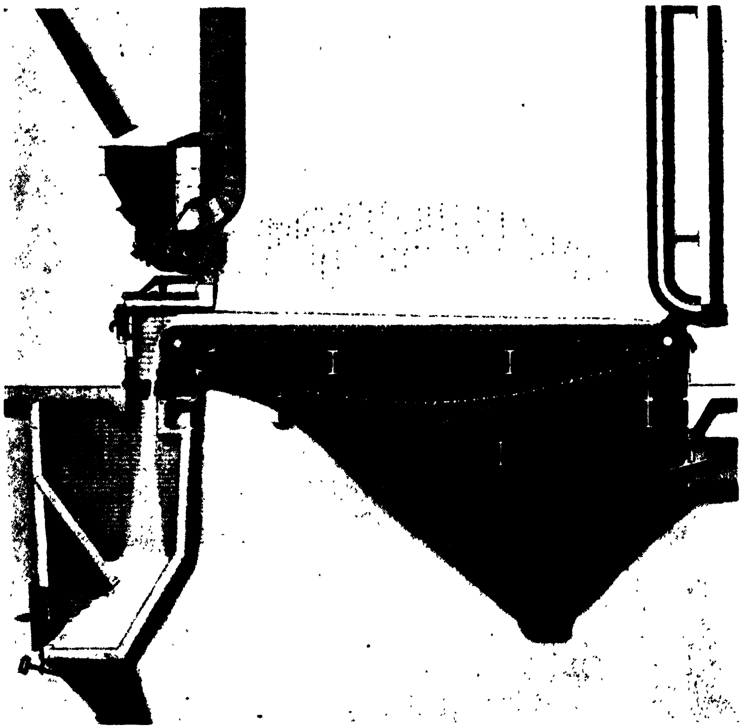


FIG. 49. SPREADER STOKER
(*Babcock & Wilcox Ltd.*)

the whole of the grate area. The finer particles burn in suspension, while the heavier fall to the grate where combustion is completed. It is usual to work with a fuel bed thickness of about 4 in., though this must depend to some extent on the type of coal being burned. Coals with high ash content would be worked with a thicker bed than higher grade fuels for a given duty, for the reason that the ash would be deposited much faster, the actual rate of deposition depending on the ash content.

Three types of grate are in general use with these stokers. One pattern is similar in conception to the moving bar grate

FIG. 50. STYLE 6 BABCOCK STOKER APPLIED TO WT BOILER
(*Babcock & Wilcox Ltd.*)

used for the more familiar sprinkler stoker, and, in this case, the great difference between the spreader and the sprinkler lies in the method by which the coal is burnt, which is rather a question of air distribution and form of coal projection than a fundamental departure in grate design.

The second form is the louvre grate, in which provision is made to lift sections of a fixed grate by external means, allowing the ash to drop through to the removal gear underneath. The grate is built up in narrow hinged sections, and a number of these may be opened at the same time.

The third design includes a moving chain grate, similar to that used for the ordinary chain-grate stoker, but moving

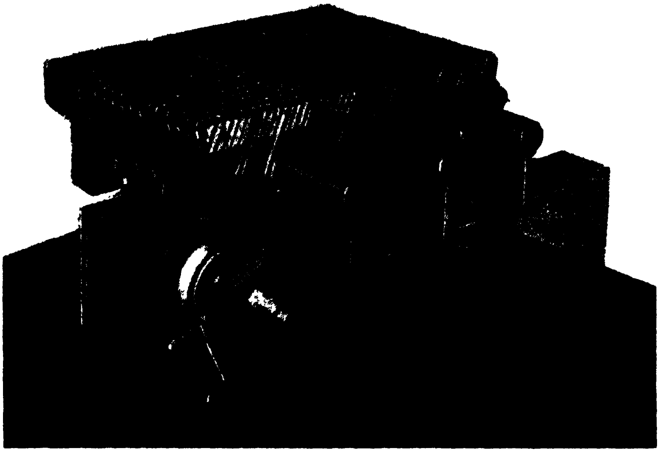


FIG. 51. DETAIL CONSTRUCTION OF BENNIS CARRIER LINK GRATE
(*Bennis Combustion Co., Little Hulton*)

towards the front of the setting instead of to the back, as is more usual.

The spreader stoker is applied to water tube boilers, and has already been built for evaporations up to 250,000 lb of steam per hour. One type is shown in Fig. 49.

Chain-grate Stokers

What may be termed the fundamental type of chain-grate stoker as applied to water tube boilers is shown in Fig. 50. From this basic design, which is still very frequently to be found in industrial installations, the more recent interpretations of the principle have developed. Most of these improvements are concerned with air supply and distribution, and take the form of additional enclosing panels and air ducts, so disposed upon the fundamental design that a very wide range of fuels may be burned.

The chain links themselves have been the subject of much thought on the part of designers, both with regard to heat-resisting properties and arrangements for adequate air passage through the grate. One ingenious departure is shown in Fig. 51, which illustrates the Bennis Carrier Link. A feature of this construction is that the links themselves are not called upon to carry any part of the working stresses of the grate, but simply act as a conveyor for the fuel.

Referring again to Fig. 50: the bare essentials of the design

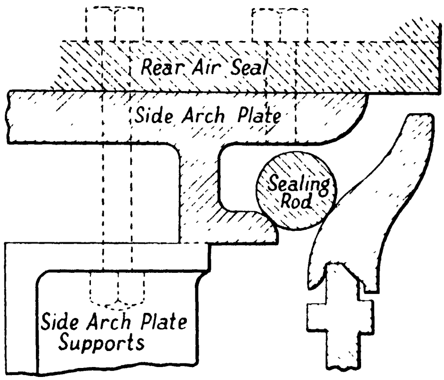


FIG. 52. DETAIL OF SIDE SEALS FOR CHAIN-GRATE STOKER, STYLE 6
(*Babcock & Wilcox Ltd.*)

are means to support the boiler brickwork adjacent to the stoker, and to provide a suitable setting for the carriage supporting the chain; means to prevent the ingress of air at the side of the grate, and the carriage itself. The carriage includes the driving mechanism for the chain, with its variable speed gear, the coal hopper and the guillotine door for varying the fire thickness. The setting for the carriage is built up of two arch plates or sill plates, which are supported on short vertical stanchions. These plates are heavy castings, specially shaped and not liable to distortion, and the refractory which forms the combustion chamber is built up on them. The carriage consists of suitable side plates, mounted on roller wheels, and braced by cross members. The whole assembly runs on shallow rails, and may be withdrawn from the setting at will. The sprocket shaft which drives the chain is mounted across the front of the carriage and a roller which supports the rear of the chain is carried at the back.

Arrangements for ash dumping and air sealing at the rear are provided by a built-up ash valve, which is operated from a lever external to the setting, and by dumper plates which rest upon a cross member and the moving chain.

Air sealing at the grate sides is ensured by the device in Fig. 52. Specially shaped loose plates are fitted upon the sides of the carriage, and when the latter is in position roller seals are inserted between these and the sill plates. The inequalities in the grate edges are taken up by slight movement of the seal plates, and by compensating movement of the rollers. The brickwork is brought out to project over the edges of the sill plates so as to protect the sealing plates.

The driving mechanism consists of a multi-speed gear-box, mounted upon the front of the carriage side plate, and chain-driven from an external shaft or motor. Speed changes are obtained by the operation of the hand wheel on the front of the gear-box. From the gear-box the drive is taken through a safety clutch to a worm shaft, and thence to a worm wheel which is mounted upon the sprocket shaft driving the grate itself. The clutch can be released at will, so that if necessary the grate can be turned by hand. The gears run in an oil bath, so that periodical attention is all that is necessary. Suitable lubrication through oil cups is provided for other parts of the stoker.

The hopper and the refractory-protected guillotine door are built up at the stoker front, and the whole assembly is bolted to special cast jambs. By releasing a few bolts the whole can be withdrawn from the setting, upon the rails provided. The replacing of links, or the complete rebuilding of the grate, is well within the capacity of works maintenance staff, and requires no special skill.

When rebuilding, a platform is set up at the stoker front, level with the top of the driving sprockets. On this, a few rows of links are assembled, with the link bars fitted and secured at the ends with split pins. A cable or chain is attached to the first row of links by a hook passed through the link rod, taken over the back roller and brought to the front of the grate, where it is secured to a hauling tackle suitably anchored to a fixed point. As successive rows are assembled, the link bars are put in position and pinned up, and the chain hauled over the sprockets. The clutch of the driving gear may be released, and the driving mechanism worked by hand to assist the forward movement of the chain. The hauling tackle at this stage is simply used to take the weight of the slack. When the chain has passed over the back roller the main weight is taken by the supporting rollers which are fitted between the carriage frames, and the hauling tackle is used to help the end of the chain over each roller in turn. When the requisite number of link rows has been assembled the lower end of the

FIG. 53. SET-UP FOR RELINKING STYLE 6 STOKER
(*Babcock & Wilcox Ltd.*)

chain is packed up before removing the hauling tackle, and the last link rod inserted, with the chain tightening nuts fully slackened off. The method is made clear in Figs. 53 and 54.

With other makes of stoker, the method may differ in detail but the principle is the same in all cases. All chains of this type consist of end and centre driving links, and the requisite number of common links to make up the width. When building up a new chain, a few sample rows should always be built up.

The distance between the sprockets should be carefully measured, and the driving links laid out to suit. The number of common links fitted between the drivers should be such that there is at least ¼ in. of space in excess of that taken up by the common links when tightly pressed together. This tolerance should be left between each set of drivers. It is absolutely necessary that the completed assembly should work freely, and it is rather better to err on the loose side than to build up a grate which may have a tendency to jam in service.

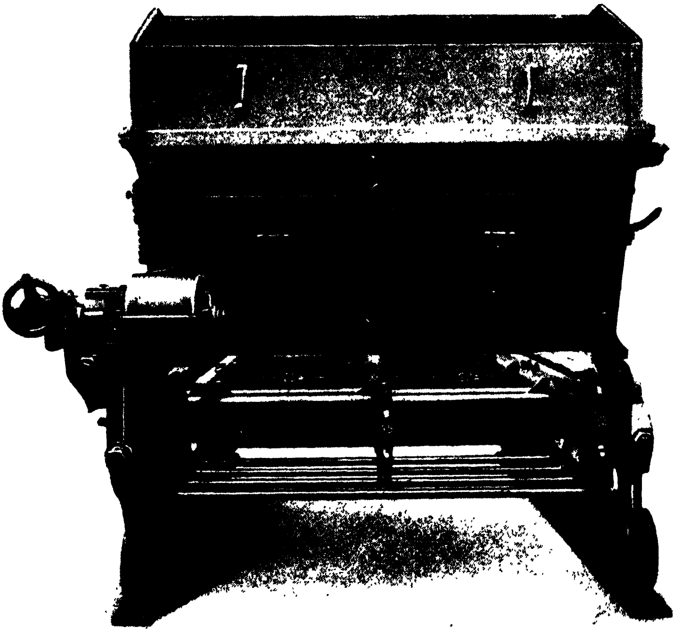


FIG. 54. VIEW OF STYLE 6 STOKER BEFORE RELINKING
(*Babcock & Wilcox Ltd.*)

Although the replacement of worn or broken links is a fairly simple matter it does involve the stopping of the grate for sufficient time to break the chain, and withdraw the link bars as required, in order to fit the new links. Unless the boiler is one of a range, in which case it might be possible to carry out the work during a period of low load, such an operation means that the boiler must be taken completely out of service. To obviate this difficulty Messrs. Babcock & Wilcox have devised an emergency replacement link. This link is made in two halves which interlock in position, as shown in Fig. 55. In the event of a link breaking or being so badly worn or burned that replacement becomes imperative, this device provides a means of keeping the plant going, at least for the time being. It is only necessary to stop the grate for sufficient time to break out the defective link, and slip the replacement into position. The chain need not be broken.

The basic form of stoker described is very frequently to be found, both in its simplest form for induced or natural draught, or fitted with various modifications, so that forced or balanced draught may be used. When so fitted, the side frames are extended beyond the front of the chain, so that the whole of the stoker front below the grate level may be enclosed, to enable an air pressure to be applied below the firebed.

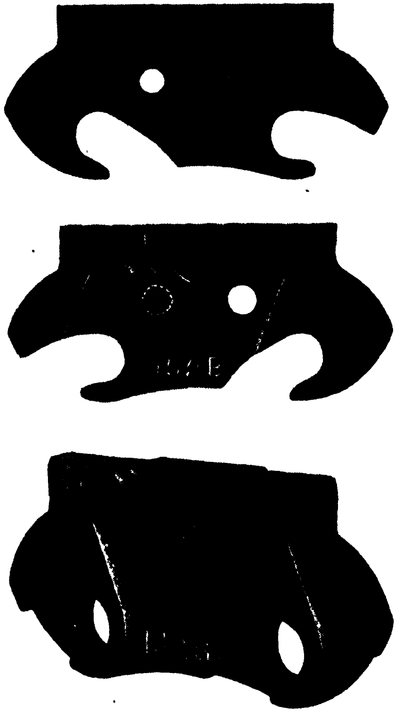


FIG. 55. BABCOCK REPLACEMENT LINK
(*Babcock & Wilcox Ltd.*)

Compartment Stokers

Yet another modification is the compartment type stoker in which air troughs are so fitted that air may be admitted to any desired portion of the grate length. Very excellent control of combustion conditions is possible by this means, since the air to each part of the fuel bed may be adjusted with great nicety to meet prevailing conditions.

A stoker of this type is shown in Fig. 56. This machine is designed for the carrier link grate, which has already been described. The forced draught is applied to the hopper shaped air ducts, each of which is provided with well-fitting dampers with machined surfaces. Control is by hand, through worm and worm wheel from the outside of the setting. Special sealing arrangements are provided to prevent the leakage of air between troughs, or through the ironwork comprising the grate surface. A front view of the completed stoker is given in Fig. 57.

The precise control of air supply, both as regards quantity and point of application, enables a very wide range of fuels to

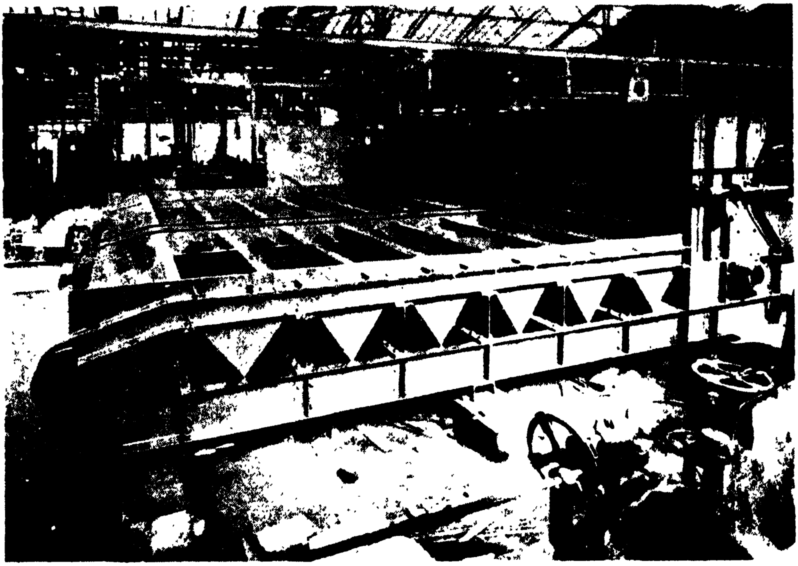


FIG. 56. BENNIS UNIT STOKER SHOWING AIR TROUGHS
(*Bennis Combustion Co., Little Hulton*)

be burned efficiently. It is claimed that this stoker will handle a variety ranging from coke breeze to highly bituminous coal.

Although the modern mechanical stoker is a sturdy and robust piece of mechanism, it must not be forgotten that it is a machine which is required to work under very severe conditions, and must therefore receive its share of maintenance. The greatest enemy is probably the abrasive dust which is inseparable from the boiler house, and it goes without saying that cleanliness pays. Routine cleaning and strict attention to correct lubrication are two potent factors in the reduction of maintenance costs. Whenever the boiler is shut down, attention should be given to the stoker. Far too often it is left to look after itself while the efforts of the maintenance section are concentrated on the boiler. Unfortunately, it is rather the rule than otherwise to allow far too little time for the necessary work to be done. This is frequently unavoidable for the very simple reason that stand-by plant is not always available.

The position of the works engineer faced with the problem of shutting down a single boiler on Saturday afternoon, opening

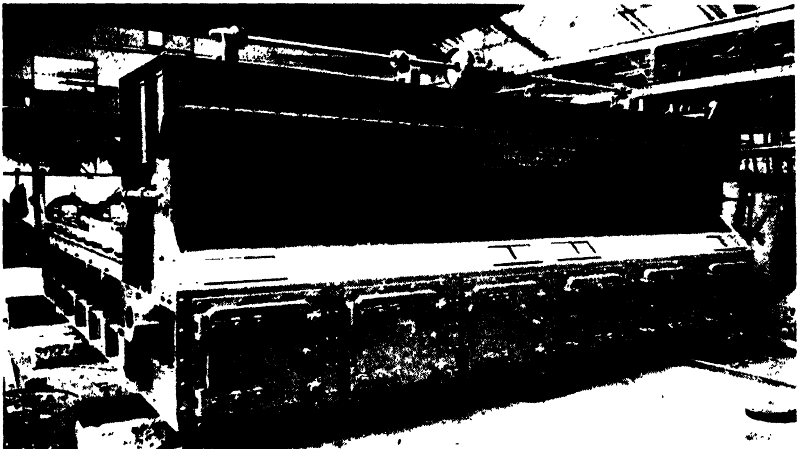


FIG. 57. COMPLETE UNIT STOKER, FRONT VIEW
(*Bennis Combustion Co., Little Hulton*)

up, rough-scaling, flueing, and closing up ready to start steaming on Sunday night, is, to say the least, unenviable. Yet not one but many works expect this to be done. Under such circumstances, there is little wonder that stoker maintenance is neglected, and that the plant is allowed to run as long as it will, that is to say until work upon it can no longer be delayed.

Something can be done by carrying an adequate stock of the more vulnerable parts, which can be quickly changed during the week-end. The parts removed can then be either replaced or repaired at leisure in the shop, or scrapped, and a new part purchased for stock. This policy may seem to be expensive in first cost but it undoubtedly pays, for the quick availability of a new part may save a delay or even a shut down which might well cost far more than a complete new stoker.

Spares

It is difficult to advise, except in the most general terms, as to the spares which should be carried. So much depends on the circumstances in which the particular plant is expected to operate. The greater the amount of stand-by plant available, the less the spares which need be held available.

For shell boilers, with sprinkler stokers fitted with renewable cam faces, some spare faces should be available, and it is well also to carry a spare star wheel. Where moving or self-cleaning grates are fitted a few grate bars would form an important item of the stock. This applies also to coking stokers. The maker's advice should always be sought on the matter of spares, and apart from this, experience with the actual machine soon reveals those parts which are most likely to need renewal.

For chain-grate stokers it is well to carry sufficient links to replace half the grate with common links and centre drivers. Some end drivers can also be stocked, but it is usually found that these will outlast several sets of the more vulnerable centre links. It also pays to carry one complete gear-box, which in emergency can be fitted quickly as needed. Where both right- and left-hand stokers are in use this must be provided for. It is useful also to carry a main worm drive wheel and worm.

An important factor in the choice of spares is the facility with which a particular part can be obtained from the makers. If deliveries are difficult it is the more essential that such parts should be available from maintenance stock, and ordered up for replacement immediately they are taken into service.

The principle of selection should be that with the least possible expenditure the stock should form a complete insurance against involuntary stoppage, by providing a range of replaceable parts which, by their availability, reduce to a minimum hurried (and frequently botched) week-end jobs.

CHAPTER V

PIPEWORK

THE modern steam pipe is a well-designed and beautifully made appliance, upon which a great deal of care and much research work have been lavished by the manufacturer. Unfortunately, the conscientious work of the maker is too often nullified by the user who, by careless selection and installation, and subsequent neglect, subjects the pipes to abuses, which only their inherent excellence enables them to withstand. Of these abuses perhaps the most common are those associated with malalinement and insufficient support. Apart from the tremendous stresses which develop in the pipe flanges from this particular cause, malalinement in a vertical direction, i.e. sagging, allows the formation of water pockets in positions from which they can only be dispersed by the insertion of specially placed, and otherwise unnecessary, traps.

Supports and Support Methods

To avoid sag and its attendant evils, it is absolutely necessary that the pipes should be well supported throughout their length. The correct support spacing is a function of the pipe diameter and thickness, but as a guide only, the following figures may be taken as correct.

2 in. bore	8 ft
3 in. ,, ...	10 ft
4 in. ,, ...	12 ft 6 in.
5 in. ,, ...	15 ft
6 in. ,, ...	18 ft

For the larger flanged pipes the placing of the flanges and other fittings will to some extent influence the support methods. Flanges should be supported on both sides, and the same applies to heavy valves, expansion bends, etc. Expansion bends which are installed horizontally to avoid drainage troubles require additional support at the apex of the bend. Existing means of support, such as building principles and the like, will naturally be used wherever possible, but the temptation to allow building features to dictate the support spacing entirely must be resisted if future trouble is to be avoided.

For road crossings and similar situations the method shown in Fig. 58 is both cheap and effective, and enables wider support spacing than normal to be used. Here again, the actual placing of the supports must be dictated by the pipe sizes, and the support section and spacing must be selected to the particular conditions. For longer distances or multiple crossings, where several pipes take the same route, the bridge method may be used. Light braced structures are built, either with angle iron of suitable section or from pipe. Fig. 59 indicates the type of structure described. By this means considerable spans may be negotiated, but it is important to note that the bridge must be so constructed, and the pipes so disposed in the structure,

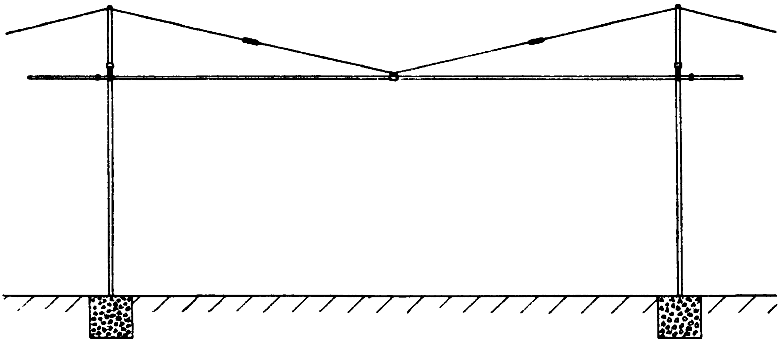


FIG. 58. STRAINER METHOD OF PIPE SUPPORT

that they are correctly supported and take no part of the bridge stresses.

Expansion Bends and Joints

Whatever type of support is used, the method of attachment or link between the pipe and the structure must be such that free movement is allowed in the direction of the length of the pipe. Only by the strictest attention to this point can excessive stresses in the fabric of the pipes and flanges be avoided.

The magnitude of the movement is a function of the coefficient of expansion of the metal, and the working temperature of the pipe, and may be assessed as follows—

Let L = length of pipe
X = coefficient of expansion
T = total temperature in degrees F
E = expansion

then $$E = L \times X \times T$$

Approximate figures for X are

200° F	0·000006
400	0·0000066
600	0·000007
800	0·0000073
900	0·0000075

The above figures are the increases in length, per unit length, per degree F rise in temperature.

EXAMPLE.—Assume a length of pipe, 600 ft, working at a

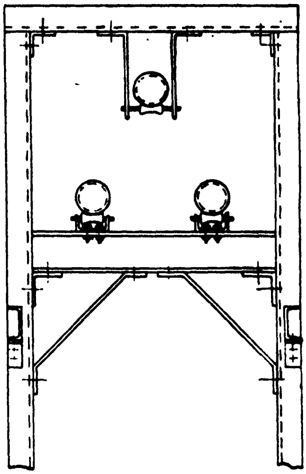


FIG. 59. BRIDGE METHOD OF PIPE SUPPORT

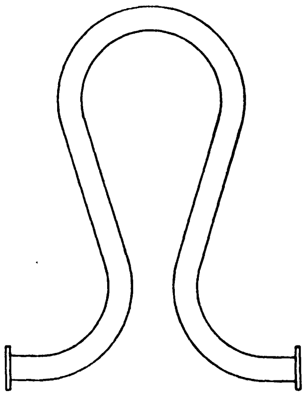


FIG. 60. PLAIN EXPANSION BEND

pressure of 150 lb per sq in. saturated steam, the temperature of which is 366° F.

The constant for 400 is the nearest, so that

$$E = 600 \times 0{\cdot}0000066 \times 366$$
$$= 1{\cdot}449 \text{ ft}$$

which is the *approximate* amount the pipe will lengthen from cold to the working temperature.

It is necessary for this expansion to be taken up either by suitable arrangement of piping, or by the provision of suitable expansion bends. The disposition and dimensions of these bends are matters for the designer who, with knowledge of the amount of movement to be provided for, must determine the numbers and sizes required, so that the stresses in the pipe system are within those allowable for the working temperature. Both plain bends and the more flexible corrugated bend, made

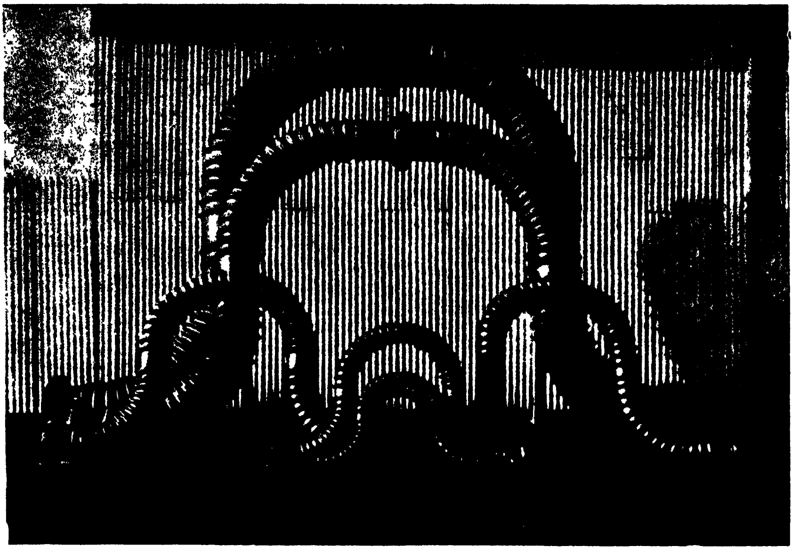


FIG. 61. CORRUGATED EXPANSION BEND
(*Aiton & Co., Derby*)

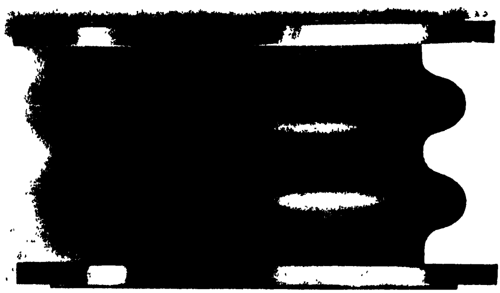


FIG. 62. COPPER EXPANSION PIECE SHOWING OVERLAP OF COPPER AT FLANGES
(*Aiton & Co., Derby*)

by Messrs. Aiton & Co., Ltd., of Derby, are in general use.

Fig. 60 shows an example of the plain bend, while Fig. 61 shows the corrugated type.

For low pressures and temperatures the expansion can be

FIG. 63. STEEL BELLOWS EXPANSION PIECES
(*Aiton & Co., Derby*)

provided by the use of copper expansion pieces, although they are somewhat limited in use, due to the very small movement obtainable per joint. They may be applied for pressures not exceeding 30 lb per sq in., and are made in bores ranging from 4 in. to 42 in.

Fig. 62 shows a typical example, and a point to be noticed is that the copper is turned over the face of the steel flanges,

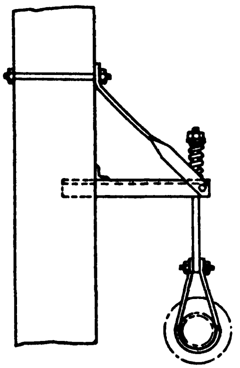


FIG. 64. EXAMPLES OF SLING SUSPENSIONS
(*Aiton & Co., Derby*)

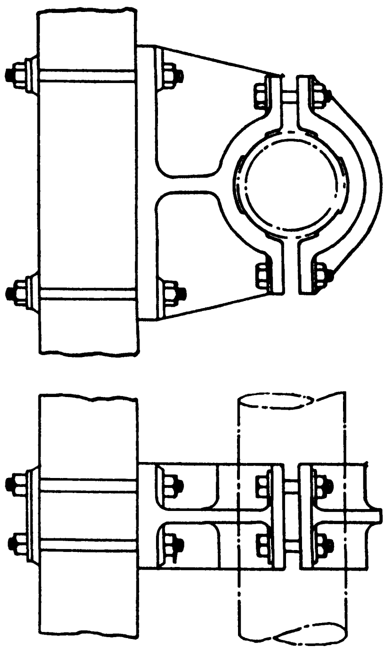


FIG. 65. PIPE ANCHOR
(*Aiton & Co., Derby*)

so that the joint is actually made on the face of the copper.

A further interpretation is the steel bellows type, similar to that shown in Fig. 63. There are different types of bellows available, and designs can be put forward to suit specific temperatures and pressures. It is recommended that not more than six bellows should be used in one expansion piece, and the movement per bellows should not exceed plus and minus $\frac{1}{4}$ in. for the 3 in. bore, to plus and minus $\frac{3}{8}$ in. for the 60 in. bore.

Considerable end thrusts result from the use of this type of expansion piece, and suitable anchors must be placed between each bellows joint in order that the movement may be controlled. The magnitude of the resulting thrusts makes it necessary for the building steel work to be designed to take the large loads.

The necessity for providing freedom of movement in pipework, while at the same time ensuring adequate support, has already been mentioned. To this end, the sling type of support suspension is now almost universally used, a common and effective type being given in Fig. 64. At predetermined positions on the pipe ranges the pipe must be anchored. Different types of anchor are necessary to suit special conditions and adequate supports and expansion bends, as already described, must be provided between the anchors, to keep the stresses within the permissible limits. A typical arrangement is shown in Fig. 65.

Flanges

The general increase in working pressures in present-day practice has provided additional problems for the designer to solve, particularly in connection with the design of flanges and joints. For steam pressures up to 450 lb per sq in., and temperatures up to 750° F, flanges screwed and expanded are now used, while the same construction is permissible for water pressures up to 600 lb per sq in., and temperatures. to 350° F.

Welded flanges are used for higher pressures and temperatures.

The normal joints used for flanges to British Standard Tables *A* to *E* inclusive are the corrugated brass type, while the nickel corrugated ring is utilized for tables *F* to *K* inclusive.

For steam pressures over 450 lb per sq in. and temperatures over 750° F the serrated ring joint as shown in Fig. 66 has been adopted. For this joint the flanges are welded solid to the pipe, the minimum proportions of the weld being to the standards set out in B.S. 806. The face of the flanges is carefully machined, the joint being formed by a steel serrated ring between the flanges, with compressed asbestos packing on either side of the ring.

"Gramophone Finish"

An alternative to this joint is shown in Fig. 67, which shows a further interpretation, in which the faces of the flanges are machined to a fine "gramophone finish" by a small-nosed tool, leaving indentations of $\frac{1}{64}$ in. on the face of the flange, at pitches up to 36 indentations to the inch. The tops of these

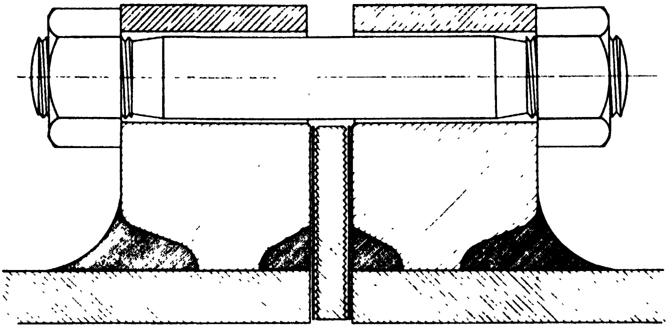


FIG. 66. SERRATED RING JOINT WITH WELDED FLANGES
(*Aiton & Co., Derby*)

indentations are lapped to a face plate, and the joint is formed by a compressed asbestos joint ring placed between the two faces.

The "Corwel" Joint

A joint in common use for pressures over 250 lb per sq in. for all steam temperatures is the Aiton "Corwel" joint, shown

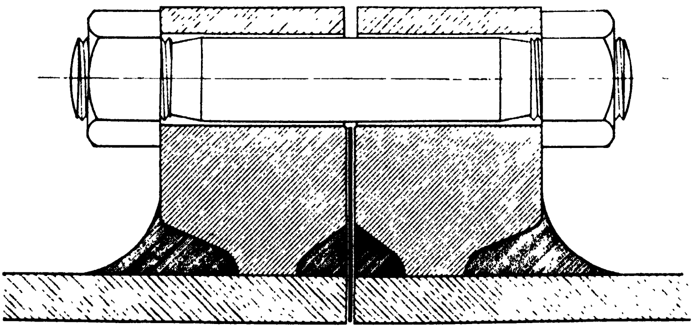


FIG. 67. "GRAMOPHONE FINISH" JOINT
(*Aiton & Co., Derby*)

in Fig. 68. The chief feature of the Corwel construction is the corrugation formed on the end of the pipe. This is clearly shown in the figure. The folded back portion of the pipe is faced off so as to form a true machine face; the outer extremity is prepared by machining, and welded round its circumference. The flanges are of the loose type, and the diameters for bolting are to the appropriate British Standard Table.

The thickness of the loose ring is fixed by the makers, to suit the working conditions.

Drainage

In the early part of this chapter mention is made of the part played by the supports in the avoidance of sag and the consequent formation of water pockets. The evils of such

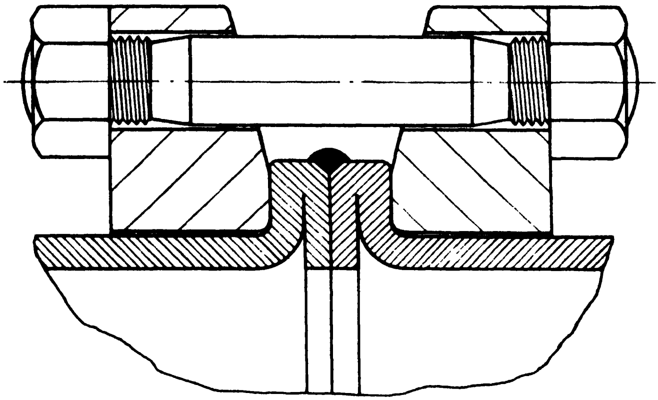


FIG. 68. "CORWEL" JOINT
(*Aiton & Co., Derby*)

formations cannot be overestimated. One has only to consider for a moment the kinetic energy of a slug of water travelling along a pipe with the velocity of steam (which may quite easily attain 70 to 100 miles per hour) to realize the tremendous stresses that can be set up by a sudden stoppage or change of direction. The repercussions of the pressure thus developed may well result in the partial destruction of the pipe line, or, at the very least, severe damage to the valves and connected apparatus is almost certain.

The most careful work in the installing stage will not, however, compensate for inadequate provision for drainage, and the selection and location of steam traps are matters of vital importance.

The pipe should normally be given a slope of at least 1 in. in 30 ft in the direction of the steam flow, and for a short straight pipe it should be theoretically possible to get rid of the water by providing a single trap at the lowest point of the run. The capacity of such a trap is easily assessed.

Heat Losses

When steam is turned into a cold system, heat is given up by the steam in raising the material of the system to the working temperature. This heat quantity depends upon the weight of the material whose temperature is so raised, and upon its specific heat. It is also convenient to know the time taken to produce a given temperature difference, so that the problem may be stated in terms of B.Th.U. per hour. But immediately the temperature of the pipe surface is raised above that of the surrounding atmosphere radiation and convection commence, giving rise to a further heat loss. This heat loss will be some function of the difference between the *average* surface temperature and the temperature of the surroundings. In considering radiation and convection, the shape, position and nature of the surface must also be taken into account.

Once the working temperature is reached, conditions remain more or less constant, with a continuous loss taking place from the surface, a loss which will be a function of the working temperature and the surroundings. This will naturally be somewhat higher than the surface loss during the heating-up period, because the average temperature difference between the surface and the surroundings will be greater.

For the sake of simplicity, take the case of a 4 in. bore pipe, 150 ft long. In its length are ten pairs of flanges, and when completed it is lagged to a thickness of 2 in. with magnesia lagging. The steam is carried at a pressure of 20 lb per sq in. saturated. The system is installed outdoors but in a reasonably sheltered position, and it is known that even under the worst winter conditions the temperature, owing to surrounding buildings, does not drop below 40° F. It is always desirable to calculate for the known worst condition, which in this case would be starting the system up from cold in winter. The pipe weighs 9·8 lb per foot run, and the flanges 20 lb per pair. The total weight of metal to be considered is therefore 1670 lb.

The specific heat is 0·114. Since the temperature of steam at 20 lb per sq in. is 259° F, the temperature rise to the working temperature is 219°. Then the heat given up by the steam in bringing the system up to this point is

$$1670 \times 219 \times 0{\cdot}114 = 41{,}600 \text{ B.Th.U.}$$

If we assume that the working temperature is attained in 15 minutes, the heat given up to the pipe is seen to be absorbed at the rate of 166,400 B.Th.U. per hour.

Convection and Radiation

It is now necessary to consider Convection and Radiation in some detail.

For natural convection the empirical formula is

$$H = C(t_1 - t_2)^{1\cdot 25} \text{ B.Th.U. per hour,}$$

where H = heat dispersed by convection per sq ft per hour
t_1 = temperature of the heated surface
t_2 = temperature of the surrounding air
C = an experimental constant depending upon the position and shape of the surface.

Values of C for various types of heated body are given in page 107, which also shows various constants for use when calculating radiation.

It has been assumed that the system under discussion is covered with a plastic lagging to a thickness of 2 in., and the overall diameter will therefore be 8 in. For purposes of the calculation, the pipe may be treated as a large cylinder, and the constant C has a value of 0·35.

The surface temperature with efficient lagging of the type mentioned will be in the neighbourhood of 90° F. During the warming-up period only the average temperature is taken, because the system is assumed to be started up from cold (40° F). The figure to be used is $\frac{90 + 40}{2} = 65°$ F.

Then
$$\begin{aligned} H &= 0\cdot 35(65 - 40)^{1\cdot 25} \\ &= 0\cdot 35 \times 51 \\ &= 17\cdot 9 \text{ B.Th.U. per sq ft per hour.} \end{aligned}$$

But with the lagging, the total surface of the system is 300 sq ft, so that the loss by convection during the warming-up period is at the rate of 5400 B.Th.U. per hour, for 15 minutes only.

For radiation, a simple application of the Stefan-Boltzman law is used, the constants for the emissivity figure being taken from Table IV. The law states that when a body of given area has an emissivity E, and a temperature of T_1 degrees absolute, with the surrounding at T_2 degrees absolute, the heat radiated in B.Th.U. per hour is given by

$$R = 17\cdot 3 \times 10^{-10} \times EA(T_1^4 - T_2^4).$$

For greater convenience, this may be transposed to

$$R = 0{\cdot}173 \times EA\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \text{ B.Th.U. per hour.}$$

Some explanation may be necessary here as to the term Absolute Temperature.

TABLE IV

EMISSIVITY FIGURES FOR VARIOUS SURFACES

Material	Surface Condition	Emissivity (Black body = 1)
Brass . . .	Brightly polished	0·057
	Rubbed with emery	0·208
Copper . . .	Polished	0·041
	Scratched	0·094
	Oxidized black	0·788
Aluminium . .	Polished	0·053
	Rough	0·072
Lead	Grey oxidized	0·284
Iron	Polished, nickel plated	0·059
	Mat nickel plated	0·114
	Fresh tinned	0·082
	Fresh zinced	0·230
	Grey zinced	0·280
	Freshly rubbed with emery	0·245
	Red rusted	0·694
	Wrought	0·664
	Coarse oxidized	0·817
	Cast	0·817
Varnished enamel .	Snow white	0·919
Aluminium . .	Varnished	0·401
Paper . . .	—	0·940
Plaster of Paris . .	½ mm thick	0·915
Oak-wood . . .	Planed	0·941
Brick	Red	0·941
Porcelain . . .	Glazed	0·936
Glass	Polished	0·950
Marble	Light grey polish	0·943
Aluminium paint .	On rough sheet iron	0·38–0·7 (Average 0·55)
Metallic paint . .	All colours	0·9–0·95
Furnace interior . .	Closed chambers at uniform temperatures	
	All surfaces	1·0
Refractory brick .	Fire brick	0·75–0·8

CONVECTION CONSTANTS C

	Value of C
Plane horizontal surface facing upward and hotter than surrounding air, or facing down, and colder than surrounding air (or gas)	0·39
For large vertical surfaces	0·3
For horizontal surfaces facing down and hotter than surroundings.	0·2
For large cylinders, or pipes over 6 in. diameter. .	0·35
Pipes: 6 in. diameter	0·37
5 in. ,, ...	0·375
4 in. ,, ...	0·399
3 in. ,, ...	0·435
2 in. ,, ...	0·45
1½ in. ,, ...	0·53
1 in. ,, ...	0·66
½ in. ,, ...	1·05

When a gas is heated, it expands $\frac{1}{493}$ of its volume for each degree F temperature rise, and it contracts a similar amount when cooled. Thus, if we continue to abstract more heat there will theoretically come a time when it can contract no more, it has no volume. This must be when we have reduced its temperature by 492° F, and this point is called the Absolute Temperature, which is $492 - 32 = 460°$ F. So when we speak of the absolute temperature of a body, we must add 460 to the thermometer reading.

In the example, the emissivity figure for a painted surface is seen from the Tables to be 0·95.

During the warming-up period the *average* temperature of the surface has been given as 65° F, so that the absolute temperature is $65 + 460 = 525$. Similarly, the absolute temperature of the surroundings is 500° F.

If we require the result in B.Th.U. per sq ft per hour the area A may be omitted from the formula.

We get then,

$$R = 0{\cdot}173 \times 0{\cdot}95 \left[\left(\frac{525}{100}\right)^4 - \left(\frac{500}{100}\right)^4\right] \text{ B.Th.U. per sq ft per hour}$$

$$= 0{\cdot}164 \times (5{\cdot}25^4 - 5{\cdot}00^4)$$
$$= 0{\cdot}164 \times (775 - 625)$$
$$= 24{\cdot}6 \text{ B.Th.U. per sq ft per hour.}$$

But the total surface is 300 sq ft, so that the heat lost by radiation during this period is at the rate of 7400 B.Th.U. per hour.

Trapping and Sizing

We are now is a position to assess the maximum capacity of the trap required, by adding the figures so far obtained, and computing the weight of water condensed from them.

B.Th.U. required to raise the temperature of the pipes: 41,600 B.Th.U. in 15 minutes; so that the hourly rate is 166,400 B.Th.U.

Heat given up by convection at the rate of 5400 B.Th.U.
Heat given up by radiation at the rate of 7400 B.Th.U.
Total 179,500 B.Th.U. per hour.

The steam pressure in the pipes is given as 20 lb per sq in. saturated, and as only the latent heat is given up, 939 B.Th.U. per lb of steam will be taken up, and the weight of condensate will therefore be

$$\frac{179{,}500}{939} = 192 \text{ lb per hour.}$$

In order to drain the system effectively the trap must be able to deal with this amount of condensate, but during normal working the quantities will be much less.

Using the methods outlined above, but calculating from the working temperatures instead of the averages from cold, we get,

For convection,

$$\begin{aligned} H &= 0{\cdot}35(90 - 40)^{1{\cdot}25} \\ &= 0{\cdot}35 \times 135 \\ &= 47{\cdot}2 \text{ B.Th.U. per sq ft per hour,} \end{aligned}$$

a total for the 300 sq ft of 14,200 B.Th.U. per hour.

For radiation,

$$\begin{aligned} R &= 0{\cdot}173 \times 0{\cdot}95(5{\cdot}5^4 - 5{\cdot}0^4) \\ &= 0{\cdot}164 \times (920 - 625) \\ &= 48{\cdot}5 \text{ B.Th.U. per sq ft per hour,} \end{aligned}$$

a total for the whole surface of 14,550 B.Th.U. per hour,

Then during working, the heat loss for both radiation and convection is 28,750 B.Th.U. per hour, and the weight of water condensed will be $\frac{28{,}750}{939} = \underline{30{\cdot}6 \text{ lb}}$.

This is the amount which the trap must pass during normal working, so that one characteristic of the trap chosen must be the ability to deal with about six times its normal working weight of condensate.

It is important to remember, too, that the back pressure against which the trap may have to operate has a great influence on the capacity, and that both this back pressure and the inlet pressure must be taken into account when the trap is selected. Further details of these important factors are given in the chapter on trapping.

It is perhaps not too much to say that by far the majority of troubles which are experienced with steam traps are not the fault of the traps themselves, but are far more often due to either wrong selection or incorrect installation. No trap, however excellent its design and construction, can be expected to operate properly unless steps are taken to see that the condensate to be removed has full and free access to the trap. It is absolutely useless to tap a small drop leg into a

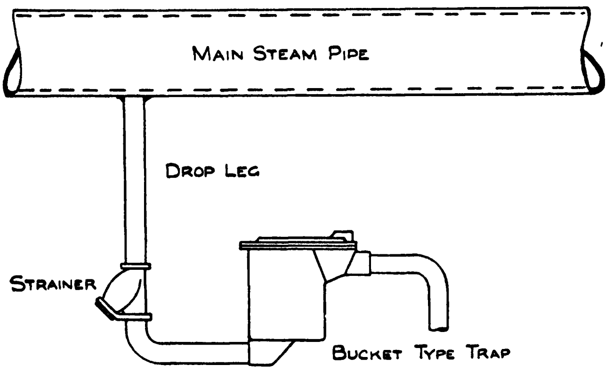


FIG. 69. WRONG METHOD OF PIPE DRAINAGE

large diameter steam main as shown in Fig. 69, and expect the trap to do its work. Even if the inside end of the small pipe is finished off flush, it is quite obvious that only a small proportion of the water moving along the pipe can possibly find egress through so small an orifice.

The correct method is shown in Fig. 70. A full bore tee piece is inserted in the pipe at the drainage point, and the drop leg to the trap fitted in the vertical branch as shown. Where the quantities to be removed are considerable, the full bore of the pipe should be carried on below the vertical branch of the tee, to form a collecting chamber sufficiently large to hold the whole of the condensate to be removed from that particular section of the pipe, with none remaining in the horizontal portion at all. The calculation given above is useful for assessing the size of this collecting chamber. The ends of pipe systems should, except for the very smallest diameters, be dealt with in the same way, and a full bore bend fitted. A very common practice, especially with screwed pipe, is to provide a reducer, frequently of the concentric type, into which the drop leg to the trap is fitted. The impossibility of effectively draining a pipe by such a method is obvious.

For the smallest systems where cost is a primary consideration, an eccentric reducer may be used, but even then it is

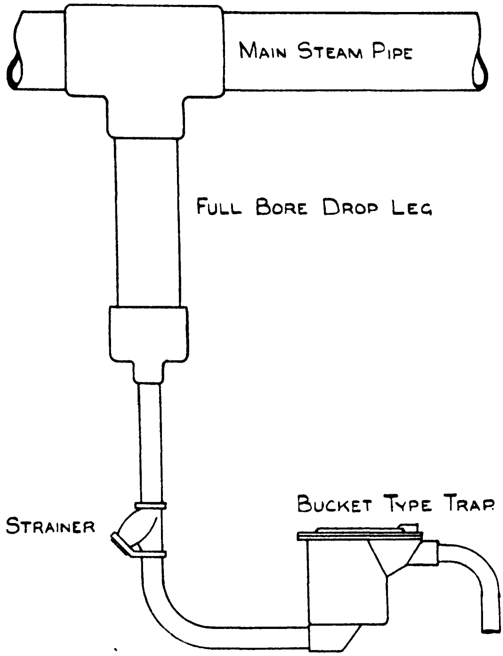


FIG. 70. CORRECT METHOD OF PIPE DRAINAGE

questionable whether the difference in cost between a reducer and a full bore bend makes it worth while to risk trouble in service by installing what can only be regarded as a compromise.

Of the greatest importance, too, is the siting of drainage points in relation to valves. Considerable amounts of condensate may accumulate in the bodies of large valves, and the makers usually provide bosses which may be drilled to receive drains. But even with small valves it is necessary to make adequate arrangements for the dispersal of the water which may accumulate during the periods when the particular valve is closed, and the rest of the system working. It is obvious that under such conditions the closed valve becomes a terminal end in which water in dangerous amounts can collect. In all such cases a tee piece should be installed immediately before the valve, in the manner described above.

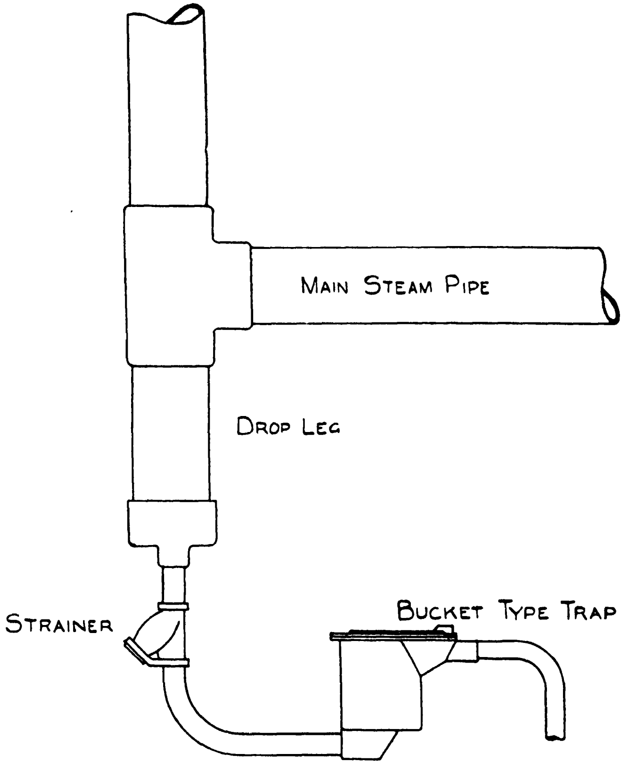


FIG. 71. CORRECT METHOD OF DRAINING VERTICAL RISER

Vertical risers should be fitted with a tee piece instead of a bend at the bottom, as shown in Fig. 71, so that the trap will drain both the riser and the horizontal pipe immediately before it.

Summarizing, it may be said that the principles to be followed when designing a successful, trouble-free pipe system are—

1. Provide such adequate support throughout the length of the pipes that there is no possibility of sagging, the

number and position of the supports to be dictated in the last resort by the characteristics of the system and not by existing building features.

2. Ensure that such arrangements are made for expansion that the stresses developed during working are within the allowable stresses for the working temperatures.

3. Control the direction of expansion during working by the provision of suitable anchorages.

4. Ensure that the condensate has ready access to such traps as may be provided.

5. Provide traps of adequate capacity and suitable design at all points where condensate can collect either while working normally or during starting up or shutting down periods.

6. When in doubt, *DRAIN*!

Pipe Sizing

In order that the required amount of steam shall be carried to the point of utilization, all the factors influencing pipe sizing must be carefully considered. Although the necessity for this would appear to be quite obvious, it is unfortunately a fact that, in a growing factory, the sizes are far too often chosen from the pipe stock the engineer happens to have on hand at the time, and the necessity of getting the job done quickly is a far more potent influence than any considered appreciation of the real requirement. The effect of these constant improvisations is cumulative and in the end costly, for the losses, inseparable from haphazard procedure, are continuous from the time the installation is put to work.

The factors involved in correct pipe sizing are—

(*a*) Steam pressure at inlet.

(*b*) Permissible pressure drop along the pipe.

(*c*) Weight of steam to be carried.

(*d*) Character of the pipe run, that is to say, bends, valves, tees, and other fittings causing obstructions to the normal flow.

(*e*) Steam state at inlet, saturated or superheated.

Of all these factors, only the permissible pressure drop along the pipe may be under the direct control of the designer. It is important, not only for this reason but also because the final selection has the greatest influence upon capital cost.

Sometimes the requirements of the apparatus with regard to steam pressure may be so related to the inlet pressure that the drop along the pipe is already fixed, in which case the selection of the pipe size simply involves the relating of all the factors in the appropriate formula.

A most reliable and useful formula, and one which at the same time is very simple to use, is the well-known "Box formula," which is

$$d = \sqrt[5]{\frac{C^2 \times L}{H}} \div 3{\cdot}7$$

where d = diameter of the pipe in inches
C = cubic feet of steam per minute
H = pressure drop along the pipe in inches of water (27 in. of water = 1 lb per sq in.)
L = length of the pipe in yards.

The length L may be designated the "flow length," that is to say, the length of the pipe as it influences the calculation, with all allowances for bends, valves, etc., added in. It may exceed the actual measured length by many feet.

General allowances to be made are—

Change of direction due to elbow . .	add	50 ft.
Change of direction due to bend . .	,,	20 ,,
Right-angle tee	,,	100 ,,

The resistance due to valves varies enormously, and is very difficult to assess, but in general it may be taken that a fully open full way valve offers a resistance equivalent to about 15 to 20 ft of pipe, and a fully open globe valve from 80 to 120 ft. A partially closed valve of any type offers a very large resistance indeed, so that it is always better to err on the high side, to allow for such contingencies. It will be understood that since the pressure drop from any cause is some function of the velocity, no single figure can serve for all circumstances, but the figures given are at least "safe" for a fairly wide range of pipe sizes. Actually they are about right for 4 in. to 6 in. diameters, a little low for higher sizes, and more than ample for the lower.

The Box formula given is very easily transposed, to give any required element if the others are known. Thus, where the pressure drop of an existing system is required,

$$H = \frac{C^2 \times L}{(3{\cdot}7d)^5}$$

or for steam quantity,

$$C = \sqrt{\frac{(3{\cdot}7d)^5 \times H}{L}}$$

If the length of pipe, to give a predetermined pressure drop with given diameter and load, is wanted,

then $$L = \frac{(3{\cdot}7d)^5 \times H}{C^2}$$

EXAMPLE.—Admission pressure, 120 lb per sq in. saturated. "flow length," that is to say, including all allowances, 140 yd. Steam load, 1200 lb per hour. Pressure drop along pipe, 2 lb per sq in. Find required diameter.

First, from the steam tables, find the specific volume of steam at the inlet pressure.

This is 3·33 cu ft per lb.

Since the weight of steam required is 20 lb per minute, we get 66·6 cu ft per minute.

The pressure drop is 2 lb per sq in. = 54 in. of water.

Then

$$d = \sqrt[5]{\frac{(66{\cdot}6)^2 \times 140}{54}} \div 3{\cdot}7$$

$$= \sqrt[5]{\frac{4440 \times 140}{54}} \div 3{\cdot}7$$

$= 5$ in. diameter to the nearest standard size.

This is assuming that the pressure drop is fixed by the exigencies of the process at the figure given, but if it may be supposed that the apparatus which is served would be operated at a pressure of, say, 100 lb per sq in., a 20 lb pressure drop could be permitted along the pipe. In that case, the factor H becomes 540, and the needs could be served with a pipe of 3½ in. diameter.

Thus, where permitted, the judicious selection of the pressure drop may be the means of saving considerable capital expenditure. Although the reduction of pipe size may at the time of installation seem desirable from consideration of cost, the principle must be used with the utmost discretion, and cannot be pushed too far. Possible future demands upon the particular service must be visualized as far as possible, and reasonable allowance made. A pipe which is adequate at the time of installation, but in which the permissible pressure drop has been pushed to the limit, immediately becomes too small at the slightest increase of load, and may require replacing with a larger size far too soon. The capital expenditure will in that case be far greater than would have been necessary if larger runs had been installed in the first place.

TABLE A

Particulars of Wrought Steel Pipes

Suitable for working gas pressures up to 30 lb per sq in. and water pressures up to 50 lb per sq in. We recommend this table for steam pressures up to 15 lb per sq in.

Nominal Bore	Actual Outside Diameter	Diameter of Flange	Diameter of P.C.	Number of Bolts	Diameter of Bolts	Thickness of Flange
in.	in.	in.	in.		in.	in.
1/2	27/32	3 3/4	2 5/8	4	1/2	3/16
3/4	1 1/16	4	2 7/8	4	1/2	3/16
1	1 11/32	4 1/2	3 1/4	4	1/2	3/16
1 1/4	1 11/16	4 3/4	3 7/16	4	1/2	1/4
1 1/2	1 29/32	5 1/4	3 7/8	4	1/2	1/4
2	2 3/8	6	4 1/2	4	5/8	5/16
2 1/2	3	6 1/2	5	4	5/8	5/16
3	3 1/2	7 1/4	5 3/4	4	5/8	3/8
3 1/2	4	8	6 1/2	4	5/8	3/8
4	4 1/2	8 1/2	7	4	5/8	3/8
4 1/2	5	9	7 1/2	4	5/8	7/16
5	5 1/2	10	8 1/4	4	5/8	1/2
6	6 1/2	11	9 1/4	4	5/8	1/2
7	7 1/2	12	10 1/4	8	5/8	1/2
8	8 1/2	13 1/4	11 1/2	8	5/8	1/2
9	9 1/2	14 1/2	12 3/4	8	5/8	5/8
10	10 1/2	16	14	8	3/4	5/8
11	11 1/2	17	15	8	3/4	5/8
12	12 1/2	18	16	8	3/4	5/8
13		19 1/4	17 1/4	8	3/4	5/8
14	—	20 3/4	18 1/2	8	7/8	5/8
15	—	21 3/4	19 1/2	8	7/8	5/8
16	—	22 3/4	20 1/2	12	7/8	5/8
17	—	24	21 3/4	12	7/8	5/8
18	—	25 1/4	23	12	7/8	5/8
19	—	26 1/2	24	12	7/8	5/8
20		27 3/4	25 1/4	12	7/8	5/8
21	—	29	26 1/2	12	7/8	5/8
22	—	30	27 1/2	12	1	5/8
23	—	31	28 1/2	12	1	5/8
24	—	32 1/2	29 3/4	12	1	5/8

TABLE A—(*continued*)

Nominal Bore	Actual Outside Diameter	Diameter of Flange	Diameter of P.C.	Number of Bolts	Diameter of Bolts	Thickness of Flange
in.	in.	in.	in.		in.	in.
25	—	32½	29¾	16	1	⅝
26	—	33¼	30¾	16	1	⅝
27	—	34¼	31¾	16	1	⅝
28	—	35¼	32¾	20	1	⅝
29	—	36¼	33¾	20	1	⅝
30	—	37¼	34¾	20	1	⅝
31	—	38¼	35¾	20	1	⅝
32	—	39½	37	20	1	⅝
33	—	40½	38	20	1	⅝
34	—	41½	39	20	1	⅝
35	—	42½	40	24	1	⅝
36	—	43½	41	24	1	¾
37	—	44½	42	24	1	¾
38	—	45½	43	24	1	¾
39	—	46½	44	24	1	¾
40	—	47½	45	24	1	¾
41	—	48½	46	28	1	¾
42	—	49½	47	28	1	¾
43	—	50½	48	28	1	¾
44	—	51¾	49¼	28	1	¾
45	—	52¾	50¼	28	1	¾
46	—	53¾	51¼	28	1	¾
47	—	54¾	52¼	28	1	¾
48	—	55¾	53¼	28	1	¾
54	—	62½	59¾	28	1⅛	1
60	—	69	66¼	32	1⅛	1⅛
66	—	75	72¼	32	1⅛	1¼
72	—	81	78¼	36	1⅛	1¼

(*Courtesy Messrs. Aiton & Co. Ltd., Derby*)

TABLE B

Particulars of Cast-iron Pipes

Suitable for working water pressures above 50 lb and up to 130 lb per sq in.

Nomina Bore	Metal	"A" Length of Elbow or Branch	"B" Radius of Elbow or Branch	"C" Length of Bend	"D" Radius of Bend	"E" Base of Foot	"F" Length of Foot	Diameter of Flange	Thickness of Flange	Number of Bolts	Diameter of Bolts	Diameter of P.C.
in.	in.	in.	in.	in.	in.	in.	in.	in.	in.		in.	in.
½	—	3½	—	—	—	—	—	3¾	½	4	½	2⅝
¾	—	3¾	—	—	—	—	—	4	½	4	½	2⅞
1	—	4	—	—	—	—	—	4¼	½	4	½	3¼
1¼	—	4¼	—	—	—	—	—	4¾	⅝	4	½	3 7/16
1½	—	4½	—	—	—	—	—	5¼	⅝	4	½	3⅞
2	—	5	3¼	10	9	2½	3½	6	11/16	4	⅝	4½
2½	—	5½	3¾	11	10	3	3¾	6½	11/16	4	⅝	5
3	0·38	6	4	12	11	3½	4	7¼	¾	4	⅝	5¾
3½	—	6½	4½	13	12	4	4½	8	¾	4	⅝	6½
4	0·39	7	4¾	14	12½	4½	4½	8½	¾	4	⅝	7
4½	—	7½	5¼	15	13½	5	5	9	¾	8	⅝	7½
5	0·41	8	5½	16	14½	6	5½	10	13/16	8	⅝	8¼
6	0·43	9	6½	18	16½	6½	6	11	13/16	8	⅝	9¼
7	0·45	10	7¼	20	18	7	6½	12	⅞	8	⅝	10¼
8	0·47	11	8¼	22	20	8	7	13¼	⅞	8	⅝	11⅜
9	0·49	12	9	24	22	9	8	14½	⅞	8	⅝	12¾
10	0·52	13	10	26	24	10	8½	16	1	8	¾	14
11	—	14	11	28	26	11	9	17	1	8	¾	15
12	0·57	15	11¾	30	28	12	9½	18	1	12	¾	16
13	—	16	12½	32	30	13	10	19¼	1	12	¾	17¼
14	0·61	17	13½	34	31½	14	11	20¾	1⅛	12	⅞	18⅜
15	0·63	18	14¼	36	33½	15	11½	21¾	1⅛	12	⅞	19½
16	0·65	19	15¼	38	35½	16	12	22¾	1⅛	12	⅞	20⅝
17	—	20	16	40	37½	17	12½	24	1⅛	12	⅞	21¾
18	0·69	21	17	42	39½	18	13	25¼	1¼	12	⅞	23
19	—	22	17¾	44	41½	19	14	26½	1¼	12	⅞	24
20	0·73	23	18¾	46	43½	20	14½	27¾	1¼	16	⅞	25¼
21	0·75	24	19½	48	45½	21	15	29	1⅜	16	⅞	26½
22	0·77	25	20½	50	47½	22	15½	30	1⅜	16	1	27½
23	—	26	21¼	52	49½	23	16	31	1⅜	16	1	28½
24	0·80	27	22¼	54	51½	24	17	32½	1⅜	16	1	29¾
25	—	28	23	56	53	25	17½	33½	1⅜	16	1	31
26	0·83	29	24	58	55	26	18	34½	1½	20	1	32
27	0·85	30	25	60	57	27	18½	35¾	1½	20	1	33¼
28	0·86	31	25¾	62	59	28	19	37¼	1½	20	1⅛	34½
29	—	32	26½	64	61	29	19½	38¼	1½	20	1⅛	35½
30	0·89	33	27½	66	63	30	20	39¼	1⅝	20	1⅛	36½
31	—	34	28½	68	65	31	20½	40¼	1⅝	20	1⅛	37½
32	0·92	35	29¼	70	67	32	21	41¾	1⅝	20	1¼	38¾
33	0·94	36	30	72	69	33	21½	43	1⅝	20	1¼	40
34	—	37	31	74	71	34	22	44	1¾	20	1¼	41
35	—	38	32	76	73	35	22½	45	1¾	24	1¼	42
36	0·98	39	33	78	75	36	23	46¼	1¾	24	1¼	43
37	—	40	34	80	77	39	24	47¼	1¾	24	1¼	44
38	1·01	41	35	82	78	41	25	48¼	1¾	24	1¼	45
39	—	42	36	84	80	42	25½	49½	1¾	24	1¼	46¼
40	1·03	43	37	86	82	43	26	50½	1¾	24	1¼	47¼
41	—	44	38	88	84	44	26½	51½	1⅞	28	1¼	48¼
42	1·06	45	39	90	86	45	27	52½	1⅞	28	1¼	49¼
44	1·08	47	41	94	90	47	28	54¾	1⅞	28	1¼	51½
46	1·11	49	43	98	94	49	29	56¾	2	32	1¼	53½
48	1·13	51	45	102	98	51	30	58¾	2	32	1¼	55½

(Courtesy Messrs. Aiton & Co. Ltd., Derby)

TABLE D

Particulars of Wrought Steel Pipes

Suitable for working steam pressures up to 50 lb per sq in.

Nominal Bore	Actual Outside Diameter	Diameter of Flange	Diameter of P.C.	Number of Bolts	Diameter of Bolts	Thickness of Flange
in.	in.	in.	in.		in.	in.
½	27/32	3¾	2⅝	4	½	3/16
¾	1 1/16	4	2⅞	4	½	3/16
1	1 11/32	4½	3¼	4	½	3/16
1¼	1 11/16	4¾	3 7/16	4	½	¼
1½	1 29/32	5¼	3⅞	4	½	¼
2	2⅜	6	4½	4	⅝	5/16
2½	3	6½	5	4	⅝	5/16
3	3½	7¼	5¾	4	⅝	⅜
3½	4	8	6½	4	⅝	⅜
4	4½	8½	7	4	⅝	⅜
4½	5	9	7½	8	⅝	7/16
5	5½	10	8¼	8	⅝	½
6	6½	11	9¼	8	⅝	½
7	7½	12	10¼	8	⅝	½
8	8½	13¼	11½	8	⅝	½
9	9½	14½	12¾	8	⅝	⅝
10	10½	16	14	8	¾	⅝
11	11½	17	15	8	¾	⅝
12	12½	18	16	12	¾	⅝
13	14	19¼	17¼	12	¾	¾
14	15	20¾	18½	12	⅞	¾
15	16	21¾	19½	12	⅞	¾
16	17	22¾	20½	12	⅞	¾
17	18	24	21¾	12	⅞	⅞
18	19	25¼	23	12	⅞	⅞
19	20	26½	24	12	⅞	⅞
20	21	27¾	25¼	16	⅞	1
21	22	29	26½	16	⅞	1
22	23	30	27½	16	1	1
23	24	31	28½	16	1	1⅛
24	25	32½	29¾	16	1	1⅛

(Courtesy Messrs. Aiton & Co. Ltd., Derby)

TABLE E

Particulars of Wrought Steel Pipes

Suitable for working steam pressures above 50 lb and up to 100 lb per sq in.

Nominal Bore	Actual Outside Diameter	Diameter of Flange	Diameter of P.C.	Number of Bolts	Diameter of Bolts	Thickness of Flange
in.	in.	in.	in.		in.	in.
$\frac{1}{2}$	$\frac{27}{32}$	$3\frac{3}{4}$	$2\frac{5}{8}$	4	$\frac{1}{2}$	$\frac{1}{4}$
$\frac{3}{4}$	$1\frac{1}{16}$	4	$2\frac{7}{8}$	4	$\frac{1}{2}$	$\frac{1}{4}$
1	$1\frac{11}{32}$	$4\frac{1}{2}$	$3\frac{1}{4}$	4	$\frac{1}{2}$	$\frac{9}{32}$
$1\frac{1}{4}$	$1\frac{11}{16}$	$4\frac{3}{4}$	$3\frac{7}{16}$	4	$\frac{1}{2}$	$\frac{5}{16}$
$1\frac{1}{2}$	$1\frac{29}{32}$	$5\frac{1}{4}$	$3\frac{7}{8}$	4	$\frac{1}{2}$	$\frac{11}{32}$
2	$2\frac{3}{8}$	6	$4\frac{1}{2}$	4	$\frac{5}{8}$	$\frac{3}{8}$
$2\frac{1}{2}$	3	$6\frac{1}{2}$	5	4	$\frac{5}{8}$	$\frac{13}{32}$
3	$3\frac{1}{2}$	$7\frac{1}{4}$	$5\frac{3}{4}$	4	$\frac{5}{8}$	$\frac{7}{16}$
$3\frac{1}{2}$	4	8	$6\frac{1}{2}$	8	$\frac{5}{8}$	$\frac{15}{32}$
4	$4\frac{1}{2}$	$8\frac{1}{2}$	7	8	$\frac{5}{8}$	$\frac{1}{2}$
$4\frac{1}{2}$	5	9	$7\frac{1}{2}$	8	$\frac{5}{8}$	$\frac{1}{2}$
5	$5\frac{1}{2}$	10	$8\frac{1}{4}$	8	$\frac{5}{8}$	$\frac{9}{16}$
6	$6\frac{1}{2}$	11	$9\frac{1}{4}$	8	$\frac{3}{4}$	$\frac{11}{16}$
7	$7\frac{1}{2}$	12	$10\frac{1}{4}$	8	$\frac{3}{4}$	$\frac{3}{4}$
8	$8\frac{1}{2}$	$13\frac{1}{4}$	$11\frac{1}{2}$	8	$\frac{3}{4}$	$\frac{3}{4}$
9	$9\frac{1}{2}$	$14\frac{1}{2}$	$12\frac{3}{4}$	12	$\frac{3}{4}$	$\frac{13}{16}$
10	$10\frac{1}{2}$	16	14	12	$\frac{3}{4}$	$\frac{7}{8}$
11	$11\frac{1}{2}$	17	15	12	$\frac{3}{4}$	$\frac{15}{16}$
12	$12\frac{1}{2}$	18	16	12	$\frac{7}{8}$	1
13	14	$19\frac{1}{4}$	$17\frac{1}{4}$	12	$\frac{7}{8}$	1
14	15	$20\frac{3}{4}$	$18\frac{1}{2}$	12	$\frac{7}{8}$	1
15	16	$21\frac{3}{4}$	$19\frac{1}{2}$	12	$\frac{7}{8}$	1
16	17	$22\frac{3}{4}$	$20\frac{1}{2}$	12	$\frac{7}{8}$	1
17	18	24	$21\frac{3}{4}$	12	$\frac{7}{8}$	$1\frac{1}{8}$
18	19	$25\frac{1}{4}$	23	16	$\frac{7}{8}$	$1\frac{1}{8}$
19	20	$26\frac{1}{2}$	24	16	$\frac{7}{8}$	$1\frac{1}{4}$
20	21	$27\frac{3}{4}$	$25\frac{1}{4}$	16	$\frac{7}{8}$	$1\frac{1}{4}$
21	22	29	$26\frac{1}{2}$	16	1	$1\frac{3}{8}$
22	23	30	$27\frac{1}{2}$	16	1	$1\frac{3}{8}$
23	24	31	$28\frac{1}{2}$	16	1	$1\frac{3}{8}$
24	25	$32\frac{1}{2}$	$29\frac{3}{4}$	16	1	$1\frac{1}{2}$

(Courtesy Messrs. Aiton & Co. Ltd., Derby)

TABLE F

Particulars of Wrought Steel Pipes

Suitable for working steam pressures above 100 lb and up to 150 lb per sq in.

Nominal Bore	Actual Outside Diameter	Diameter of Flange	Diameter of P.C.	Number of Bolts	Diameter of Bolts	Thickness of Flange
in.	in.	in.	in.		in.	in.
½	27/32	3¾	2⅝	4	½	⅜
¾	1 1/16	4	2⅞	4	½	⅜
1	1 11/32	4¾	3 7/16	4	⅝	⅜
1¼	1 11/16	5¼	3⅞	4	⅝	7/16
1½	1 29/32	5½	4⅛	4	⅝	½
2	2⅜	6½	5	4	⅝	⅝
2½	3	7¼	5¾	8	⅝	⅝
3	3½	8	6½	8	⅝	⅝
3½	4	8½	7	8	⅝	¾
4	4½	9	7½	8	⅝	¾
4½	5	10	8¼	8	¾	¾
5	5½	11	9¼	8	¾	⅞
6	6½	12	10¼	12	¾	⅞
7	7½	13¼	11½	12	¾	⅞
8	8½	14½	12¾	12	¾	1
9	9½	16	14	12	⅞	1
10	10½	17	15	12	⅞	1
11	11½	18	16	16	⅞	1⅛
12	12½	19¼	17¼	16	⅞	1⅛
13	14	20¾	18½	16	1	1⅛
14	15	21¾	19½	16	1	1¼
15	16	22¾	20½	16	1	1¼
16	17	24	21¾	20	1	1¼
17	18	25¼	23	20	1	1⅜
18	19	26½	24	20	1⅛	1⅜
19	20	27¾	25¼	20	1⅛	1⅜
20	21	29	26½	24	1⅛	1½
21	22	30	27½	24	1⅛	1½
22	23	31	28½	24	1⅛	1½
23	24	32½	29¾	24	1¼	1⅝
24	25	33½	30¾	24	1¼	1⅝

(*Courtesy Messrs. Aiton & Co. Ltd., Derby*)

TABLE II

Particulars of Wrought Steel Pipes

Suitable for working steam pressures above 150 lb and up to 250 lb per sq in.

Nominal Bore	Actual Outside Diameter	Diameter of Flange	Diameter of P.C.	Number of Bolts	Diameter of Bolts	Thickness of Flange
in.	in.	in.	in.		in.	in.
$\frac{1}{2}$	$\frac{27}{32}$	$4\frac{1}{2}$	$3\frac{1}{4}$	4	$\frac{5}{8}$	$\frac{1}{2}$
$\frac{3}{4}$	$1\frac{1}{16}$	$4\frac{1}{2}$	$3\frac{1}{4}$	4	$\frac{5}{8}$	$\frac{1}{2}$
1	$1\frac{11}{32}$	$4\frac{3}{4}$	$3\frac{7}{16}$	4	$\frac{5}{8}$	$\frac{9}{16}$
$1\frac{1}{4}$	$1\frac{11}{16}$	$5\frac{1}{4}$	$3\frac{7}{8}$	4	$\frac{5}{8}$	$\frac{11}{16}$
$1\frac{1}{2}$	$1\frac{29}{32}$	$5\frac{1}{2}$	$4\frac{1}{8}$	4	$\frac{5}{8}$	$\frac{11}{16}$
2	$2\frac{3}{8}$	$6\frac{1}{2}$	5	4	$\frac{5}{8}$	$\frac{3}{4}$
$2\frac{1}{2}$	3	$7\frac{1}{4}$	$5\frac{3}{4}$	8	$\frac{5}{8}$	$\frac{3}{4}$
3	$3\frac{1}{2}$	8	$6\frac{1}{2}$	8	$\frac{5}{8}$	$\frac{7}{8}$
$3\frac{1}{2}$	4	$8\frac{1}{2}$	7	8	$\frac{5}{8}$	$\frac{7}{8}$
4	$4\frac{1}{2}$	9	$7\frac{1}{2}$	8	$\frac{5}{8}$	1
$4\frac{1}{2}$	5	10	$8\frac{1}{4}$	8	$\frac{3}{4}$	1
5	$5\frac{1}{2}$	11	$9\frac{1}{4}$	8	$\frac{3}{4}$	$1\frac{1}{8}$
6	$6\frac{1}{2}$	12	$10\frac{1}{4}$	12	$\frac{3}{4}$	$1\frac{1}{8}$
7	$7\frac{1}{2}$	$13\frac{1}{4}$	$11\frac{1}{2}$	12	$\frac{3}{4}$	$1\frac{1}{4}$
8	$8\frac{1}{2}$	$14\frac{1}{2}$	$12\frac{3}{4}$	12	$\frac{3}{4}$	$1\frac{1}{4}$
9	$9\frac{1}{2}$	16	14	12	$\frac{7}{8}$	$1\frac{3}{8}$
10	$10\frac{1}{2}$	17	15	12	$\frac{7}{8}$	$1\frac{3}{8}$
11	$11\frac{1}{2}$	18	16	16	$\frac{7}{8}$	$1\frac{1}{2}$
12	$12\frac{1}{2}$	$19\frac{1}{4}$	$17\frac{1}{4}$	16	$\frac{7}{8}$	$1\frac{1}{2}$
13	14	$20\frac{3}{4}$	$18\frac{1}{2}$	16	1	$1\frac{5}{8}$
14	15	$21\frac{3}{4}$	$19\frac{1}{2}$	16	1	$1\frac{5}{8}$
15	16	$22\frac{3}{4}$	$20\frac{1}{2}$	16	1	$1\frac{3}{4}$
16	17	24	$21\frac{3}{4}$	20	1	$1\frac{3}{4}$
17	18	$25\frac{1}{4}$	23	20	1	$1\frac{7}{8}$
18	19	$26\frac{1}{2}$	24	20	$1\frac{1}{8}$	$1\frac{7}{8}$
19	20	$27\frac{3}{4}$	$25\frac{1}{4}$	20	$1\frac{1}{8}$	2
20	21	29	$26\frac{1}{2}$	24	$1\frac{1}{8}$	2
21	22	30	$27\frac{1}{2}$	24	$1\frac{1}{8}$	$2\frac{1}{8}$
22	23	31	$28\frac{1}{2}$	24	$1\frac{1}{8}$	$2\frac{1}{8}$
23	24	$32\frac{1}{2}$	$29\frac{3}{4}$	24	$1\frac{1}{4}$	$2\frac{1}{4}$
24	25	$33\frac{1}{2}$	$30\frac{3}{4}$	24	$1\frac{1}{4}$	$2\frac{1}{4}$

(*Courtesy Messrs. Aiton & Co. Ltd., Derby*)

TABLE J

Particulars of Wrought Steel Pipes

Suitable for working steam pressures above 250 lb and up to 350 lb per sq in.

Nominal Bore	Actual Outside Diameter	Diameter of Flange	Diameter of P.C.	Number of Bolts	Diameter of Bolts	Thickness of Flange
in.	in.	in.	in.		in.	in.
½	27/32	4½	3¼	4	⅝	⅝
¾	1 1/16	4½	3¼	4	⅝	⅝
1	1 11/32	4¾	3 7/16	4	⅝	¾
1¼	1 11/16	5¼	3⅞	4	⅝	¾
1½	1 29/32	5½	4⅛	4	⅝	⅞
2	2⅜	6½	5	4	¾	1
2½	3	7¼	5¾	8	¾	1
3	3½	8	6½	8	¾	1¼
3½	4	8½	7	8	¾	1¼
4	4½	9	7½	8	¾	1⅜
4½	5	10	8¼	8	⅞	1⅜
5	5½	11	9¼	8	⅞	1½
6	6½	12	10¼	12	⅞	1½
7	7½	13¼	11½	12	⅞	1⅝
8	8½	14½	12¾	12	⅞	1⅝
9	9½	16	14	12	1	1¾
10	10½	17	15	12	1	1⅞
11	11½	18	16	16	1	1⅞
12	12½	19¼	17¼	16	1	2
13	14	20¾	18½	16	1⅛	2
14	15	21¾	19½	16	1⅛	2⅛
15	16	22¾	20½	16	1⅛	2⅛
16	17	24	21¾	20	1⅛	2¼
17	18	25¼	23	20	1⅛	2⅜
18	19	26½	24	20	1¼	2⅜
19	20	27¾	25¼	20	1¼	2½
20	21	29	26½	24	1¼	2½
21	22	30	27½	24	1¼	2⅝
22	23	31	28½	24	1¼	2⅝
23	24	32½	29¾	24	1⅜	2¾
24	25	33½	30¾	24	1⅜	2¾

(*Courtesy Messrs. Aiton & Co. Ltd., Derby*)

TABLE K

Particulars of Wrought Steel Pipes

Suitable for working steam pressures above 350 lb and up to 450 lb per sq in.

Nominal Bore	Actual Outside Diameter	Diameter of Flange	Diameter of P.C.	Number of Bolts	Diameter of Bolts	Thickness of Flange
in.	in.	in.	in.		in.	in.
$\frac{1}{2}$	$\frac{27}{32}$	$4\frac{1}{2}$	$3\frac{1}{4}$	4	$\frac{5}{8}$	$\frac{3}{4}$
$\frac{3}{4}$	$1\frac{1}{16}$	$4\frac{1}{2}$	$3\frac{1}{4}$	4	$\frac{5}{8}$	$\frac{3}{4}$
1	$1\frac{11}{32}$	5	$3\frac{3}{4}$	4	$\frac{5}{8}$	$\frac{7}{8}$
$1\frac{1}{4}$	$1\frac{11}{16}$	$5\frac{1}{4}$	$3\frac{7}{8}$	4	$\frac{5}{8}$	$\frac{7}{8}$
$1\frac{1}{2}$	$1\frac{29}{32}$	6	$4\frac{1}{2}$	4	$\frac{3}{4}$	1
2	$2\frac{3}{8}$	$6\frac{1}{2}$	5	8	$\frac{5}{8}$	1
$2\frac{1}{2}$	3	$7\frac{1}{4}$	$5\frac{3}{4}$	8	$\frac{3}{4}$	$1\frac{1}{8}$
3	$3\frac{1}{2}$	8	$6\frac{1}{2}$	8	$\frac{3}{4}$	$1\frac{1}{4}$
$3\frac{1}{2}$	4	9	$7\frac{1}{4}$	8	$\frac{7}{8}$	$1\frac{1}{4}$
4	$4\frac{1}{2}$	$9\frac{1}{2}$	$7\frac{3}{4}$	8	$\frac{7}{8}$	$1\frac{3}{8}$
$4\frac{1}{2}$	5	10	$8\frac{1}{4}$	8	$\frac{7}{8}$	$1\frac{1}{2}$
5	$5\frac{1}{2}$	11	$9\frac{1}{4}$	12	$\frac{7}{8}$	$1\frac{5}{8}$
6	$6\frac{1}{2}$	12	$10\frac{1}{4}$	12	$\frac{7}{8}$	$1\frac{5}{8}$
7	$7\frac{1}{2}$	$13\frac{1}{2}$	$11\frac{1}{2}$	12	1	$1\frac{3}{4}$
8	$8\frac{1}{2}$	$14\frac{1}{2}$	$12\frac{1}{2}$	12	1	$1\frac{7}{8}$
9	$9\frac{1}{2}$	16	14	16	1	2
10	$10\frac{1}{2}$	17	15	16	1	2
11	$11\frac{1}{2}$	$18\frac{1}{2}$	$16\frac{1}{4}$	16	$1\frac{1}{8}$	$2\frac{1}{8}$
12	$12\frac{1}{2}$	$19\frac{1}{4}$	17	16	$1\frac{1}{8}$	$2\frac{1}{4}$
13	14	$21\frac{1}{2}$	19	16	$1\frac{1}{4}$	$2\frac{3}{8}$
14	15	$22\frac{1}{2}$	20	16	$1\frac{1}{4}$	$2\frac{3}{8}$
15	16	$23\frac{3}{4}$	$21\frac{1}{4}$	20	$1\frac{1}{4}$	$2\frac{1}{2}$
16	17	[illegible]	$22\frac{1}{4}$	20	$1\frac{1}{4}$	$2\frac{5}{8}$

(Courtesy Messrs. Aiton & Co. Ltd., Derby)

TABLE R

Particulars of Wrought Steel Pipes

Suitable for a working steam pressure up to 600 lb per sq in.

Nominal Bore	Actual Outside Diameter	Diameter of Flange	Diameter of P.C.	Number of Bolts	Diameter of Bolts	Thickness of Flange
in.	in.	in.	in.		in.	in.
½	27/32	4½	3¼	4	⅝	¾
¾	1 1/16	4½	3¼	4	⅝	¾
1	1 11/32	5	3¾	4	⅝	⅞
1¼	1 11/16	5¼	3⅞	4	⅝	⅞
1½	1 29/32	6	4½	4	¾	1
2	2⅜	6½	5	8	⅝	1
2½	3	7¼	5¾	8	¾	1⅛
3	3½	8	6½	8	¾	1¼
3½	4	9	7¼	8	⅞	1¼
4	4½	9½	7¾	8	⅞	1⅜
4½	5	10	8¼	8	⅞	1½
5	5½	11	9¼	12	⅞	1⅝
6	6½	12	10¼	12	⅞	1¾
7	7½	13½	11½	12	1	1⅞
8	8½	14½	12¾	12	1	2
9	9½	16	14	16	1	2⅛
10	10½	17	15¼	16	1	2¼
12	12½	20	18	16	1⅛	2½
14	15	23	20¾	16	1¼	2¾
16	17	25¼	23	20	1¼	3

(Courtesy Messrs. Aiton & Co. Ltd., Derby)

CHAPTER VI

STEAM TRAPPING

It is possible that there is no piece of industrial apparatus more maligned than the steam trap. Yet, in general, the traps offered by manufacturers are adequate and well designed, and given proper conditions these devices are capable of performing their function satisfactorily. The question of conditions is the crux of the matter. As was stated in the last chapter, it is useless to install a trap which is efficient in itself, in a position or in such a manner that the condensate, which it is the duty of the trap to remove, cannot flow freely to it. This particular installation fault has already been dealt with, but there are many others.

It is not too much to say that eighty per cent of the troubles in trapping installations are due not to the traps themselves, but to incorrect choice and/or fitting. If the installation is to be successful it is necessary first of all to consider the conditions under which the trap or traps will be required to work, from every aspect.

Roughly, the points to be appreciated are—

1. Quantity of condensate to be discharged.
2. Character of the discharge, that is to say, the ratio of the maximum to the average, and the frequency of the peaks.
3. Temperature and pressure of the steam with which the appliance to be drained is served.
4. General character of the appliance, particularly with regard to the liability to water hammer, air pocketing, etc.
5. Possible situations for the traps, with special regard to length of drop legs, return condense connections, and the like.
6. Whether the trap will be required to lift the condensate, and the effect upon the discharge capacity of back pressure.

Metallic Traps

At this point it will be well to describe in some detail some of the most common forms of trap, and to discuss the working characteristics of each. In very general terms, traps may be divided into two classes—thermostatic and mechanical. There is a third class, the pumping trap, to which special and separate consideration must be given.

The simplest form of thermostatic trap is shown in Fig. 72. As the name implies, it depends for its action upon the difference in temperature between steam and condensate. Within an outer casing is fitted a metallic tube *A*, rigidly attached to the casing at one end, and formed to a valve seating *B* at the other. Fitted through the casing at the valve end of the trap is a metal rod with a valve end *C*, which may be adjusted in relation to the tube by a screw and lock-nut passing through the casing.

When steam is turned on to the appliance, the trap being cool the air in the system is ejected through the trap. The

FIG. 72. SIMPLE FORM OF METALLIC EXPANSION TRAP
(Spirax Manufacturing Co., Cheltenham)

cool condensate follows and continues to discharge at the full capacity of the trap until the temperature rises sufficiently to close the valve partially. If correctly adjusted, the valve should close completely at the first whiff of steam. The trap is adjusted by slacking back the adjusting screw until steam blows, and then turning in the opposite direction until the discharge just stops. This means that the trap is set to close at that particular temperature, but it is important to remember that, should the pressure of the steam (and therefore the temperature) in the system vary from that at which the setting was made, the working of the trap will be upset.

It is obvious that since the closing point of the trap is determined by a fixed temperature, if the steam pressure is reduced the valve will open before condensate has completely formed, and live steam (at the reduced temperature) will be blown out. On the other hand, should the pressure in the system be increased, the valve will not open until the condensate has fallen to the predetermined temperature, so that condensate will be held up. Because the actual movement of the elements closing the valve is so small through the working temperature range, the appliance must be made large in order to be effective, so that this simple form of trap is apt to be cumbersome. In spite of this, its simplicity, and the important fact that air is discharged freely and will not lock the trap makes the appliance very useful for many purposes where space is not vital. Its worst feature is that variations in steam pressure upset its operation.

Liquid Expansion Trap

An attempt to combine the advantages of the metallic expansion trap with a more compact design is shown in Fig. 73. This is the liquid expansion trap. The construction is clearly

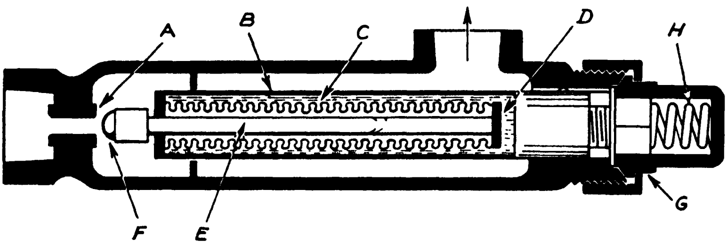


FIG. 73. LIQUID EXPANSION TRAP
(*Spirax Manufacturing Co., Cheltenham*)

shown. Instead of the expanding metal rod the motive element is a tube *B*, filled with a suitable liquid, which freely responds to variations of temperature. This liquid is sealed within the tube, in which there is fitted a piston *D* whose rod *E* carries a valve *F* at the extremity outside the tube. The seating for the valve *A* is formed in the body of the trap. Leakage from the tube past the piston is prevented by a special bellows gland *C*, which extends the whole length of the rod inside the tube, and is sealed both to the piston and to the tube itself.

Like the metallic expansion trap, the action depends upon the difference in temperature between steam and condensate, but the larger movement made possible by the use of the liquid element allows the trap to be designed on very much smaller lines. There is, however, one important difference between the liquid trap and the one previously described. The liquid element is set at the discharge end of the trap, and therefore its temperature will not be higher than that at which condensate is formed, normally for atmospheric discharge 212° F. Where the trap discharges to a back pressure this will be correspondingly higher, as the condensate will then form at some higher temperature, determined by the pressure. But since the trap normally discharges at some temperature below 212° F, a drop in the steam pressure will not affect its working, provided that the pressure of the steam is such that its temperature does not fall below 212° F. The output will be affected by a fall of pressure only because of the decreased differential pressure across the trap, but the appliance will continue to work normally.

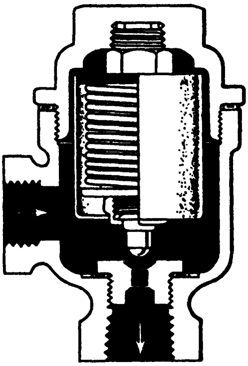


Fig. 74. Balanced Pressure Trap

(*Spirax Manufacturing Co., Cheltenham*)

The effect of an *increase* of pressure is to slow down the working of the trap, because time must elapse for the condensate to fall to the temperature at which the trap operates. Except at light loads the discharge from the trap is continuous, the valve assuming a position at which condensate is discharged as formed.

For use with superheated steam, where the elevated temperatures may cause excessive closing pressures on the valve, a safety-spring *H* is incorporated in such a way as to relieve the stresses before damage is done to the element.

Characteristics of the trap are: free discharge of air, continuous discharge at temperatures below 212° F. Output is affected by variation of steam pressure, but the trap does not stop working.

Balanced Pressure Trap

Yet a third type of thermostatic trap is the balanced pressure trap (Fig. 74). Although in its essentials there are many points of similarity between this and the types which have been described, there is an important difference in principle, by which the balanced pressure trap is made to work effectively through wide limits of steam pressure. Briefly, this principle is such that there is always sufficient closing pressure in excess of the steam pressure to ensure that the trap will work, whatever that pressure may be. Superheat will however damage the element, and the trap must not be used where superheat is necessary.

The element consists of a bellows vessel, which is partly filled with a mixture of water and a volatile fluid, whose boiling point is less than that of water. Thus, for any given temperature, the pressure within the element will always exceed that of the steam surrounding it by an amount depending on the difference in the vapour pressures of water and the mixture. The traps can be designed to close at some predetermined temperature below that of saturated steam. Normally, the pressure difference between the interior of the element and the body of the trap under working conditions will not exceed a predetermined figure, but should superheated steam have access to the trap it is easy to see that because the temperature surrounding the element exceeds that of the corresponding pressure for saturated steam, the vapour pressure within the element, in responding to the higher temperature, is exerted at a lower pressure, and damage may be done.

Assume for instance that the mixture within the element is such that, with saturated steam, the bellows will exert a pressure of 20 lb in excess of the steam. Let the system on which the trap is fitted be working at 100 lb per sq in. gauge. Then, when steam at this pressure enters the trap, the internal pressure of the element is 120 lb per sq in. But the temperature of saturated steam at 100 lb per sq in. is 338° F, so that we may say that a pressure of 120 lb per sq in. within the element corresponds to a temperature of 338° F. But if steam at 100 lb per sq in. and 100° F of superheat were admitted to the trap, the temperature would be 438° F, which corresponds to a saturated steam pressure of 355 lb per sq in. The trap being designed to exert 20 lb per sq in. more than the corresponding pressure for saturated steam, the pressure within the element is now 375 lb per sq in. Thus the pressure difference between the inside of the element and the surrounding steam will be 275 lb per sq in. instead of the designed 20 lb, an excess which no apparatus could be expected to withstand.

The characteristics of the balanced pressure trap are, therefore, free discharge of air with the steam and condensate; it will work over a wide range of pressure; it cannot be used with superheat. The discharge from the trap is intermittent, and the shut-off is definite. Owing to the inherent weakness of the bellows element in a lateral direction, the trap should not be used in situations where water hammer is likely.

Mechanical Traps

Mechanical traps are marketed to-day in very great variety and an amazing diversity of design. Speaking very generally, they may be divided into three classes—the bucket trap, the inverted bucket trap, and various types of float trap. It is to be understood that all of these three classes vary very greatly within themselves, and that there are many different methods by which the fundamental principle of each class is carried out.

Bucket Trap

Fig. 75 shows the simplest possible design of bucket trap. The bucket may be used as shown or through the medium of levers, while some designers, using the direct fall method, fit small pilot valves and helical guides on the central tube, so as to give a rotary motion to the bucket as it falls away from the valve seat. This combats the tendency for the valve spindle to jam at any part of its travel. But whatever the final design may be, the operation of this type of trap has certain definite characteristics.

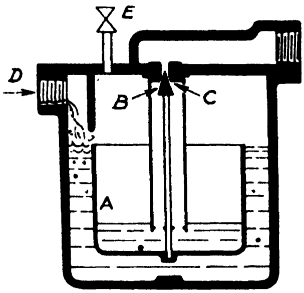


FIG. 75. FUNDAMENTAL OPEN TOP BUCKET TRAP
(*Spirax Manufacturing Co., Cheltenham*)

Condensate pours into the body of the trap *D*, and the bucket *A* floats higher and higher in the body, until the valve *B* at the top is closed. Condensate then pours over the lip of the bucket (which, when the valve is closed, can rise no higher) and begins to fill it. When the weight within the bucket is sufficient to cause it to sink, the downward movement opens the valve, and the condensate is blown out. The weight of the bucket is so arranged that a little water always remains to form a seal with the central tube. The greatest difficulty with this type of trap is air locking. Air collects in the top of the trap body, and unless some means is available to release it, the pocket thus formed will prevent the ingress of further condensate, and the trap stops working. An air valve *E* is provided in the cover plate, but this must be operated manually at the appropriate time in the working cycle. The trap is fairly substantial, it does not easily get out of order, the discharge is intermittent, and the condensate is discharged at steam temperature.

Since the forces opening and closing the valve depend, on the one hand, on the force exerted by the condensate upon the surface of the bucket to close the valve, and on the other hand, on the weight of the bucket to open the valve against the steam pressure, it follows that the pressure at which the trap is required to work must influence the size, and for the higher pressures this type of trap tends to be cumbersome and heavy.

Inverted Bucket Trap

The inverted bucket trap is somewhat similar in external appearance to the trap described above. There are many variations of the fundamental design which is shown in Fig. 76, in which an inverted bucket is the operating element.

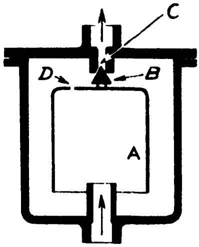


FIG. 76. FUNDAMENTAL INVERTED BUCKET TRAP

(Spirax Manufacturing Co., Cheltenham)

Besides the lever method, in which the movement of the falling bucket is transmitted to the valve in such a way that a mechanical advantage is obtained, an arrangement is sometimes used in which the impetus of the bucket is utilized to open or close the valve. In this design the bucket is allowed some motion free of the valve, so that it possesses some slight momentum before performing work in operating the valve.

An important feature in the fundamental design is the small hole *D* in the top of the bucket. To some extent, the size of this hole determines the speed of working of the trap, or rather the ability of the trap to respond to the variations of the steam cycle. In some makes, a thermostat is fitted instead of the usual small hole, but a little consideration will show that the temperature difference available to operate the device is so small that its utility is perhaps of doubtful value.

The operation of the trap is as follows. When empty, the bucket *A* lies on the bottom of the trap, and the valve *C* is open, held so by the position of the bucket. When steam is turned on, any air which may be present is blown out round the bottom of the bucket, and through the hole *D* in the top. This continues until the condensate which follows is in sufficient quantity to close the bottom of the bucket, when the remaining air can escape only through the hole. The condensate now fills the trap and the bucket, the rate of filling the bucket being determined in some degree by the size of the hole, and therefore the rate of air displacement from the inside. When steam reaches the trap it is passed directly to the inside of the bucket, and, although some slight amount is lost through the hole, most of it is used to displace the water, so that the bucket is given buoyancy which causes it to rise and close the valve. Further steam coming to the trap will now be condensed by the water, which continues to rise until the buoyancy of the bucket is lost, and the cycle is repeated.

Condensate is discharged from these traps intermittently, and practically at steam temperature. The trap is substantial and sturdy. Normally the trap is made to operate at one pressure, which must be specified when ordering. Since the valve is kept closed by the steam pressure, and the opening force is the weight of the bucket, it follows that the relation of bucket weight to valve diameter must be determined for any particular steam pressure. Also, for closing, the bucket must be capable of approaching the valve to its seat so that the final closing pressure may be applied by the steam pressure.

It is obvious therefore that the trap will only work satisfactorily at the one pressure for which it is designed. Should the pressure be increased, the weight of the bucket will not be sufficient to pull the valve away from its seating at the appropriate time, and the trap stops working. On the other hand, an appreciable fall of pressure not only decreases the discharge capacity of the trap but allows dribbling to take place, because the unit pressure on the valve area is insufficient.

Closed Float Trap

Perhaps the largest group of all purely mechanical traps is that which embraces all those appliances which come under the general description of closed float traps. The variety of design in this class alone is so great that it is not possible in a general work of this nature to give more than general principles of some of the more usual types. The reader is referred to Mr. L. G. Northcroft's excellent and comprehensive *Steam Trapping and Air Venting* for more detailed descriptions of other designs.

Loose Float Trap

Fig. 77 shows a simple form of loose float trap. The sketch is self-explanatory. Before putting to work, the body of the trap C must be filled with water. When pressure is applied the ball moves towards the outlet orifice A but cannot seal it until the condensate level has fallen to that of the outlet. At this point only, the position of the ball is such that an effective seal is made by the pressure of the system forcing the float against the orifice. In practice, the ball assumes a position relative to the outlet, depending on the rate at which condensate is formed, so that the discharge is continuous and at steam temperature.

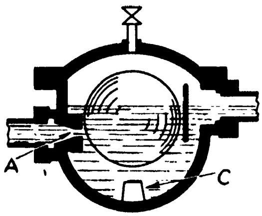


FIG. 77. LOOSE BALL FLOAT TRAP
(*Spirax Manufacturing Co., Cheltenham*)

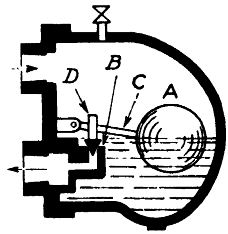


FIG. 78. CLOSED FLOAT TRAP WITH MANUALLY OPERATED AIR RELEASE
(*Spirax Manufacturing Co., Cheltenham*)

Since the discharge pipe is below the level of the inlet the trap has a tendency to air lock, and provision is made to release the entrapped air by a small manually-operated valve. The power to open the valve is the buoyancy of the ball, which must exceed the out-of-balance pressure across the trap in order for the appliance to work. Therefore, much the same considerations apply as were discussed in relation to the inverted bucket trap, that is to say, the loose float trap will only work satisfactorily and with full output at one pressure for which it must be designed.

Lever Float Trap

Another form of closed float trap is given in Fig. 78, which shows a design in which the buoyancy of the float A is utilized through a lever C to open and close the valve B. In this trap also special provision must be made for the release of air, and the valve must be operated when the appliance is first put into service. Following the released air, the condensate fills the trap, and raises the float which opens the discharge valve. Because of the lever system the buoyancy of the float is utilized at some mechanical advantage, so that some latitude is permissible in the range of pressure over which the trap will continue to work. Discharge is continuous, and at steam temperature.

An improved form of the foregoing design is shown in Fig. 79. This is essentially the same trap with a slightly different lever system, and fitted with a thermostatic air-venting device. The thermostatic element *D* will open with either air or condensate, and closes with steam. Therefore there is no need either to fill the trap with water when first setting to work, or to operate any air valve, the action of which is entirely automatic, because when the trap is cold, the thermostatic element holds its valve *E* open, and air is free to escape through that valve when pressure is first applied to the trap. Moreover, the air valve *F* will remain open until steam fills the trap. This feature has the advantage that during the period of initial condensation, that is to say, while the system is warming up, and the condensation rate is at its highest peak, the trap possesses what is practically a double discharge. During normal working, the thermostatic valve will remain closed until an accumulation of air in the upper portion of the trap body permits its temperature to fall below that of the steam. The valve then opens, and the air is discharged automatically.

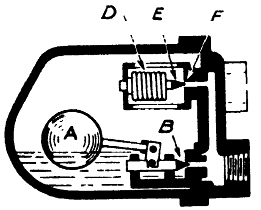


FIG. 79. CLOSED FLOAT TRAP WITH THERMOSTATIC AIR RELEASE
(*Spirax Manufacturing Co., Cheltenham*)

Pumping Trap

Where it is necessary to return condensate against a pressure exceeding that of the steam within the trap, special arrangements must be made to augment that pressure to the desired amount, from an outside source. By this means it is possible to return the condensate from low pressure systems direct to the boiler, or against heads far in excess of those which would be possible with ordinary traps.

The device used is the pumping trap, a simple form of which is shown in Fig. 80. Essentially, the apparatus consists of a container fitted with inlet and outlet check valves near the bottom, and a guided float, which is free to move vertically as the condensate flows by gravity into the trap. In the cover of the container are two valves, actuated by the movement of the float, and connected respectively to the inlet and exhaust systems of the auxiliary pressure supply. This may be either steam or compressed air. The action is as follows. Condensate flows to the trap through the inlet check valve *A*, causing the float to rise vertically on its guide. Near the top of its travel two functions are performed by the float, i.e. the

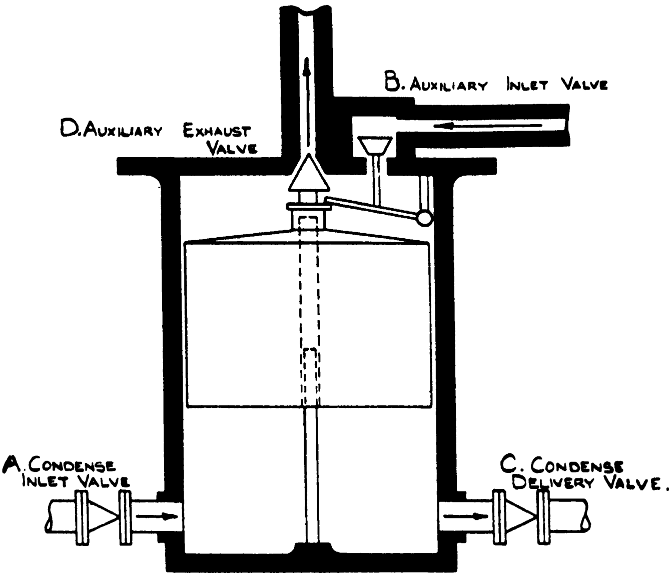


FIG. 80. PUMPING TRAP

auxiliary inlet valve is opened, and the exhaust valve closed. Because the exhaust valve is closed, the pressure admitted through the inlet valve *B* forces the float down, driving the condensate through the delivery valve *C*. But the downward movement of the float opens the auxiliary exhaust valve *D*, which releases the pressure from the upper portion of the trap, so that the cycle is free to commence again. Each time the trap works a puff of the auxiliary medium, either steam or compressed air, is emitted through the exhaust.

The foregoing descriptions, while not exhaustive, cover a fairly wide range of the more common forms of trap, and give sufficient information for the subject of trap selection to be discussed in some detail. Brief mention has been made in a previous chapter of the necessity of ensuring that the trap must be so fitted that the condensate has free and unrestricted flow to it, the case quoted in particular being in connection with the drainage of steam mains.

Air Locking

But the provision of adequate drop legs as shown in Figs. 70 and 71 is not the whole story, elementary as it may appear, and when dealing with apparatus rather less straightforward than steam mains, some consideration must be given to the situation of the trap, with a view to the prevention, if possible, of two other phenomena—air binding and steam locking.

Certain types of trap discharge air automatically. Roughly, it may be said that all traps which remain open when cold will not suffer from air locking because, being open when steam is first turned on, any air which may be entrapped in the system can freely escape through the trap before the arrival of the condensate. Should air collect in the steam space during working, its arrival at the element of a thermostatic trap will operate that element in the same way as cool condensate, so that either the metallic expansion trap, the liquid expansion trap, or the balanced pressure trap, forms a complete answer to the problem of air binding. But from other considerations it is not always possible to install either of the above, and it may be necessary to fit one or other of the bucket or closed float traps. All these remain closed until the arrival of a certain amount of condensate, and all, with the exception of the special design shown in Fig. 79, are prone to air locking. True, they are for this reason provided with air release vents, but unless much steam is to be wasted, or on the other hand much production time is to be lost due to cooled apparatus, these vents should be operated only when required, and at the correct time in the working cycle. Unfortunately this operation, simple in itself, can seldom be performed satisfactorily in practice, either through the inaccessibility of the trap, lack of personnel, or just plain ignorance on the part of the machine workers. It is a little too much to expect the maintenance engineer or his staff to keep track each hour or so of every trap in a large factory, so that the position frequently arises that either the air vents are left open altogether—greatly to the detriment of the steam economy—or, especially in those cases where air is liable to collect in the steam spaces of the plant during working, a perfectly good and reliable trap is removed, and that abomination of desolation, the cracked or drilled valve, is substituted.

The only satisfactory way of dealing with the problem is to devise some means of releasing the air automatically at the correct point in the working cycle. In the design shown in Fig. 79 this has already been done by the incorporation of a thermostatic unit in the body of the trap. This particular trap cannot air-lock, for the reason already given, i.e. air passing into the body of the trap at a lower temperature than that of the steam acts upon the element in exactly the same way as the cool condensate, and causes the valve to open. In other words, the fact that the thermostatic element is responsive either to air or condensate below the steam temperature makes it ideal for the purpose of automatic air release.

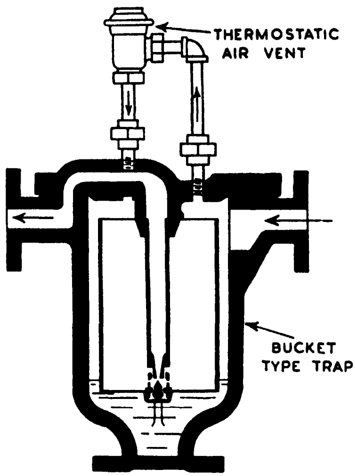


FIG. 81. BUCKET TRAP WITH BALANCED PRESSURE TRAP AS AIR RELEASE
(*Spirax Manufacturing Co., Cheltenham*)

Combined Traps

A simple and effective method of combining the advantages of the thermostatic trap for air release with those of the mechanical trap is shown in Fig. 81, which illustrates a mechanical trap fitted with a by-pass, in which is inserted a balanced pressure trap of the type already described. The inlet to the latter is taken from the point on the mechanical trap where the air vent would normally be fitted, that is to say the top cover, and the outlet from the thermostatic trap is taken to the mechanical trap outlet, so that air (and when heavily loaded possibly some condensate as well) can pass via the element to the outlet. As an alternative, a complete by-pass could be fitted entirely independent of the mechanical trap, by using an arrangement as shown in Fig. 82. Provided that the inlet pipe to the thermostatic element rises above the mechanical trap so that the air naturally collects at that point, the fit-up is perfectly satisfactory. In the opinion of the author, one or other of the above arrangements is the only really satisfactory method of dealing with the problem, since the thermostat is the only simple device which will ensure that the air vent is opened at the correct point in the working cycle.

Steam Locking

Perhaps one of the most common evils associated with steam-trapping installations is steam locking. This phenomenon, which is far more common than is generally realized, may be due to faulty installation, but very frequently the design of the apparatus to be drained is such that steam locking is almost bound to occur. In the latter case neither the trap itself nor the method of installation can be blamed. The symptoms are not easy to pin down. With a continuous discharge type of trap, an almost complete cessation of discharge may give the impression, not that the working of the trap is at fault but that no condensate is being formed, while if the trap is one which normally discharges intermittently, there may be no clue whatever that steam locking is taking place, for the only effect may be to lengthen the periods between the discharges. This might quite naturally be put down to some reduction in the rate of formation of the condensate due to some cause quite outside the scope of the steam engineer, and rather in the province of the production people.

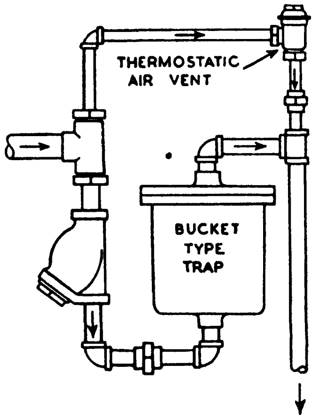


FIG. 82. BUCKET TRAP WITH BALANCED PRESSURE TRAP IN INDEPENDENT BYPASS
(*Spirax Manufacturing Co., Cheltenham*)

Consider the diagram in Fig. 83. The plant to be drained is fitted with a long horizontal pipe down to the trap. When the system is thoroughly heated condensate will be formed in the apparatus more rapidly than the steam can condense in the pipe and the trap. The result is that the trap is kept closed by the steam within it, and no more condense can flow from the body of the apparatus until this steam has either been condensed or dispersed. Eventually it will be condensed, but the effect is to slow down the working of the trap, and in a serious case of steam locking this slowing down may be so great that the working of the apparatus from the production standpoint is seriously impaired.

A thermostatic by-pass such as that described for air locking will obviously not help in this case, because the cause of the

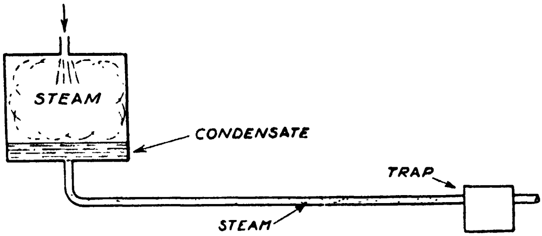


FIG. 83. STEAM LOCKING IN LONG HORIZONTAL PIPE
(*Spirax Manufacturing Co., Cheltenham*)

trouble is not air but steam, and the thermostat would therefore remain closed at the crucial moment. A by-pass taken back to the body of the apparatus may help, but the real cure in this particular instance is to shorten the pipe from the apparatus to the trap, so that the steam locking does not take place.

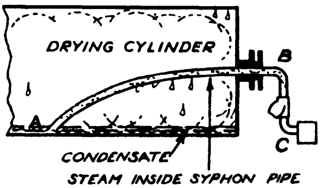


FIG. 84. CYLINDER WITH TRUNNION OUTLET, ILLUSTRATING STEAM LOCKING
(*Spirax Manufacturing Co., Cheltenham*)

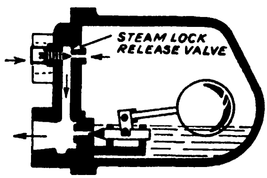


FIG. 85. CLOSED FLOAT TRAP WITH STEAM LOCK RELEASE
(*Spirax Manufacturing Co., Cheltenham*)

Unfortunately this may not always be possible, and in certain classes of apparatus, for instance some types of calender, and heaters with cored jackets, nothing that can be done in the arranging of the external piping will mitigate the liability to steam-lock, because the trouble really lies within the apparatus itself.

Steam Lock Release Traps

Fig. 84 illustrates an example in which the arrangement of internal piping forms a steam siphon, which is bound to steam-lock to some extent, and which no arrangement of external piping can prevent. The only cure in this case is, in effect, to allow a small leak from the steam trap so that the steam which is the cause of the trouble can dissipate.

A trap fitted with a steam lock release is shown in Fig. 85. This device consists of a small needle valve fitted as an integral part of the trap, in such a way that the leak-off steam is led back to the trap outlet.

A form of steam locking which is purely an installation fault and which is easily cured is shown in Fig. 86. This is not steam locking proper, and the effects are not so subtle.

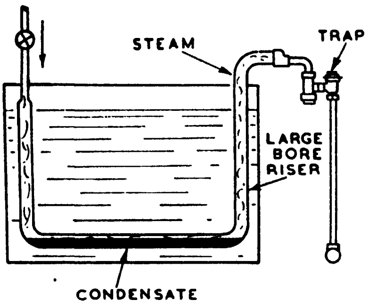


FIG. 86. TANK HEATING ELEMENT, ILLUSTRATING STEAM LOCKING
(*Spirax Manufacturing Co., Cheltenham*)

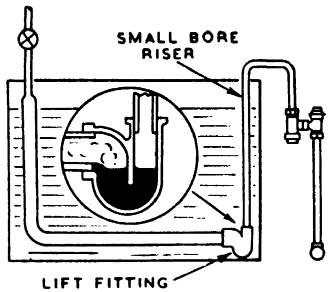


FIG. 87. WATER SEAL APPLIED TO TANK HEATING ELEMENT TO PREVENT STEAM LOCKING
(*Spirax Manufacturing Co., Cheltenham*)

As has been explained, true steam locking may never be suspected, but in the case now to be discussed, the effect is so drastic that trouble is in evidence at once. It is to be regretted that usually the trap is blamed or perhaps the amount of heating surface in the tank is increased, with the result that not only is the trouble not cured, that is to say the material to be treated will still not heat up, but more often than not, the provision of further pipes, and therefore more water volume, actually worsens the position. Consider the diagram for a moment. The apparatus consists of a very common form of heating appliance for the treating of liquids, a heating coil in the bottom of a tank, the outlet from the coil to the trap being a vertical riser taken from the easiest position (for the pipe fitter), a tee piece with its outlet at the top into which the riser fits. From the tee piece the riser continues vertically up the side of the tank and over the edge to the trap which is below the highest point of the riser.

When steam is turned on to this arrangement, the air (in the case of a balanced pressure or similar trap) is expelled through the trap outlet. Steam is condensed in the heating coils, and the condensate lies in the bottom of the pipes. Little if any condensate can reach the trap because the steam can pass freely over the condensate and up the riser to the trap, which consequently remains closed. Moreover, the trap must remain closed so long as steam has access to it, that is to say, until the pipes are completely filled with condensate. Then, and only then, some is forced up the riser to the trap, which then opens. But only a very little condensate needs to be blown out before the steam again has access to the trap, which then closes until the pipes are again filled. The result is that the process liquid, if heated at all, is heated by warm water instead of steam.

Fortunately, the cure is simple. Instead of the tee piece with top outlet, a small water seal is fitted at the bottom of the riser, as shown in Fig. 87. This little fitting seals the riser directly condensate begins to form, and thus prevents the direct access of steam to the trap. In order to prevent true steam locking, the trap is also brought nearer to the tank itself.

Dirt

One of the greatest enemies to the correct working of all forms of steam trap is dirt. In all traps, the outlet is small, in some it is particularly so, and in the case of most expansion traps, the total movement of the valve from full open to shut is very small indeed. It follows therefore, that the smallest piece of pipe scale, or loose dirt of any kind, is sufficient to interfere with the working if it is allowed to enter the trap.

Two points may be made clear at once. First, however long the system may have been in service, and however clean the pipes may supposedly be, there is always the possibility of dirt getting to the trap from some part of the system, and secondly, there is but one satisfactory means of dealing with it. This is an efficient and well-maintained strainer installed immediately before the trap. Dirt pockets, although very frequently fitted, can only be regarded as a palliative, and may under certain circumstances be worse than nothing at all. Apart from the fact that the smaller particles may be swept through to the trap without entering the pocket at all, there is always the possibility that any of the disturbances that are always taking place in a steam system may be sufficient to stir up the deposits already in the pocket and carry them into the trap, with results that can be imagined.

Strainer

A well-designed strainer is effective, provided that it is well serviced and maintained, but the service must include routine blowing out. Such a strainer is shown in Fig. 88. It is simple and inexpensive, and adequate maintenance can be carried out in a few minutes. It is always well to carry a small stock of spare strainer elements on hand, at all events sufficient to supply the more important traps at need.

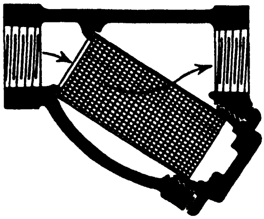


FIG. 88. STEAM STRAINER (*Spirax Manufacturing Co., Cheltenham*)

Choice of Traps

Having dealt with trap characteristics and some of the more common installation faults, one has to consider the correct choice of trap for use with factory apparatus. Obviously the field is so large that it is impossible to treat the subject exhaustively, and it is therefore proposed to deal with only a few instances, chosen as typical of industrial practice. Perhaps one of the most common uses for traps in almost any factory is in connection with space heating, and very frequently for cheapness, ease of installation, and space saving, this is carried out by means of overhead pipes.

A typical lay-out is shown in Fig. 89. Each leg should be separately trapped as indicated. Study for a moment the conditions under which the traps will have to operate. During the lay-off in the summer it is inevitable that the whole system will become filled with air, so that the trapping installation must provide for automatic air release. When steam is first turned on there will be heavy initial condensation, so that in addition to the air release feature, the traps must be capable of large overload capacity. This may quite easily amount to five times the normal trap output. If the system is a large one with very long pipes it may not be possible to avoid a certain amount of water hammer, and this is an important factor in the choice of trap. If the system is properly designed with short runs and adequate fall to the traps, the first choice would be the balanced pressure thermostatic trap, on account of lightness, low first cost, and the characteristic air release feature. These traps are particularly good for the lower pressure ranges, they have good overload capacity, and, as already explained, are not affected by changes of steam pressure.

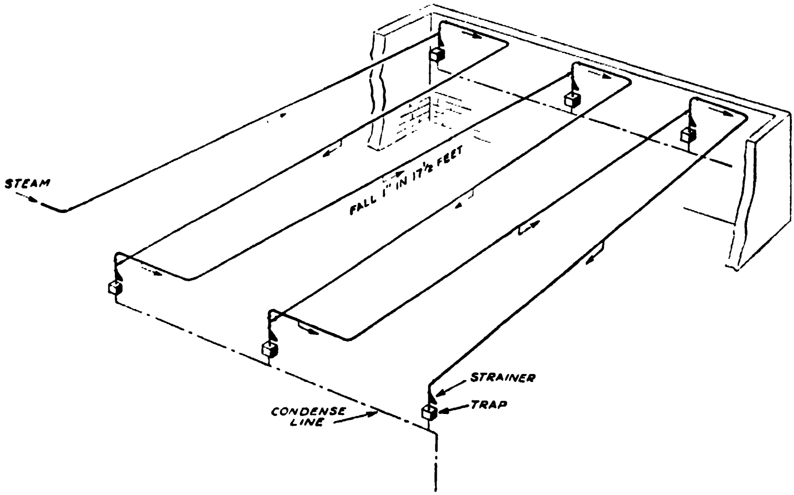


FIG. 89. LAY-OUT FOR SPACE HEATING WITH OVERHEAD PIPES
(Spirax Manufacturing Co., Cheltenham)

Liquid expansion traps would be suitable, since they possess the air release qualification, but are affected somewhat by pressure changes. Increases of pressure would tend to slow the trap down somewhat, so that there would be a tendency for hot water to lie in the pipes. So long as this water is giving up its heat to the surroundings at an adequate rate, all may be well, but if the heating surface has been designed for steam temperatures it will not be sufficient to give comfort conditions with the pipes full of water below 212° F.

Neither of the above traps may be used if the system is liable to water hammer.

Open top bucket traps may be used, but provision must be made for automatic air venting. A good combination would be an open top bucket trap fitted up with an expansion trap as air vent, as shown in Figs. 81 and 82. This combination is suitable for the lower pressures but, since the size of the bucket trap is to some extent influenced by the pressure, it is apt to become cumbersome where the system operates above 120 lb per sq in.

The closed float trap with automatic air release incorporated is an effective solution, and possesses the additional advantage that the overload capacity is exceptional, for the reason that when the condensate accumulation is really heavy, some of it is discharged through the air release by-pass duct. The closed float trap must not, however, be used if water hammer is really heavy.

Unit Heaters

Unit heaters are probably the second most common form of appliance in use for industrial space heating. Here the problem is a little different. First of all, the heater is usually suspended from the roof, so that there is some restriction as to weight. Under certain conditions, notably when the heaters are used in conjunction with thermostatic control, load fluctuations are very violent. Condensate accumulations are exceptionally heavy, so that the discharge from the trap should be continuous and, since the whole of the air in the pipe system leading to the heater must pass through the trap, it must be disposed of on arrival. Air discharge is thus of primary importance, and this fact alone points to the choice of the balanced pressure trap. This choice is additionally supported by the wide range of pressure over which the trap will operate. This feature is so important that when, as frequently happens, the heaters are operated either from an exhaust steam main or through a single reducing valve, so that every load change will affect the pressure, the balanced pressure trap, although not ideal, may be the only type which will give satisfactory working. The one feature of the trap which militates against its choice is the fact that the discharge is not continuous.

The trap which possesses all the characteristics required for this particular service is again the closed float trap fitted with automatic air release. It has the requisite ability to deal both with air and heavy load fluctuations, and is not affected by normal pressure changes. Water hammer of sufficient violence to upset or damage the trap is not likely to occur. The only objectionable feature is that the trap is a little heavy, and first cost is rather high.

Both types of trap should, if possible, be fitted below the heaters. If this is not possible, then lift fittings as previously described *must* be fitted at the bottom of the vertical leg rising to the trap. If the thermostatic trap is chosen, the makers should be informed as to the particular service, so that suitable near-to-steam elements can be provided. Should the closed float trap be chosen, and should it be necessary to install it above the heater, in that case the trap must be provided with a steam lock release device. This entails a small continuous steam leak from the upper part of the trap to the discharge outlet, and some steam wastage is therefore inevitable. This is one of the strongest reasons for not fitting the trap in this position unless there is no alternative.

Pans

Jacketed boiling pans are often the cause of much heart-burning among operatives. Even in these enlightened days there are many users who will not hear of traps in connection with these appliances. The reason is not far to seek. As a general statement, it is true to say that with by far the majority of boiling pans the shape and position of the steam passages do not lend themselves to easy trapping. Again speaking generally, owing to the inherent design of the apparatus, a system of trapping alone can seldom, if ever, be successful. A skilful combination of steam traps and automatic air vents is nearly always demanded.

Consider for a moment the duties demanded of the trap. When steam is first turned on, the initial condensation will be very heavy indeed, due to the large cold surfaces. In this respect the requirements of the trap are similar to those demanded by unit heaters but, unlike unit heaters, the condensate formation falls off very rapidly, and when once the required temperature is attained the trap load may be very light. Therefore one characteristic of the trap chosen must be the ability to work efficiently over wide ranges of load, combined with the capacity to handle large quantities of condensate as quickly as possible.

Fixed Type Jacketed Pan

But a further aspect of the problem is made apparent by a study of Fig. 90, which shows a fixed type jacketed pan. Incidentally this is the easiest type of pan to deal with, for by fitting the trap at the bottom, as shown, the tendency to steam locking can be eliminated. It is clear, however, that the considerable volume of the steam jacket will be filled with air before steam is turned on, and this air must be eliminated before the steam surfaces can become effective. The trap must therefore be of the automatic air discharge type, or, automatic discharge of air must be arranged by the provision of a thermostatic by-pass, as illustrated in Figs. 81 and 82. The first choice would undoubtedly be the closed float trap with automatic air release, or the balanced pressure thermostatic trap with near-to-steam element.

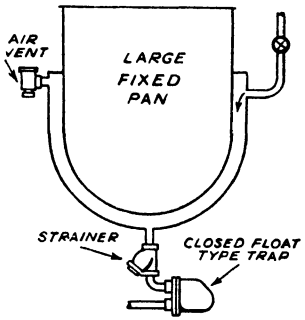


FIG. 90.
FIXED TYPE JACKETED PAN WITH BOTTOM TRAP AND AIR VENT
(Spirax Manufacturing Co., Cheltenham)

For slow processes, where the rapid elimination of the air is not of such vital importance, the inverted bucket trap may be considered, but this form is ruled out for rapid boiling pans and short runs, because the air discharge capabilities are to some extent determined by the diameter of the hole in the bucket. The open top bucket trap would be effective only if provided with a by-pass as explained above, but it must be realized that this is actually an arrangement of two traps to perform the service of one, and might be difficult to justify on grounds of cost.

The trap fitted at the lowest position of the steam space will not deal with all the air present, and it is quite clear from the diagram that pockets must inevitably form in the upper portion. No trap situated at any position below the top of the jacket can deal effectively with these pockets which will accumulate more and more air with continued use. It is thus possible for the whole of the heat transfer surfaces above the level of the steam trap to become quite ineffective, unless some special provision is made to clear the pockets of air. It is no doubt the fact that the importance of air clearance from all parts of steam jacketed apparatus has not been sufficiently realized that has led to the majority of trapping difficulties with these pans, and to the consequent wholesale and quite unjust condemnation of all types of trap for this purpose.

The solution lies in the provision of an air vent or vents as shown, at the highest possible position in the jacket. The vent may be a simple manually-operated valve, but unless the operators thoroughly understand the principles of air venting, and can be relied on to open only as and when required, this expedient is likely to lead to the waste of much steam. It is much better to fit a thermostatically-controlled vent, which simply consists of a balanced pressure trap, with near-to-steam element.

Tilting Pan

Tilting pans, especially those required for quick boiling, are a little more difficult to deal with, because both the steam inlet and the outlet to the trap must be fitted at the trunnion. For mechanical reasons it is nearly always essential to fit the trap on the floor, so that the siphon arrangement formed by the steam space of the pan and the down pipe to the trap may have a tendency to steam lock. Whatever trap is selected it must therefore be provided with means to release the "locking steam." and it is far better to fit a trap with a steam release incorporated with it than to give the operator another valve to play with, which means another source of waste. For the tilting pan, then, the choice would be a closed float trap fitted with steam lock release.

Calenders and drying rolls and all similar apparatus which must be vented through the trunnion, have the same tendency to steam lock, and consideration must be given to some form of steam release in each case.

With this class of appliance the vent pipe to the trap is usually continued through the trunnion to a pick-up point near the bottom of the inside of the roll. This pick-up point formed by the end of the pipe should be as close as possible to the wall so that the pipe end is always covered with condensate under all working conditions. The condense is forced up the pipe by steam pressure, and so long as there is enough water to prevent steam getting to the trap, steam locking will not be serious. At light loads there may be some difficulty when the rolls are thoroughly heated up so that little condense is formed. It is to cope with such conditions as these that the steam lock release becomes necessary.

But whatever trap is used it must have large capacity, in order to deal effectively with the large volume of condense due to initial condensation, and during the greater portion of the working time it will be required to operate well below its full load. But because of the importance of keeping the outlet to the trap under water it is vital that too much should not be discharged at one time. This condition at once places some limitation on the choice of trap because it is evident that the character of the discharge will have some effect on the tendency of the system to steam loss.

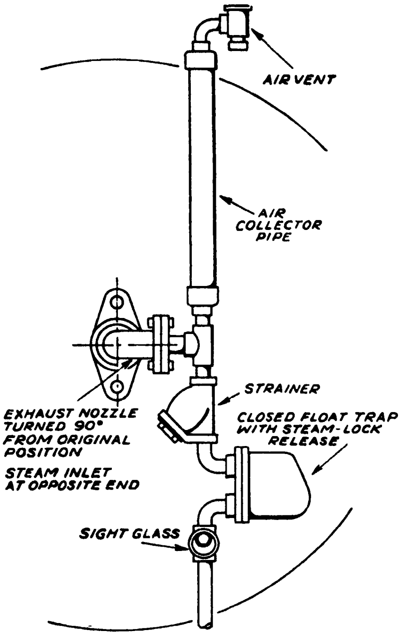


FIG. 91. CYLINDER FITTED WITH TRAP AND AIR COLLECTOR VENT
(*Spirax Manufacturing Co., Cheltenham*)

Of two traps of equal capacity under the same conditions, one giving an intermittent and the other a continuous discharge, the latter must be the correct choice because (bearing in mind the fact that either trap must be of large capacity) the intermittent discharge trap having to clear the same amount of water in bursts as the other does continuously, might well at each blast lower the level to the point at which steam has access to the pipe. Under these conditions, steam locking would be a major nuisance. The correct trap is therefore the closed float trap with steam lock release. Air venting of rolls and calenders needs special consideration, and may be a matter of some difficulty.

Air Collector Pipe

The most usual and effective means is to fit an air collector pipe as shown in Fig. 91, connected to the trunnion from which the outlet to the trap is taken. Very complicated trapping problems are presented in connection with multiple roll machines, and it is not possible to indicate any general method. Each must be taken on its merits, and due consideration given to the special circumstances involved. Here it cannot be too strongly emphasized that where special difficulties are encountered, expert advice should be sought without hesitation. Trapping is an expert's business, and there are few problems that cannot be solved. It is far better to seek advice in the early stages of a problem and be assured of satisfaction and economical working than hurriedly to condemn a perfectly good trap, install a valve, and pave the way for months or years of steam wastage.

Calorifiers

Calorifiers are frequently used in industry in two main forms, either for space heating or for domestic hot water service. No difficult problem is presented, although slightly different conditions must be considered for each type. For space heating, the system is usually turned on at the beginning of the winter and left working till the spring. The load on the trap is therefore a steady one except where thermostatic control is installed, and even then the fluctuation on the trap may not be large.

Either an open top or inverted bucket trap would serve, and with the former, although an air release must be provided, this need not be automatic if the attendant can be relied on to vent the system when steam is turned on. Since this probably will be not more often than once a week, the manual arrangement may be quite satisfactory. If thermostatic control of the off/on type is fitted then automatic air venting should be provided. For small jobs the balanced pressure thermostatic trap will work quite well provided that the correct type of element is chosen. This will depend to some extent on the actual situation of the trap in relation to the apparatus. If close, a near-to-steam element should be fitted, but if the connecting pipe is longer than three or four feet then an element with a lower temperature range will be more satisfactory. For larger jobs, a closed float trap with automatic air release may be justified in spite of its greater cost.

Air venting of the appliance itself will rarely be necessary except on very large plant where the steam space volume may be considerable.

Calorifiers for domestic hot water supply present a somewhat different problem, in that the load on the trap due to the varying demands may fluctuate considerably. If the calorifier itself has large storage capacity, and is thermostatically controlled, some of the load shock, as it were, will be taken up by the heat storage capacity of the water, but the trap must still be capable of working at wide load differences. An open type bucket trap would be quite satisfactory but it must in this case be fitted with automatic air release, that is to say the combination of two traps previously described. The expense of this arrangement would probably be more than that of the single closed float trap with its automatic air release characteristic, and this latter is preferable.

Heating elements for plenum systems of space heating installations need special consideration, with regard to both air venting and steam locking. Much will depend upon the actual design, in which there is much latitude and, as a consequence, many variations. Some tendency to steam-lock may be introduced by the situation of the trap which in most cases must, by the very design of the apparatus, be fitted some little distance away from the element itself. Because of the fact that once the apparatus is installed steam locking may continue unsuspected, and be a source of inefficiency throughout the life of the plant, it is far better to fit a trap with steam release at once. The difficulty of detecting steam locking has already been explained.

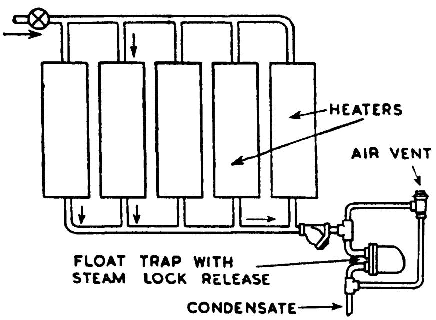


FIG. 92. SINGLE HEADER HEATER BANK WITH STEAM LOCK RELEASE AND THERMOSTATIC BY-PASS
(*Spirax Manufacturing Co., Cheltenham*)

Since water will form in considerable quantities, and since the best performance is aimed at, there must be no period during which water logging occurs, and the trap should be of the continuous discharge type.

Single Header Heater

If a trap with steam lock release is provided, then the air venting must be dealt with separately. If the heating element has a single header as in Fig. 92 the thermostatic trap used as a by-pass as described previously is quite effective. With very large headers, or in the case of elements with double headers, it may be necessary to air-vent from both ends. Should this be so, the by-pass arrangement is used at the steam trap, but an additional balanced pressure trap is fitted on a vertical vent pipe at the remote end of the element.

Small driers for certain purposes are sometimes fitted with a number of small heater elements, each consisting of a simple loop of pipe, rising vertically from a divided header. The arrangement of the pipes and the small volume of the heater itself makes for reasonably good steam distribution, and if properly installed there is little danger of water hammer. Very frequently too, the headers themselves are arranged to form the floor of the chamber enclosed by the casing which surrounds the pipes only. It is thus possible to fit the trap or traps sufficiently close to the headers to mitigate the danger of steam locking. When this can be done an altogether simpler trapping system may be used. The one important point is that each header must be separately trapped and, although because of the proximity of the separate elements to each other the temptation is great, group trapping should not be considered. The correct arrangement is shown in Fig. 93.

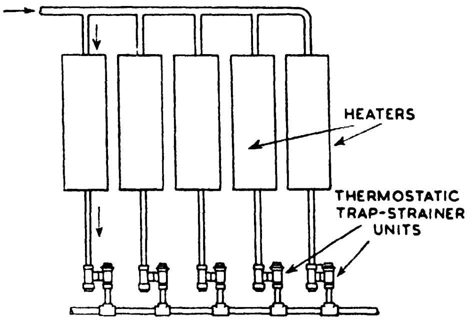


FIG. 93. SMALL DIVIDED GROUP HEATERS WITH STRAINERS AND BALANCED PRESSURE TRAPS
(*Spirax Manufacturing Co., Cheltenham*)

It is evident that each member of the group forming the complete element must be treated separately, for, assuming that the air for drying is blown in a direction at right angles to the length of the headers, the condensation rate must differ from header to header. The rate will be highest in the tube group receiving the initial impact of the cold air, and lessen progressively as each successive group is passed. Under these conditions it is not possible that group trapping should be successful.

Fortunately, because of the small danger of water hammer and steam locking, a comparatively inexpensive trap may be used. Each header should be fitted with a balanced pressure thermostatic trap with near-to-steam element, and preceded by a strainer. Most of the air will be vented by the trap when steam is turned on, and as formed during working, and with short headers it may not be necessary to provide further venting. Where the headers are long, however, an additional trap of the same type should be fitted as an air vent as described above, and at the other end of the header. If this is not done there may be some danger of air pockets forming at the bends of the tubes remote from the steam trap.

Trap Sizing

Before concluding this chapter, it is necessary to touch very briefly on the subject of trap sizing.

It is far too frequently found in practice that the trap size has been selected, not after due consideration of the factors involved but simply in order to fit without alteration, the outlet branch of the machine. The trap size chosen on this basis may possibly be right, but when it is considered that traps offered by various makers vary very greatly to start with, and that outputs from individual traps depend entirely upon the particular conditions under which they are expected to work, it is plain that such an arbitrary method of selection is far more likely to be wrong.

In determining the correct size of trap for any particular duty, the first essential is a knowledge of the steam consumption of the appliance both under working conditions and at starting. Some indication of the great difference there may be between the two was given in the last chapter dealing with pipework, and various methods of making reasonable assessments, in the absence of steam meters or other means of actual measurement, are given in a later section.

Next, the probable pressure across the trap while working must be determined, and this entails knowledge of both the pressure at the *inlet to the trap*, and the back pressure upon it. The former figure may not necessarily be the same as the inlet to the appliance to which the trap is fitted, and may vary from it to a greater or less degree during working, but the back pressure is a more stable factor, depending upon whether or not the trap is required to lift its condensate, and the height of the lift. Theoretically also, the friction in the condense pipes should be taken into account, but for all practical purposes it is sufficient to reckon 1 lb of back pressure for every 2 ft of lift.

When ordering traps for any specific purpose it is therefore necessary to furnish the manufacturer with certain definite information as regards—

1. The maximum and average loads upon the trap.
2. The pressure of the supply. If this is superheated and any of the superheat is likely to arrive at the trap, as in the case of steam main drainage, the total temperature of the steam or degree of superheat should be stated.
3. If the trap is required to lift, the height of the lift must be stated.
4. A brief description of the apparatus to be drained is always helpful.

Never use haphazard methods either in the selection or installation of traps. It should always be remembered that traps in general are well thought out and carefully designed for specific conditions, and only when some approach to those conditions is attained in practice can they be expected to give complete satisfaction. Design and workmanship are the responsibility of the maker, but the onus of installation, which in the case of traps is unfortunately of almost equal importance, lies with the user. Too much emphasis cannot be placed on this point.

The makers are ready to help at all times. After all, it is the reputation of their product which is at stake, and in cases of difficulty their expert advice should be sought at once.

CHAPTER VII

STEAM ESTIMATION AND PLANNING

In previous chapters reference has been made to the necessity of ordered planning in connection with steam generating and consuming apparatus. Too often, especially in older factories, it is found that the plant just grows, and that extensions are added to cope with the needs of the moment, without due thought being given to the possible effect of the new addition on the plant as a whole. Much of this is inevitable, for the engineer must at all costs, sometimes very hurriedly, satisfy the needs of the production people, who are not primarily concerned with the engineering side.

Departmental Survey

Sooner or later a state of affairs is reached when the continued employment of expedients itself produces problems of such magnitude that a complete survey of the whole of the factory arrangements becomes imperative.

Logically such a survey should be departmental in form, but however it be carried out, the result should furnish the fullest information. It should be possible to visualize a complete and accurate picture of the factory steam requirements, starting from the boiler house, and showing the maximum and average demands for each department.

Only in this way is it possible to obtain all the data required by the steam engineer, for pipe sizing, route planning, trap selection, and the like. Reference has already been made in Chapter I to the methods of graphically illustrating these factors by superimposing graphs, whose construction involves not only magnitude but also incidence of flow.

In a long experience of steam estimating for widely differing types of factory, the author would say without hesitation that this is the only satisfactory method of assessing the true conditions, and that the time and care necessary to construct the graphs is well repaid. That the method is tedious is unfortunately true, for in the early stages it may be necessary to study the behaviour of individual machines throughout the 24 hours.

It is obvious that the methods to be used may well be as varied as the appliances themselves, and their name is legion, but it is also fortunately true that there are certain basic principles which, with slight variations in the methods of application, may be used for most forms of commercial apparatus.

Apart from the production of power, it may be said broadly that steam in industry is used for two main purposes: space heating and process work. In all forms of space heating, whatever particular method is used, the principle is the same. The steam may be put through plain suspended pipes, radiators, or unit heaters through which air is blown by a fan, or again it may be supplied to a calorifier whose secondary circuit gives heat up to the surroundings through the medium of hot water. But these differences are only superficial, and the basically important point is that whatever the appliance, heat is eventually given up to the atmosphere, and if this heat can be calculated the steam required to produce it can be assessed.

It may be said at once that the most direct method of estimating the steam consumption of any heat absorbing appliance is to collect the condensate at an appropriate point in the steam cycle, that is to say, at the discharge from the traps. Simple as it may seem, the method has its pitfalls, as will be seen later. There is also the possibility that the disconnecting of the particular trap may cause such dislocation of some other part of the system that it becomes impracticable. In such circumstances some method of calculation must be resorted to, and the estimation made by the use of formulae involving radiation and convection.

A very common form of heating appliance is the simple system of overhead pipes. Most usually these are 2 in. bore, the size being easy to handle, inexpensive, and providing adequate heating surface per unit length.

Some mention of the method to be used was given in Chapter V, but it is repeated here in more detail, because the principles are fundamental and may be applied wherever heat is given up from a heated surface to its surroundings.

Calculation of Heat Losses

Table IV in Chapter V gives the constants which may be used for both radiation and convection, and the calculation which follows is similar to the previous example, except that the pipe surfaces are, of course, unlagged.

EXAMPLE.—Assume that the system consists of 700 ft of 2 in. piping, suspended horizontally in the building at eave level. It is supplied with steam at 5 lb per sq in. gauge, is adequately trapped, and thermostatically controlled to maintain a temperature of 62° F in the building under all conditions. The pipe surface is ample for the required duty.

When the pipes have attained their full working temperature the surface may be said to be maintained at the same temperature as the steam within. This is not strictly true but sufficiently near for the present purpose. With the steam at 5 lb per sq in. the surface temperature is therefore 228° F, and as that of the building is 62° F the difference (which is maintained by thermostatic control) is 166° F.

Dealing first with convection, and using the formula in Chapter V—

$$H = C(t_1 - t_2)^{1\cdot25}$$

From Table IV, page 107, the constant C for 2 in. pipe is 0·45, so that

$$\begin{aligned} H &= 0\cdot45(166)^{1\cdot25} \\ &= 0\cdot45 \times 600 \\ &= 270 \text{ B.Th.U. per sq ft per hour.} \end{aligned}$$

This is the heat loss from the surface of the pipe to the surrounding atmosphere, by convection only.

Now for radiation. Here, from the previously stated Stefan-Boltzman law, the calculation is made, using as terms an emissivity figure given in the tables, and the *absolute* temperatures of the surfaces and the surroundings.

Then $$R = 0\cdot173 \times E\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right]$$

The constant E may be taken as 0·95 (a dull painted surface)

so that $$R = 0\cdot173 \times 0\cdot95\left[\left(\frac{688}{100}\right)^4 - \left(\frac{522}{100}\right)^4\right]$$

$$\begin{aligned} R &= 0\cdot164 \times (2300 - 680) \\ &= 266 \text{ B.Th.U. per sq ft per hour approx.} \end{aligned}$$

Then the heat emitted from the pipe surface (both radiation and convection) is 536 B.Th.U. per sq ft per hour.

A 1 ft run of 2 in. steam pipe has a surface area of 0·334 sq ft so that the total length of 700 ft has an area of 700 × 0·334 = 233 sq ft.

The heat given up to the atmosphere by the whole system is therefore 233 × 536 = 124,500 B.Th.U. per hour.

Steam at 5 lb per sq in. has a latent heat of 960 B.Th.U. per lb, so that the steam consumption per hour will be

$$\frac{124{,}500}{960} = \underline{130 \text{ lb per hour.}}$$

Actually, some of the sensible heat of the condensate may be given up as well if the correct type of trap has been fitted, and, for a rough practical calculation it is often considered sufficiently accurate to divide the B.Th.U. derived from the two formulae by 1000, in order to obtain the steam consumption.

It is to be clearly understood that the figures obtained by the above method are true for steady conditions only. Means for calculating the steam conditions at starting, which involve the heat used in raising the temperature of the metal, and in which the actual weight of the pipes is a necessary factor, are fully explained in Chapter V. Such a system as is visualized here would, however, be turned on and off at very infrequent intervals, possibly only seasonally, so that the peak load which occurs on starting up may, for the present purpose, be virtually ignored. In any case, the duration of the peak is so short that it only becomes significant when selecting the steam traps, which *must* clear the initial condensate quickly.

Heating Systems

Radiators are still a common means of comfort heating, especially in offices or small industrial buildings. They may be supplied with either steam or hot water, and examples of both methods are found with almost equal frequency. The steam system, in which the steam is actually supplied to the radiators themselves, has the advantage that it is more flexible and controllable than the hot water method, and the effect of turning individual units off or on is more quickly felt. The pipework associated with this system is usually much smaller, and so forms a less proportion of the total heating surface. There is, however, the disadvantage that surface temperatures are higher, so high that, unless the units are suitably protected, there may even be some danger of burns.

The hot water system, worked from a calorifier to which the steam is supplied, overcomes the disadvantage of high surface temperatures to some extent, but it is altogether more cumbersome and difficult to control. The fact that comparatively large water volumes must lose or gain heat before an effect is felt, in itself militates against controllability, and the further fact that adjustments at any one point are apt to affect the whole system is by no means an advantage. Moreover, the considerable proportion of the total heating surface taken up by the large diameter circulating pipes (which must function the whole time the system is working) limits the range of individual control which can be applied.

Whether the radiators are fed by hot water or directly supplied with steam the principles governing the assessment of steam consumption are similar.

Most manufacturers of radiators issue tables of constants referring to their products, and these are helpful when available. As a rule, they take the form of a figure for B.Th.U. per sq ft per hour per degree F of temperature difference between the hot surface and the surrounding air. This most convenient method may be used for either hot water or steam, with sufficient accuracy for practical purposes. Typical constants are given in Table V.

TABLE V

AVERAGE RADIATOR TRANSMISSIONS

B.Th.U. per sq ft per degree difference per hour

Position of Radiator	2-Col.		4-Col.		6-Col.		Window		Wall		Hospital 3 in. Wide		Hospital 5⅜ in. Wide		Hospital 7⅜ in. Wide	
	Water 160–60	Steam 215–60	Water 160–60	Steam 215–60	Water 160–60	Steam 215–60	Water 160–60	Steam 215–60	Water 160–60	Steam 215–60	Water 160–60	Steam 215–60	Water 160–60	Steam 215–60	Water 160–60	Steam 215–60
Radiator fixed 2⅜ in. from wall	1·85	2·10	1·70	1·95	1·60	1·82	1·58	1·80	1·71	1·95	1·85	2·10	1·58	1·80	1·51	1·72
Fitted with Crane deflecting Shield*	—	—	1·75	2·00	1·65	1·90	1·65	1·89								
In open recess not less than 3 in. clear of radiator top and sides	1·70	1·95	1·55	1·78	1·47	1·68	1·45	1·67	1·55	1·78	1·70	1·95	1·45	1·67	1·40	16·0

*Representing equivalent heating effect at lower levels of room

Sometimes, for radiators of unusual design, the figures for total output are given. When this is done it is usual to provide three alternative values—for steam, gravity flow hot water, and accelerated hot water. The advantage of the first method, which takes into account the actual temperature difference, is first that only one set of constants is required for all ordinary conditions, and second, the calculation, although a little more tedious to carry out, is likely to result in a more accurate

TRANSMISSION TABLE
(For radiators fixed 2⅜ in. from wall)
B.Th.U. per sq ft per hour

Radiator	Temperature Difference (degrees F) Water						Steam	
	70	80	90	100	110	120	155	160
2-Col. Pall Mall	116	140	162	185	208	233	327	340
4-Col. ,, ,,	105	128	150	170	192	215	300	312
6-Col. ,, ,,	100	120	140	160	180	200	282	294
Window	99	119	138	158	178	200	278	290
Wall	106	128	150	171	192	215	300	312
3 in. Hospital	116	139	162	185	208	234	327	340
5⅜ in. ,,	99	119	138	158	178	199	278	290
7½ in. ,,	94	113	132	151	170	190	266	275

HANDY RADIATOR TRANSMISSION TABLE
(For radiators fixed 2⅜ in. from wall)

	Hot Water Gravity Circulation (mean rad. temp. assumed 155° F)			Hot Water Accelerated Circulation (mean rad. temp. assumed 170° F)			Steam L.P. or Vapour Systems (rad. temp. assumed 215° F)		
Room Temperature	55° F	60° F	65° F	55° F	60° F	65° F	55° F	60° F	65° F
2-Col. Pall Mall	185	174	162	215	208	194	340	327	315
4-Col. ,, ,,	170	159	150	196	192	179	312	300	290
6-Col. ,, ,,	160	150	140	185	180	168	294	282	270
Wall	171	159	150	196	192	179	290	300	290
Window	158	148	138	183	178	166	312	278	270
3 in. Hospital	185	173	162	215	208	194	340	327	315
5⅜ in. ,,	158	148	138	183	178	167	290	278	270
7½ in. ,,	151	142	132	175	170	159	275	266	253

The above transmissions are based on 30 in. high (except window) radiators, transmissions increasing for lower radiators and decreasing for higher.

It is recommended that radiators be fixed as far from the wall as the situation will allow, in order to obtain the maximum heating effect. 2½ in. is suitable if space permits.

Radiators painted with metallic bronzing paints such as aluminium have less efficiency than radiators painted with flat paints or with enamels.

assessment. The constants are calculated for usual radiator positions, that is to say 2⅜ in. from the wall, and the influence of the wall upon the convection currents is taken into account.

In estimating the total steam consumption of a system it is obviously necessary to take into account the supply pipes to the radiators. Where hot water is the heating medium these are of special importance for the reasons already given. For small schemes the heat loss figures given in Table VI may

TABLE VI

TRANSMISSION TABLE (WROUGHT IRON PIPES) UNCOVERED

B.Th.U. per lineal ft per hour

Size of Pipe	Temperature Difference (degrees F)									
	Water							Steam		
	80	85	90	95	100	105	110	150	155	160
½ in.	42	45	48	51	56	59	63	95	98	103
¾ in.	53	56	60	64	70	75	80	118	123	130
1 in.	60	64	70	74	80	85	90	135	140	147
1¼ in.	73	78	85	90	98	103	110	168	175	180
1½ in.	82	87	96	102	110	116	124	185	195	202
2 in.	97	103	113	120	130	137	145	220	228	238
2½ in.	115	122	135	142	155	165	175	260	272	283
3 in.	140	150	165	175	188	200	212	320	330	345
4 in.	174	185	203	215	232	245	262	395	410	426
5 in.	212	225	247	265	282	300	318	480	500	522
6 in.	250	270	290	310	330	350	375	560	585	610

The above is for single pipes freely exposed. For three or more pipes installed one above the other in the form of a pipe coil, reduce by 10 per cent for pipes under 1¼ in. size, and by 20 per cent for pipes 1¼ in. and over.

Covered Pipes

The transmission from covered pipes varies according to the efficiency and thickness of the non-conducting material employed. For pipes covered with ½ in. hair felt allow one-half, 1 in. asbestos one-third, and 1½ in. magnesia one-quarter of the above values.

be used as a rapid means of estimation, but for large and important installations it is better to use the methods demonstrated for steam pipe overhead heating.

Where there are considerable runs of vertical pipes an allowance must be made, because the heat loss from the upper lengths is very much less than that at the lower by reason of the warm convection currents rising along the whole length of the pipe. Much will depend of course on the actual situation of the vertical portion, and the prevalence or otherwise of cool air currents in its vicinity. In practice, a reasonable allowance may be made by considering some portion of the pipe to be ineffective. Some judgment is necessary, but usually the allowance may range from as low as one-tenth to as high as one-fifth of the vertical length. Where the whole length is exposed to definite draughts no allowance need be made, while on the other hand a vertical pipe in still air, not at any time exposed to external air movement, would need the full allowance of one-fifth.

Where the accelerated hot water system is used, it is possible to make an estimate of the heat loss throughout the whole of the system, from considerations of temperature drop from end to end, and quantity of water flowing. From the steam engineer's standpoint this may well be the best method. It is only necessary to take temperatures at the return and flow branches and calculate the B.Th.U. from the water actually passing through the pump. The latter is not always easy to obtain, for, except in the very largest systems, heating installations are seldom provided with adequate instruments, but it is usually possible to compare the work being done by the motor with the name plate information, and so proportion the stated full-load output with the actual output.

The simplicity of the final calculation is seen from the following—

Temperature of flow . . .	160° F
Temperature of return . . .	130° F
Difference	30° F
Water flowing per hour . . .	600 gal.

Then B.Th.U. given up to atmosphere by the whole system

$$= 30 \times 600 \times 10 = 180{,}000 \text{ B.Th.U. per hour.}$$

If the calorifier is supplied with steam at 5 lb per sq in. (latent heat 960) the steam required is

$$\frac{180{,}000}{960} = \underline{187 \text{ lb per hour.}}$$

Unit Heaters

In large industrial buildings suspended unit heaters are becoming increasingly popular for space heating. They are convenient, easily installed, and have the inestimable advantage from the industrial standpoint, that they take up no floor space whatever.

From the heating engineer's point of view they are most convenient because they may be purchased for standard outputs, so that the heating requirements of a given building may be supplied with certainty. There is so much variation in the details of design that it is not possible to give a method of steam estimation sufficiently flexible to cover all cases. The measurement of the heating surface of existing apparatus is so troublesome as to be hardly practicable. It is better to refer at once to the makers' catalogues, and take the designed output as correct. Provided that the elements are kept clean and reasonably free of dust there is no reason to suppose that the output should fall below that for which the particular heater was originally designed. Heat output will fall with reduction of steam pressure, and will vary with fan speed. This is important to note when referring to makers' data, as it is usual for one size of heater to be rated at several outputs, depending on one or both of the above factors.

Computation of Heat Losses

Where makers' information is not available, a very rough estimate may be made by taking the air temperature immediately in front of the heater, that is to say the hot side, and obtain from this the temperature rise above the general atmosphere in the shop. Air volume must then be measured by anemometer or estimated by fan speed and diameter

The weight of the air can be obtained from Table IX, Chapter VIII, and the specific heat taken as 0·2.

Then

$$\text{Heat emitted} = \text{Air weight} \times \text{temperature rise} \times 0{\cdot}2$$

$$\text{Steam consumption} = \frac{\text{Heat emitted per hour}}{\text{Latent heat of the steam}}$$

The steam estimation of apparatus for process work, as distinct from space heating is, by the very variety of the appliances, a much more involved business, but it is important to realize that the same fundamental principles apply. Steam, or rather heat, is supplied to the machine. Some is usefully absorbed in performing its work on the product, the remainder is rejected in various ways. It is unfortunately true that in by far the majority of cases the rejected heat is many times greater than that which is usefully employed.

Direct and Indirect Heating

The heat is conveyed to the product either direct, as when the steam is blown into a liquid or semi-liquid mass, or by contact of the product with some surface which is heated by the steam. Again, the product may be heated by contact with, or immersion in, some other medium such as air or water which receives heat from the steam. The "lost" heat may be radiated or convected from the exposed surfaces of the appliance not in contact with the product, or rejected from the exhaust, either as sensible heat in the condensate, or blown through an open nozzle or valve.

The formulae for convection and radiation, together with other expedients, may be applied to obtain a summation of the various losses which, added to the heat quantity usefully applied to the product (which can nearly always be easily calculated), gives an estimate of the total heat requirements of the machine. The calculation method, depending as it does upon separate assessments of the various losses, gives information as to the heat distribution within the appliance, which cannot be obtained in any other way. The value of this type of information cannot be overestimated, for knowledge of the extent and character of the various losses frequently points the way to practical methods of reducing them.

Take the case of a directly heated tank. This form of appliance is a very common means of raising the temperature of liquids and semi-liquids. When heated with low pressure steam and fitted with thermostatic control it can be reasonably efficient, but, as usually arranged, with steam pressure of the order of 30 to 60 lb per sq in., and with the control at the mercy of the operator, it is most wasteful. What happens is that steam is allowed to flow at the same rate all through the process, and, since the condensing rate must decrease as the temperature of the product approaches boiling, considerable volumes of practically uncondensed steam bubble straight through to atmosphere.

But for the present purpose the important point is that such steam as is condensed increases the volume of the contents of the tank. Therefore where the volume of the product is not affected by the process, that is to say, it neither shrinks nor swells under the influence of heat, the increase of volume is a measure of the steam absorbed, in a given time. Therefore it is only necessary to measure the rise of the liquid or the overflow, where this is provided, to get a direct figure.

Since the rate of condensing is not constant it is necessary that frequent readings should be taken, the simplest method being a graduated stick immersed in the tank, in a position where the turbulence is least. It is particularly important that during the first 15 minutes readings should be taken at least every 2 minutes. If the tank is not thermostatically controlled, it is safe to say that when the boiling point is reached the steam flow per hour will be at least $\frac{2}{3}$ the average figure for the first 30 minutes. The figures obtained will include the steam condensed in compensating for the heat loss by radiation and convection from the outside of the vessel.

When completed, a graph should be drawn, and, if the investigation is part of a larger survey involving the whole factory, the time of day for the start and completion of the test must be clearly stated upon it. This applies to all classes

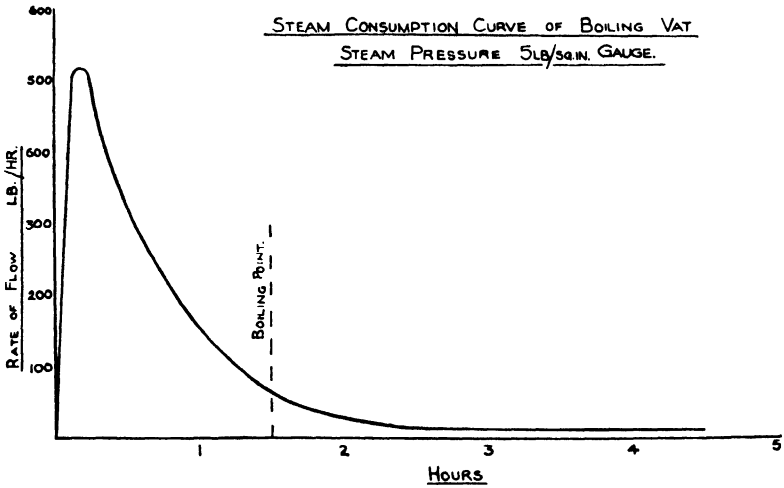


FIG. 94. STEAM FLOW CURVE OF BOILING TANK

of apparatus, but especially is it necessary in those cases where the characteristic flow is known to develop a considerable peak.

Boiling vats which are heated by steam coils or jackets need a somewhat different method. Here, the steam is not condensed within the product, and the only direct measurement which may be possible is the condensate ejected from the trap or traps. Unless very great care is taken, this method may be quite misleading. First, it is necessary to see that the pipes or jackets are well drained before the test commences, and secondly, if the condensate is collected as it should be, as close as possible to the vat, the steam flashed off at the trap must be taken into account. It is no use whatever taking a snap test lasting only a few minutes. Condensate should be measured continuously from the start to the finish of the process, the readings being recorded at intervals which must be as short as possible.

The actual time intervals must depend on the process itself in so far as the rate of condensation is affected, longer periods being permissible as the rate slows down. The curve in Fig. 94 is characteristic and is of interest for purposes of comparison with the result obtained by calculation.

EXAMPLE.—Consider a galvanized tank, 6 ft long, 3 ft wide, and 2 ft deep. The contents are to be brought to the boil in $1\frac{1}{2}$ hours, after which they are maintained at boiling point for 3 hours. The ambient temperature is 70° F, the specific gravity of the product is 1·2, and its specific heat is 0·35. The weight of the tank itself is 5 cwt. Lagging is provided, such that the outside surface temperature, taken by surface pyrometer at various points, attains a maximum of 100° F.

Since the maximum temperature to be attained by the product is 212° F, and this is reached in $1\frac{1}{2}$ hours, we may assume that the heat absorbed in raising the temperature of the tank itself is given up during the heating-up period. After this, there is no further increase of temperature, but heat is transferred through the walls and lagging to the atmosphere.

The heat used to raise the temperature of the tank may therefore be assessed at once, as follows—

Temperature rise . . .	$212 - 70 = 142$ °F
Weight of tank . . .	560 lb
Specific heat of iron . .	0·14

Then $\quad 142 \times 560 \times 0{\cdot}14 = 11{,}100$ B.Th.U.

This takes place in the first $1\frac{1}{2}$ hours. In the same period, the same temperature rise takes place in the product.

The volume of the tank is 36 cu ft, and as the specific gravity of the product is given as 1·2, the weight is

$$36 \times 62{\cdot}2 \times 1{\cdot}2 = 2690 \text{ lb.}$$

The specific heat is 0·35, so we get

$$2690 \times 142 \times 0{\cdot}35 = 133{,}000 \text{ B.Th.U.}$$

which is the heat given up to the product.

But immediately the temperature of the product exceeds that of the surroundings, heat is transferred through the walls of the tank, raising the temperature of the external surfaces, and radiation and convection commence.

For the calculation of these losses the formulae already quoted are used, with the constants (Table IV) appropriate to the particular conditions. When dealing with convection, sides, top, and bottom must be calculated separately, because as seen from the Tables the position of the surface in question affects the constant. Radiation calculations are not influenced in this way, and the whole of the surface may be dealt with in a single computation, employing an emissivity figure whose value depends on a single factor, which is the nature of the surface.

In the case of very large tanks or where a high degree of accuracy is required, it may be necessary to pay special attention to the bottom of the vat. If this is placed directly on the floor there is no convection, but heat is lost by conduction to the supporting surface. On the other hand, where the vat is placed on battens so situated that there is free circulation of air across the unlagged bottom, the convection losses may well be higher than would result from using the constants given in the Tables with the convection formulae. In these circumstances it is quite safe to increase the constant to 0·37 or 0·40.

To return to the example. The average temperature of the surfaces during the heating up period is

$$\frac{70 + 100}{2} = 85^\circ \text{ F.}$$

The lagging is painted so that from the Tables the emissivity figure is 0·95.

The Stefan-Boltzman law is stated in terms of the *absolute* temperature, so that 460 must be added to the temperature of the surfaces, and of the room. Then for radiation,

$$\begin{aligned} R &= 0{\cdot}173 \quad 0{\cdot}95 \left[\left(\frac{545}{100}\right)^4 - \left(\frac{530}{100}\right)^4\right] \\ &= 0{\cdot}164 \times [(5{\cdot}45)^4 - (5{\cdot}30)^4] \\ &= 0{\cdot}164 \times (890 - 800) \\ &= 14{\cdot}75 \text{ B.Th.U. per sq ft per hour.} \end{aligned}$$

From the dimensions given, the surface area, excluding the bottom, is 54 sq ft, so that the heat loss from radiation per hour is 795 B.Th.U. Then the heat loss during the $1\frac{1}{2}$ hours heating up will be 1190 B.Th.U.

If we assume the bottom to be unlagged the average temperature will be

$$\frac{212 + 70}{2} = 141^\circ \text{ F.}$$

Radiation will therefore be

$$R = 0{\cdot}164 \times [(6{\cdot}01)^4 - (5{\cdot}30)^4]$$
$$= 0{\cdot}164 \times (1320 - 800)$$
$$= 85{\cdot}5 \text{ B.Th.U. per sq ft per hour.}$$

The area of the bottom is 18 sq ft, so that the radiation loss per hour is 1540 B.Th.U., and in $1\frac{1}{2}$ hours 2310 B.Th.U.

Now for convection. The general formula is

$$H = C(t_1 - t_2)^{1{\cdot}25}$$

where C is a constant from the Tables appropriate to the position and shape of the surface; t_1 and t_2 are the temperature of the surface, and the surrounding air respectively.

Since the temperature difference is assumed to be the same for all parts of the tank except the bottom, the main portion of the equation may be worked out first of all, and then multiplied by the constant chosen as applicable to the particular surface concerned.

The average temperature of the lagging at the surface is already given as 85° F, and the air temperature as 70° F, so that the difference is 15° F.

Then $$15^{1{\cdot}25} = 30 \text{ approx.}$$

From the Tables, the constant C is given as 0·30 for plane vertical surfaces and 0·39 for plane surfaces facing upward.

Therefore, for the sides and ends of the tank, the loss is

$$0{\cdot}30 \times 30 = 9{\cdot}0 \text{ B.Th.U. per sq ft per hour,}$$

and for the top,

$$0{\cdot}39 \times 30 = 11{\cdot}7 \text{ B.Th.U. per sq ft per hour.}$$

The area of the top is 18 sq ft, so the loss per hour is $11{\cdot}7 \times 18 = 210$ B.Th.U., or, for the $1\frac{1}{2}$ hours, 360 B.Th.U.

The area of the sides and ends is 36 sq ft, so that $36 \times 9 = 324$ B.Th.U.per hour, or, for the $1\frac{1}{2}$ hours, 460 B.Th.U.

For the bottom, which is unlagged, the average temperature is $142 - 70$.

As the tank is assumed to be supported on battens, and convection air currents may be considerable, we may use 0·40 for the constant "C."

Then $$H = 0{\cdot}4 \times (142 - 70)^{1{\cdot}25}$$
$$= 0{\cdot}4 \times 72^{1{\cdot}25}$$
$$= 0{\cdot}4 \times 210 = 84 \text{ B.Th.U. per sq ft per hour.}$$

The area is 18 sq ft,

so $$18 \times 84 = 1520 \text{ B.Th.U. per hour,}$$

or $$2260 \text{ B.Th.U. for the } 1\tfrac{1}{2} \text{ hours.}$$

We are now in a position to compute the steam absorbed during the first period of the process.

Absorbed by the product . . .	133,000	B.Th.U.
Raising temperature of tank . .	111,000	,,
Lost by radiation (excluding bottom) .	1190	,,
Lost by radiation (bottom) . .	2310	,,
Lost by convection (top) . . .	360	,,
Lost by convection (sides and ends) .	460	,,
Lost by convection (bottom) . .	2260	,,

Total B.Th.U. during the $1\frac{1}{2}$ hours heating-up period, 250,580.

If we assume that the steam is supplied at 5 lb per sq in. saturated, the steam required is

$$\frac{250{,}580}{960} = 261 \text{ lb.}$$

The subsequent boiling period is treated in exactly the same way, but it must be remembered that there is no further rise of temperature of the product, or of the tank, but that the steam supplied compensates for the losses through the walls. During this period also, the walls have attained their full temperature, so that the losses will be slightly higher, the temperature differences being 100 − 70, instead of 85 − 70, for the lagged portions, and 212 − 70, instead of 141 − 70, for the unlagged bottom.

Proceeding as before, the figures are—

Radiation for all surfaces except the bottom	1,875	B.Th.U./hour.
Radiation for the bottom . .	3,900	,,
Convection for all surfaces except bottom	1,248	,,
Convection for the bottom . .	3,600	,,
Total	10,623	B.Th.U./hour.

B.Th.U. lost during the boiling period of three hours,

10,623 × 3 = 31,900, and the steam absorbed, 33 lb.

The *rate of flow* during this period is therefore 11 lb per hour.

We now have two average figures for the steam flow, i.e. 261 lb total during the heating-up period, = 174 lb per hour, and 33 lb during the boil = 11 lb per hour.

These figures are sufficiently accurate for a departmental survey, where questions of pipe sizing and boiler loading only may be involved, but it is to be understood that they do not show the magnitude of the maximum peak flow. Because of its very short duration, this may not be important for the purposes mentioned, but, as in the case of the overhead heating system, the peak must be known for purposes of trap selection.

Reference to Fig. 94 shows that the peak is attained in the first five minutes of the heating-up period. Roughly $\frac{1}{6}$ of the total steam absorbed during this period flows in the first 5 minutes, so that in the case in point, $\frac{261}{6} = 43{\cdot}5$ lb flows, and the rate of flow, which is the figure needed for trap selection, is $43{\cdot}5 \times 12 = 525$ lb per hour.

So much for boiling vats and kindred apparatus.

Immersed Heating Surfaces

Immersed heating surfaces, which take many forms in industry, and are used in very varied types of apparatus, heat exchangers, calorifiers, and the like, may be calculated from a basic heat emission figure.

Copper pipes, heated by steam and immersed in water, will transmit 146 B.Th.U. per sq ft per degree F temperature difference per hour.

Where water instead of steam is the heating medium, the figure is 73 B.Th.U.

For iron pipes instead of copper, the figures are 140 and 70 respectively, while for lead pipe, which is sometimes used with very low steam pressures for corrosive processes, the values are 122 and 50.

In the computation of the temperature differences, the mean temperatures, both of the steam or heating water and of the product, must be used. Thus, in a boiling process using immersed pipes, supplied with steam at 5 lb per sq in., the steam will be discharged from the trap at 212° F. and the mean temperature is

$$\frac{228 + 212}{2} = 220° \text{ F.}$$

If the product is to be raised from 60° F to the boiling point, the mean temperature will be

$$\frac{60 + 212}{2} = 136° \text{ F,}$$

and the mean temperature *difference* is 84° F.

If iron pipes are used, the heat transmission will be

$$140 \times 84 = 11{,}800 \text{ B.Th.U. per sq ft per hour}$$

during the heating-up period, and the steam absorbed, an average figure of $\frac{11{,}800}{960}$ which is equal to 12·3 *lb of steam per sq ft of heating surface.*

When the boiling point is reached, however, the temperature difference drops to 8° F, and the heat transmission to 1120 B.Th.U. per sq ft per hour. The steam equivalent is 1·17 lb per sq ft per hour.

This is a useful method for calculating either the steam consumption of existing apparatus, where the area of the heating surface can be ascertained, or the heating surface required to perform a known duty.

Steam calenders and rolls of various types are used in industry for very many processes, as widely divergent as paper making, textile manufacture, milk drying, and the rubber industry. In connection with plastics alone there are many quite different processes within the single industry, each one of which must receive separate and special consideration when the heat requirements are assessed.

In many processes it is only necessary to heat the rolls at the start of the cycle, after which the steam is turned off, and the temperature maintained by some chemical action within the product, or by the friction brought about by the mixing action of the rolls. Sometimes it is even necessary to cool the rolls during the greater part of the working cycle. Quite obviously, a careful study of the working cycle is of the utmost importance if a true picture of the conditions is to be obtained, and, although it is a simple enough matter to estimate the steam demand at any one time by any particular machine, that knowledge alone is of little value.

Here again it is the presence of the peaks which is the vital factor. The prospect of, say, twenty sets of large rolls, each set taking perhaps seven times its normal load during a warming-up period of one and a half hours, and all turned on at once at the start of the day's work, is not a pleasant one for the steam engineer, but it is one which can only be avoided by the co-operation of the production people. This, as is only to be expected, will be forthcoming only if exact data as to the behaviour of their machines are presented as evidence of the desirability of planned loading. It is the business of the steam engineer to supply this evidence.

The actual steam estimation may be performed in three stages.

1. Heating-up stage, involving weight and specific heat of the heated metal, the temperature rise, and the radiation and convection losses during this period.
2. Heat given up to the product, involving temperature rise, weight, and specific heat.
3. Convection and radiation losses from the constantly changing exposed surface of the rolls, and from parts of the machine frame whose temperature has been raised above that of the surroundings by conduction.

Sufficient data have been given in this chapter for all the above calculations to be carried out. For convection, the rolls themselves may be regarded as large pipes, the value of the constant C being 0·35.

For milk drying, and other processes which involve the evaporation of moisture, it is necessary to know something of the moisture content of the product before and after the process. The heat allocated to this is that amount necessary to turn that moisture into steam at atmospheric pressure.

Exactly the same formulae are used, and exactly the same principles are involved.

Platen Presses

Platen presses, either single- or multi-daylight, are used in industry for a multiplicity of purposes, and again the same formulae and the same principles may be applied. As, however, they form a somewhat more complex example of the application of those principles, it is proposed here to work out a definite case.

EXAMPLE.—A multi-daylight press is built up of eight platens, each 40 in. × 40 in., and 3 in. thick. Because the temperature required is critical, it is necessary to supply steam at 75 lb per sq in. gauge. Each platen is suitably trapped and adequately air-vented so that the temperature of the surfaces, when fully warmed up, is 320° F. When fully closed, the edges of the platens may be treated as four vertical surfaces, each 40 in. × 24 in., and amenable to calculation by the usual convection and radiation methods. Thus—

Convection

Constant $C = 0{\cdot}30$. If the temperature of the shop is 70° F, we get

$$
\begin{aligned}
H &= 0{\cdot}30 \times (320 - 70)^{1{\cdot}25} \\
&= 0{\cdot}30 \times 1000 \\
&= \underline{\underline{300 \text{ B.Th.U. per sq ft per hour.}}}
\end{aligned}
$$

Radiation

From Tables IV, emissivity figure is 0·69.

Using the absolute temperatures as before, by adding 460 to the Fahrenheit reading,

$$
\begin{aligned}
R &= 0{\cdot}175 \times 0{\cdot}69 \left[\left(\frac{780}{100}\right)^4 - \left(\frac{530}{100}\right)^4\right] \\
&= 0{\cdot}119 \times (3750 - 780) \\
&= \underline{\underline{353 \text{ B.Th.U. per sq ft per hour.}}}
\end{aligned}
$$

Thus, the total for radiation and convection for the vertical surfaces formed by the fully closed platens is 653 B.Th.U. per sq ft per hour.

Dealing now with the surfaces between the platens of the opened press, conditions are a little different. At the widest point, the platens are not more than a few inches apart, so that the radiation from one heated surface to another, closely adjacent and maintained at the same temperature, may be neglected.

The convection constant for use under these particular circumstances may also be reduced to a value of 0·2.

With this figure, the heat lost from the inter-platen surfaces is 200 B.Th.U. per sq ft per hour.

Totalling all the above,

Inter-platen surfaces, convection only (no radiation) 145 sq ft at 200 B.Th.U.	29,000 B.Th.U./hour.
Edges, convection and radiation 27 sq ft at 653 B.Th.U. . . .	17,650 ,,
Total	46,650 B.Th.U./hour.

If the press head is of large area, and unprotected by insulating covering, it is necessary to allow for the heat lost from this and other exposed areas heated by conduction. In order to avoid undue complication it is a good plan to take a number of spot readings with the surface pyrometer (a suitable type of instrument is shown in Fig. 95), and average these out. The points should be carefully selected to take in as wide a range as possible. It must be remembered that the surface area of the whole press structure may be greater than the working surface of the platens, and although the average temperatures are much lower, the total heat losses from this source are very far from negligible.

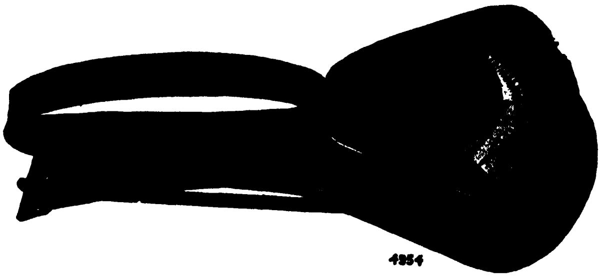


FIG. 95. SURFACE PYROMETER
(*Cambridge Instrument Co.*)

For the press in question, let it be assumed that the surface area of the head and other parts of the heated structure is 200 sq ft, and that the average temperature is 120° F.

Using suitable constants,

Convection . . .	47 B.Th.U. per sq ft per hour
Radiation . . .	39 ,, ,, ,,
Total . . .	86 B.Th.U. per sq ft per hour.

Then the heat lost from the surfaces heated by conduction is

$$86 \times 200 = 17{,}200 \text{ B.Th.U. per hour.}$$

Add this to the heat loss for the platens, and we get

$$63{,}850 \text{ B.Th.U. per hour.}$$

The latent heat of the steam at the required pressure (75 lb per sq in.) is 893 B.Th.U. per lb, so that the steam equivalent of the losses so far is $\frac{63{,}850}{893} = 72$ lb per hour, and this is *the no-load steam consumption of the press.*

In order to complete the calculation it is now only necessary to know the weight of the product per cycle, the average specific heat of the product throughout the cycle, and the number of complete cycles per hour. For this purpose, let it be assumed that the press is dealing with a synthetic product, whose specific heat is 0·35. The weight per cycle is 35 lb, and the cure time is 15 minutes, so that there are 4 cycles per hour. The product is inserted in the press at room temperature, and there is a drop of 15° F between the platens and the mould. The maximum temperature attained by the product is therefore 305° F.

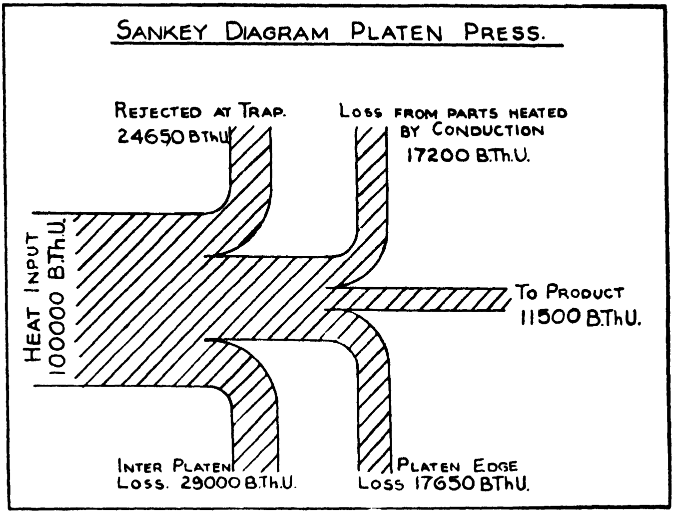


FIG. 96. SANKEY DIAGRAM OF PLATEN PRESS

Then the heat taken up by the product is

$$4 \times 35 \times 0{\cdot}35(305 - 70)$$
$$= 11{,}500 \text{ B.Th.U. per hour.}$$

and the steam equivalent is 13 lb per hour.

The total steam demand is therefore 85 lb per hour, only 13 lb of which are given up to the product. The Sankey diagram of the press is given in Fig. 96, and the steam consumption curve in Fig. 97.

It should be noted that here is a particularly clear example of the value of the calculation method of steam assessment. The separation of the losses made possible by this means shows clearly and unmistakably just where those losses lie. Moreover, since the magnitude of the heat loss is made available in a form which is easily translatable into terms of monetary value, the savings possible by simple forms of heat insulation *at the points where the losses have been shown to be greatest* may be expected to be the next subject for investigation.

Cooled Presses

There is another type of press, usually single-daylight, which demands special and entirely different treatment. In this case, the product must be heated during the first part of the cycle, that is to say, in the period immediately prior to,

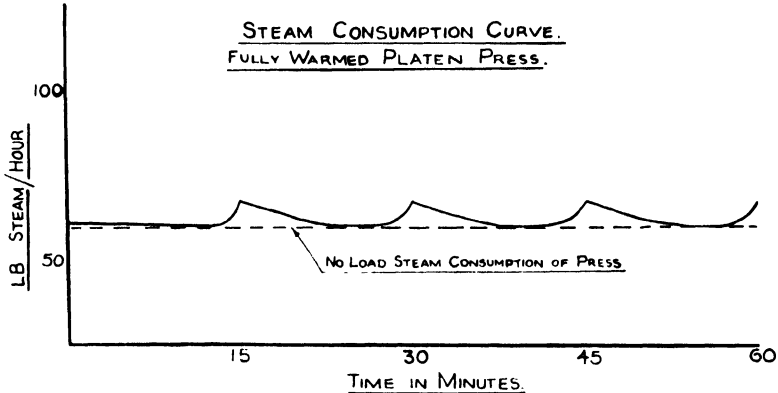


FIG. 97. STEAM CONSUMPTION CURVE OF PLATEN PRESS

and during the descent of, the top tool, after which, when the press is completely closed, rapid cooling is necessary.

From considerations of production it is usually necessary that the alternate heating and cooling shall be done as quickly as possible, so as to shorten the cycle time as far as practicable.

The press is not a platen press in the true sense, although the top and bottom tools are mounted on substantial flat surfaces. The tools themselves are of cored construction, hollowed out to form steam spaces. During the cooling part of the operation a considerable body of cold water is forced into the steam space, usually following exactly the same path as the heating steam, and making use of the same exhaust passages. Herein lies one of the major difficulties from the heat recovery standpoint. Although theoretically a trap fitted to the exhaust should open wide when flooded with cold water, no trap will both stand up to the shock conditions and clear the large water volume with sufficient rapidity. Systems are in use whereby the trap may be isolated during the cooling operation, but these are not favoured by the production people (however desirable they may be for heat conservation) because it is usually necessary for the operator to perform two additional movements in each cycle.

In some cases the design of the tools themselves may militate against the use of traps, it being frequently necessary to use a sectional construction, whose joints are difficult to make steam-tight. No packing can be used, so that reliance must be placed on a dead accurate machined face—by no means a reliable form of joint in the particular circumstances for pressures of more than a pound or two above atmosphere.

For all these reasons it is far more usual to find these presses fitted, not with traps but with a steam and water outlet of about $\frac{1}{4}$ in. bore. Through this the steam escapes full bore to atmosphere, or to a common exhaust main during the heating part of the cycle, the water being ejected through the same orifice during cooling.

For these conditions it is not possible to calculate the steam consumption with great accuracy, without the expenditure of much more time than is usually at the disposal of the steam engineer in a busy industrial concern. A method is given here, however, which, though academically not correct, enables a very fair estimate to be made.

Let us take the case of a hypothetical press, in which both the top and bottom tools are heated with steam at 20 lb per sq in. gauge. The steam exhausts through an open-ended pipe of $\frac{1}{4}$ in. bore.

For the sake of simplicity, it may be assumed that the tools are well insulated from the main structure of the press, so that the heat lost by conduction may be neglected. If desired, of course, it is perfectly simple to calculate this loss by the methods detailed for the previous example.

The weight of both tools together is 180 lb, and 8 lb of a product whose specific heat is 0·2 is put into the press at each cycle. The cycle consists of a heating period of three minutes' duration, which occurs twelve times in the hour, and, after each heating period the tools are cooled by flushing the steam spaces with cold water.

Since the tools are cool when the press is opened it follows that radiation and convection may be neglected, so that there is no loss at this point. But at the beginning of each cycle the mass of metal comprising the tools must be heated up to the desired temperature.

The quantities to be calculated are therefore—

1. The steam equivalent of the heat absorbed by the product.

2. The steam equivalent of the heat required to raise the temperature of the tools to the desired point, say in this instance, 180° F.

3. The steam exhausted through the open pipe to atmosphere.

Dealing with these items in their order,

Heat given up to product (temperature of shop, 70° F)

$$8 \times (180 - 70) \times 0{\cdot}2 = 176 \text{ B.Th.U. per cycle}$$
$$= 2{\cdot}25 \text{ lb steam per hour.}$$

During each cycle 180 lb of tool metal, whose specific heat is 0·14, must be raised to 180° F.

Then

$$180 \times (180 - 70) \times 0{\cdot}14 = 2770 \text{ B.Th.U. per cycle}$$
$$= 35{\cdot}4 \text{ lb steam per hour.}$$

Now to calculate the steam emitted from the exhaust. If the pipe end be considered as a nozzle, it may be said that the pressure of the steam approaching the "throat" is about 0·58 of the initial pressure. This is true only when the ratio of the final pressure to the initial pressure is less than that quantity. For ratios greater than 0·58 the throat pressure may be taken as the same as the final pressure.

Using absolute pressures throughout the calculation,

$$35 \times 0{\cdot}58 = 20{\cdot}4 \text{ lb per sq in.} = \text{throat pressure.}$$

Inspection of the total heat/entropy diagram gives—

Entropy at 35 lb absolute, saturated .	1·69	
Total heat	1165	B.Th.U.
For steam at 20·4 lb absolute, and the same entropy, the total heat is .	1125	,,
Difference	40	,,

Referring to the velocity scale, a drop of 40 B.Th.U. gives a velocity of 1400 ft per sec.

Now refer to the total heat/pressure diagram.

For steam at 20·4 lb absolute, and 1125 B.Th.U. total heat, the specific volume is 19 cubic ft per lb.

The area of the outlet is 0·049 sq in., so that

$$\frac{0{\cdot}049 \times 1400 \times 60}{144 \times 19} = 1{\cdot}5 \text{ lb per minute}$$

and for twelve three minute cycles the steam required is 54 lb per hour.

The weakness of the calculation lies in the assumption that adiabatic expansion takes place through the steam spaces. This, of course, cannot be so, because the very nature of the machine causes the steam to give up some of its heat during its passage, but we have to some extent compensated for this by calculating this heat separately.

In practice, by adding the three figures together, a very close approximation to the actual results is obtained.

Thus the final figures are—

Steam given up to product . . .	2·25 lb per hour.
Steam given up to tools . . .	35·4 ,, ,,
Steam rejected at exhaust . . .	54·0 ,, ,,
Total	91·65 lb per hour.

Here again is an example of the advantages of detailed analysis as compared with more direct methods of steam assessment. The inefficiency of the machine is shown up at once, and an indication given of the point of greatest loss. As it stands, the efficiency of the press is of the order of 2·45 per cent. Of the rejected heat, the exhaust loss represents roughly $54 \times 1125 = 60{,}700$ B.Th.U.

If an efficient trapping system were fitted, this huge proportionate loss could be reduced to 37·65 (the amount of steam required to raise the temperature of the product and the tools) $\times$ 180 — (70 — 32), assuming the condensate to be discharged from the traps at 180° F, which is equal to 5375 B.Th.U. per hour, or the equivalent of 4·8 lb of steam per hour.

Thus, with one alteration, the efficiency (thermal) is brought up from 2·45 per cent to 5·5 per cent.

Vulcanizing Pans

Another class of apparatus which requires special methods is that which comprises the large vessels used for open vulcanizing. Here, the steam is injected in direct contact with the product, which is placed in a large heavily-built vessel, equipped with a steam-tight door. The vessels (vulcanizing pans) are usually fitted with a steam trap, which takes care of the condensate formed during the warming-up and cure periods, and a separate blow-off valve, for releasing the steam pressure within the vessel at the completion of the cure.

The pans form a very serious load. The peaks are high and sudden, and it is usually not possible to recover the condensate from the blow-off, which is contaminated by contact with the product. Sometimes it may be possible to recover some of the heat by means of simple heat exchangers, by which the hot contaminated blow-off is piped through elements immersed in water to be heated for various other process purposes, or even for preheating feed water. The condensate which has actually been in contact with the product is diverted to drain.

The process is usually somewhat as follows—

The goods, placed on suitable wheeled racks, are placed within the vessel, the blow-off valve is closed, and the steam turned on. It is common practice to supply the steam through a reducing valve set to a predetermined pressure.

In many processes a limit is set to the time taken to reach the cure pressure. When the desired pressure is attained it must be maintained for a definite period, which is dictated by the particular character of the goods and the steam pressure.

In the rubber industry much experimental work has been carried out, and some latitude is allowable, but there is always a relationship between the cure time and the pressure at which it is carried out. During the cure, the steam trap is working normally, dealing first with the initial condensation while warming up, and with the radiation and convection losses both during warming and when full temperature has been attained. On the completion of the cure, the steam supply is turned off, and the blow-off valve opened until the pressure is completely released.

It is convenient to assess the steam absorbed by the complete cycle of operations in three parts.

1. Condense ejected from the trap during the warming up. This includes the initial condensation due to heat taken up in raising the temperature of the vessel itself, plus the heat absorbed by the product, and the radiation and convection from the outside.

2. Condensate discharged from the trap during the cure. This includes the condense resulting from external heat losses, and further heat which may be absorbed by the product.

3. Condensate discharged from the blow-off on the

completion of the cure, which is the volume of the steam remaining in the vessel after the supply is cut off.

The whole process of vulcanizing does not lend itself to easy calculation, and the difficulty of devising a general method with wide application is further increased by the fact that some of the vessels are provided with steam jackets in order to decrease the initial condensation.′

Undoubtedly the best method of dealing with the problem is to collect the condensate, and if this is done with care a sufficiently accurate picture of the whole cycle may be obtained. A description of an actual test is given.

The pan, a large open vulcanizer, unjacketed but lagged externally, was of the following dimensions—

Length . . .	12 ft 9 in.
Diameter . . .	4 ft 6 in.
Volume . . .	203 cu ft.

In order that production should not be interfered with, it was necessary to take the various parts of the tests in a rather different order from that given above, but there was no break between the parts.

The steam pressure during the cure was 40 lb per sq in.

The pan was loaded with a normal batch of goods, and the process proceeded with. In this particular instance it was necessary to reach the full pressure of 40 lb in 15 minutes. Condensate was not collected during this first warming up because the peak load from cold was already known, and the particular information required concerned the behaviour of the plant under normal working conditions, that is to say, with the pan retaining some heat from the previous cure. When the cure was completed the steam was shut off, the blow-off valve opened, and the steam content of the vessel allowed to bubble into a calibrated drum, part-filled with cold water.

Amount collected . 48·6 lb.

But taking dry saturated steam as a basis, the weight of steam corresponding to the volume of the pan is 26 lb, so that the dryness fraction of the steam after cure must have been some figure less than 60 per cent.

During the next warming-up period condensate was collected from the trap.

Amount collected . . 105 lb in 15 minutes.

When full pressure was attained, condensate was collected from the trap, and during the four hours' cure, readings were taken every half hour.

The average of these readings was 27·5 lb.

If now these figures are stated in terms of rate of flow, we can, by plotting them on a time base, form a very clear idea of the variations of flow throughout the whole process.

Warming Up

It may be assumed that the whole of the steam filling the body of the vessel flows during the warming-up period, that is to say, during the first fifteen minutes. But in addition to the steam filling the body, some is condensed in raising the temperature of the vessel, and is collected from the trap.

The total flow is therefore,

	48·6 lb	collected from blow-off after cure.
	105·0 lb	collected from trap during warming.
Total	153·6 lb.	

Since the full pressure must be attained in 15 min, it may be said that the *flow rate* is $153 \cdot 6 \times 4 = 614 \cdot 4$ *lb per hour*, for a duration of fifteen minutes. There at once is an indication of the peak flow, remembering that the figure given is *the average flow rate over the peak period.* A still more accurate perception is gained by the assumption (from experience) that $\frac{1}{6}$ of the total flow takes place in the first $\frac{1}{10}$ of the period, so that 25·6 lb flow in the first $1\frac{1}{2}$ minutes.

The flow *rate* at the height of this short peak is therefore 1024 lb per hour.

During the cure the average condensate collected was 27·5 lb every half hour, so that the rate of flow is 55 lb per hour.

The resulting graph constructed from this test is given in Fig. 98.

The Table on page 182 gives useful figures of the allowances which must be made for flash steam, whenever condensate is collected at steam temperature from a trap into an open vessel.

The foregoing chapter claims nothing more than to give examples of a very few types of apparatus. Quite obviously there are many others, of which not a few demand special treatment. The analysis of heat losses alone could form the subject of a special work, and it is therefore impossible to treat it here with anything like the completeness it merits. The examples given are, however, in common use in industry, and are of the type most likely to come within the experience of the works engineer. Also, the methods shown are fundamental, and can be applied with suitable modifications to a much wider

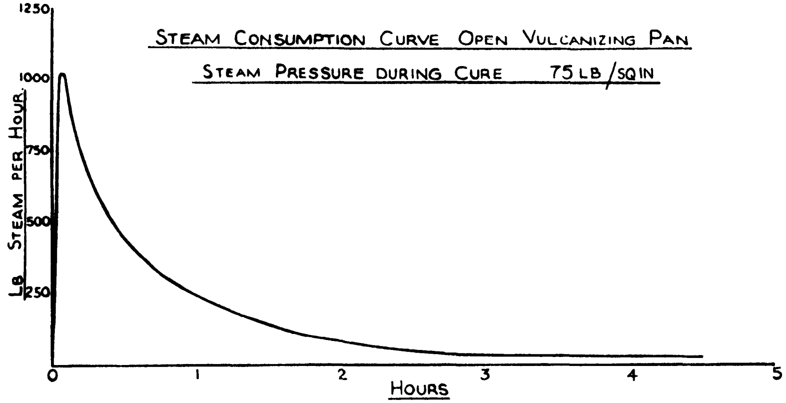


Fig. 98. Steam Consumption Curve of Vulcanizing Pan

range of plant than is actually described. There are undoubtedly simpler methods of assessing the overall steam quantities required for particular processes, but the tracking down and identification of the various losses can only be done by some such methods as are here given.

TABLE VII

Allowances to be Made when Steam is Flashed Off from Various Condensate Pressures to Lower Pressures

Condensate Pressure lb sq in. gauge	Percentage Condensate Flashed off when Pressure is Reduced to			
	40 lb/sq in.	20 lb/sq in.	10 lb/sq in.	Atmospheric pressure
200	11·5	14·3	16·2	18·8
150	9·0	11·8	13·0	16·4
100	5·8	8·6	10·6	13·3
80	4·2	7·1	9·1	11·9
60	2·3	5·2	7·3	10·0
40	—	3·0	5·0	7·8
20	—	—	2·1	5·0
10	—	—	...	2·9

Heat Losses from Buildings

Although the business of calculating the heat losses from a building properly belongs to the province of the heating engineer and architect, it very frequently happens that the works engineer is required to go into the question of space heating a new shop or building, or to provide some special condition of temperature, without which certain work cannot be carried out. A specimen calculation is therefore given here, using the constants set out on page 185.

EXAMPLE.—The building is 80 ft long by 70 ft wide, and 15 ft high to the eaves. The total height to the roof ridge is 25 ft. The wall construction is $4\frac{1}{2}$ in. brick, with suitable buttresses, and $\frac{1}{3}$ of the wall area is glass. At each end is a large door, 7 ft high by 12 ft wide. The roof is unlined corrugated asbestos cement sheeting, and $\frac{1}{5}$ of the roof area is glass. The building is erected on a 6 in. concrete floor, over earth. Ventilating schemes within the building are such that there are three complete air changes per hour.

It is required to maintain 65° F within the building, with an outside temperature of 30° F.

Volume of building to eaves

$70 \times 80 \times 15 = 84{,}000$ cu ft

Volume above eaves . . . . . 27,000 ,,

Total volume . . . . . . 111,000 ,,

Volume of air to be heated per hour

$111{,}000 \times 3 = 333{,}000$,,

Required temperature rise . . . $65 - 30 = 35°$ F.

One cu ft of air raised through 1 degree F requires 0·019 B.Th.U.

Then the heat required to maintain the air temperature is

$$333{,}000 \times 35 \times 0{\cdot}019 = \underline{\underline{224{,}000 \text{ B.Th.U. per hour.}}}$$

Brick areas of side walls

$$80 \text{ ft} \times 15 \text{ ft} \times 2 \times \frac{2}{3} = 1600 \text{ sq ft}$$

Brick area of each end =

$$\left[\left(70 \times 15\right) + \left(\frac{10 \times 70}{2}\right)\right] - (12' \times 7') \text{ sq ft}$$

$$= 1750 - 84 \text{ sq ft}$$

$$= 1666 \text{ sq ft.}$$

For both ends 3332 sq ft less $\frac{1}{3}$ for glass	= 2220	sq ft
Area of glass in sides	800	,,
Area of glass in ends	700	,,
Area of floor (concrete)	5600	,,
Area of roof (asbestos) 47 ft × 80 ft × 2 × $\frac{4}{5}$ =	6000	,,
Glass in roof	1500	,,
Then total area of brickwork is . . .	3820	,,
Glass in walls	1500	,,
Concrete floor	5600	,,
Asbestos roof	6000	,,
Glass in roof	1500	,,
Doors	168	,,

Referring now to the constants in the Table, the heat transmission through $4\frac{1}{2}$ in. brick wall is 0·5 B.Th.U. per sq ft per degree F per hour. Therefore,

3820 × 35 × 0·5 = 92,000 B.Th.U. per hour.

Similarly for glass (K = 1·07)
1500 × 35 × 1·07 = 56,000 ,, ,,

Concrete floor (K = 0·49)
5600 × 35 × 0·49 = 96,000 ,, ,,

Asbestos roof (K = 1·45)
6000 × 35 × 1·45 = 305,000 ,, ,,

Glass in roof (K = 1·16)
1500 × 35 × 1·16 = 61,000 ,, ,,

Doors (K = 0·45)
168 × 35 × 0·45 = 2640 ,, ,,

The total heat loss through the fabric of the building is therefore 612,640 B.Th.U. per hour, and, adding to this the 224,000 B.Th.U. required to maintain the air temperature, we get the total figure

836,640 B.Th.U. per hour.

With steam supplied at 5 lb per sq in. the steam required is 890 lb per hour, and the conditions could be met by eight unit heaters, each giving a nominal output of 100,000 B.Th.U. per hour.

TABLE VIII

Heat Loss Coefficients from Buildings

	Thickness, in.	K
Walls		
Brick walls, plain	4½	0·5
	9	0·35
	13½	0·27
	18	0·23
	22	0·20
	27	0·18
Brick walls with plaster one side	4½	0·45
	9	0·33
	13½	0·25
	18	0·22
	22	0·19
	26	0·17
Brick walls with 2 in. air space, plastered	11	0·25
	16	0·20
	20	0·17
	24	0·15
	28	0·14
Hollow flettons, plain	4½	0·42
	9	0·30
	13½	0·23
Hollow flettons, plastered	4½	0·38
	9	0·27
	13½	0·20
Portland stone and brick, plastered—		
Stone	9	0·31
Brick	4½	
Stone	9	0·27
Brick	9	
Stone	12	0·23
Brick	9	
Concrete walls, plain	4	0·70
	6	0·88
	8	0·51
	10	0·45
	12	0·41
	16	0·34
Concrete walls, plastered one side	4	0·62
	6	0·52
	8	0·46
	10	0·41
	12	0·38
	16	0·32
Sandstone and granite walls, plain	8	0·56
	12	0·47
	16	0·43
	18	0·38
	24	0·33

TABLE VIII (*continued*)

	Thickness, in.	K
Sandstone walls, with battens, lath and plaster	8	0·50
	12	0·40
	16	0·35
	18	0·33
	24	0·28
Limestone walls, plain	8	0·60
	12	0·49
	16	0·46
	18	0·41
	24	0·35
Limestone walls with battens, lath and plaster	8	0·50
	12	0·40
	16	0·36
	18	0·34
	24	0·30
Asbestos walls, corrugated	$\frac{1}{8}$	1·50
	$\frac{1}{4}$	1·25
Steel walls, corrugated on 1 in. T & G boards	$\frac{3}{16}$	0·48
,, ,, plain	$\frac{3}{16}$	1·20
Asbestos walls, flat sheets	—	0·70
Timber walls, T & G boards	1	0·45
	1½	0·35
Concrete walls, battens, Celotex and plaster	4	0·20
	6	0·19
	8	0·18
	12	0·17
	16	0·16
	20	0·15
Floors		
Concrete on earth	4	0·55
	6	0·49
Concrete 5 in., wood blocks 1¼ in.		0·25
Boarded floor on joists with 4½ in. surface concrete		0·07
Roofs		
Flat roof, asphalt on 6 in. concrete with cement screed		0·32
Flat roof, asphalt on 8 in. concrete with cement screed		0·30
Closed roof spaces—		
Slates or tiles on battens and rafters, lath and plaster to room		0·35
Slates with 1 in. T & G boards under tiles		0·25
,, with roofing felt between		0·19
Open roofs—		
Slates or tiles on battens or rafters		0·80
,, with 1 in. T & G boards		0·35
,, with roofing felt between		0·25

TABLE VIII (*continued*)

	Thickness, in.	K
Glass		
Single windows		1·07
Double windows		0·60
Single skylight		1·16
Double skylight		0·62
Doors		
All-wood		0·45
All-steel		1·20
Wood, upper portion glass		0·75
Steel, upper portion glass		1·15
All-glass, wood framing		0·95
All-glass, steel framing		1·05

CHAPTER VIII

BOILER INSTRUMENTATION AND OPERATION

In principle, the boiler house should be regarded as a self-contained factory, receiving raw materials and producing a finished product. The raw materials are water, fuel, and air, and the finished product is steam, in a defined condition with regard to pressure and final temperature.

Any departure from the state of the finished product may possibly have dire effects upon the goods which the steam consumer is trying to produce. The "goods" may be power derived from some form of prime mover or manufactured articles for the production of which steam is required, or not infrequently, a combination of both.

In the not very remote past a state of affairs existed in by far the majority of industrial concerns, where the engineer was expected to supply steam as and when required, with no more elaborate instruments than the pressure gauge and water gauges which the law compelled. Few in that unfortunate era could have stated with any degree of certainty the amount of steam produced per lb of fuel burned, and with coal at the low price level which then existed, managements were prone to regard such a figure as of purely academic interest, and could by no means be persuaded that the necessary capital expenditure on adequate instruments was worth while.

To-day, although many of the smaller factories are still woefully ill-equipped, it may be said that industry in general is far more alive to what may be termed boiler-house economics than ever before.

Instruments

Instruments may very broadly be divided into two classes—those which indicate or record performance, and those which enable a certain predetermined standard of performance to be maintained. The latter may be termed operational instruments. For instance, the simple measurement of fuel input and steam output will, with suitable calculation involving the calorific value of the fuel and the heat content of the steam produced, furnish an efficiency figure for the plant. But the measurement of these two quantities alone will not enable that figure to be maintained or improved. For this, means are required to indicate the minute-to-minute condition of the plant during operation, so that comparison may be made with a previously established optimum. Thus, although it would neither be practicable nor desirable to provide all boiler plants with the elaborate instrumentation which is usual in power station practice, there is a certain minimum equipment, without which no boiler plant can be expected to work efficiently.

It may be said at once, however, that it is quite useless to provide the simplest range of instruments unless the operatives themselves are not only instructed in their use but can be persuaded to take an intelligent interest in them. That the necessary interpretations are well within the reach of the intelligent operative the present chapter will show.

Steam Flow Meters

The steam flow meter, of which there are several beautiful examples available, is one of the most important instruments for the measurement of boiler output. It is important because, not only can the boiler output be read off at any moment, but also any change of load is indicated at once, before the pressure actually begins to alter. Without it, the only indication to the operative that a change of load has taken place is the rise or fall of the pressure gauge, and it is then too late to take the appropriate steps to remedy matters.

For this reason alone the steam flow meter is preferred by many engineers to the alternative method of water measurement. Apart from the fact that it measures actual steam output from the boiler, it is not so susceptible to action by the boiler operatives. A moment's thought shows that where the water meter alone is employed the most astonishingly high outputs can be recorded, by simply opening the boiler blow-down.

Principles of Operation

Steam flow meters in general are all operated upon the same principle, although they differ very greatly in details of design.

An orifice or venturi tube (one or the other) is inserted in the steam main, tappings are taken off at the appropriate distances on either side, and the pressure drop is measured. It is in the means taken to indicate and record this pressure drop that the various makes of instrument differ most widely. Some employ purely mechanical methods, while others use an electrical system. But whatever method is used, the relationship between the pressure drop on either side of the orifice or venturi may be expressed by the fundamental formula, $V = \sqrt{2gh}$ where V is the velocity of the steam through the orifice, h is the resulting pressure drop, and g the gravitational coefficient $= 32{\cdot}2$. Thus, it may be stated that the velocity is proportional to the square root of the pressure drop.

But the velocity must vary as the volume of the flow, so that if the measuring instrument be calibrated in terms of the rate of flow in suitable units, a direct reading is given at once.

The operating components of a very well-known type of instrument are shown in Fig. 99. The orifice plate consists of a thin disc of stainless steel or Monel metal in the centre of which an orifice, whose size is carefully calculated for the particular conditions of steam state, is provided. A smaller hole, whose function is to provide means for the free passage of condensate along the bottom of the pipe, is bored in the lower portion of the plate. The plate is simply inserted between two flanges in the pipe system. On either side of the flanges bosses are formed, either during the manufacture of the pipe or more frequently by welding at a later stage. The position of these bosses is important, the one on the upstream side of the orifice plate being situated farther from the plate than its fellow. The bosses are tapped and fitted with connections for small reservoirs or condensing chambers.

The outlets of these chambers are fitted with valves, from which impulse pipes are taken to the meter body proper. This is essentially a U tube manometer, in which a column of mercury responds to the pressure fluctuations transmitted down the impulse pipes from the tappings. At this point a most ingenious device is employed, to transfer the movements of the mercury column to the reading instrument.

Suspended from the cover of the meter body is a specially-designed resistance element, formed of a number of conductor rods of varying lengths, each connected electrically with a section of resistance in the body of the element. As the height of the mercury column is raised progressively above the zero point, more and more of the conductor rods are contacted, and the resistance is cut out in exact proportion to the movement of the column. But as the column is moved by the variations in the differential pressure, which in its turn varies as the square of the velocity, already shown to be proportional to the volume of steam flowing, it follows that variations of steam flow have

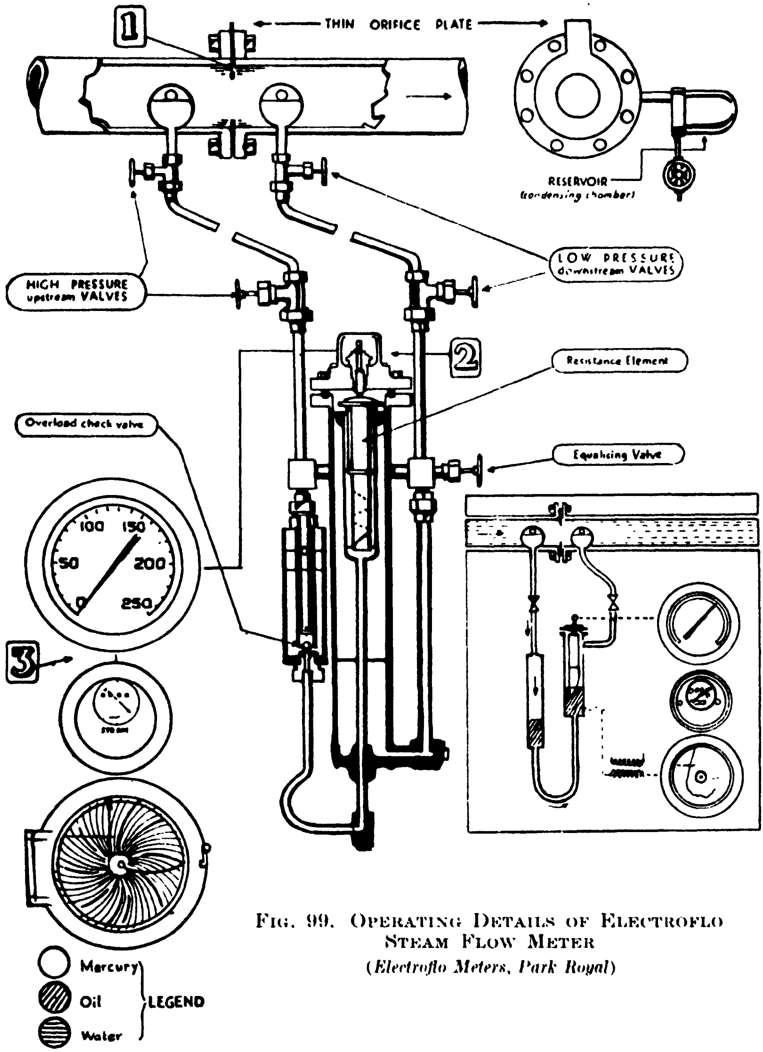


FIG. 99. OPERATING DETAILS OF ELECTROFLO STEAM FLOW METER
(*Electroflo Meters, Park Royal*)

now been translated into electrical terms, that is to say, variations of electrical resistance.

If now the element be connected to an electrical supply, and to a meter, so constructed that it will measure *resistance*, we

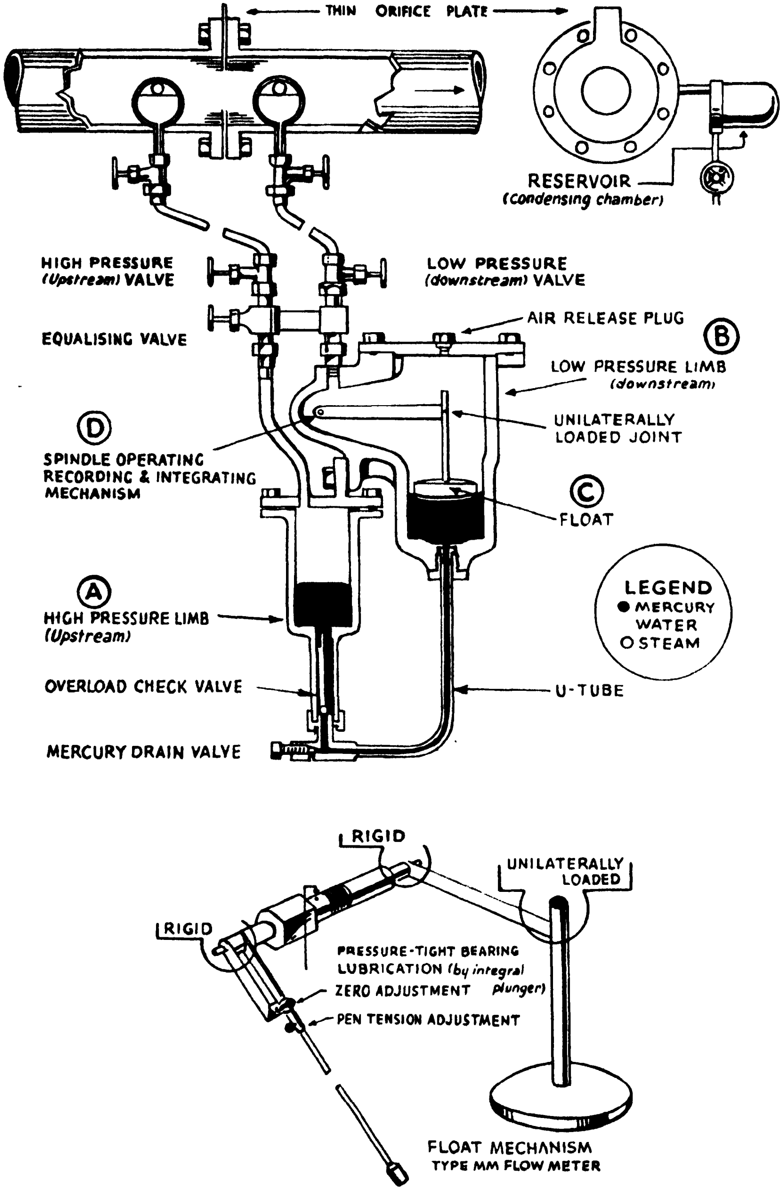


FIG. 100. OPERATING DETAILS OF MM TYPE STEAM FLOW METER
(*Electroflo Meters, Park Royal*)

have a means of reading directly from that meter the steam flow rate it is desired to measure. It is important to remember that the reading instrument must measure *resistance*, and must not be affected by voltage variations in the electrical supply. This is an important feature of the equipment illustrated.

Another type of instrument is illustrated in Fig. 100. Here, exactly the same method of transmitting the pressure drop variations on either side of an orifice is used, and as before, these fluctuations are made to operate a mercury column. At this point the resemblance ceases, and purely mechanical means are employed to transmit the movements of the mercury to the reading instrument.

In common with the electrical meter, this movement can be made to operate integrating and recording appliances if desired.

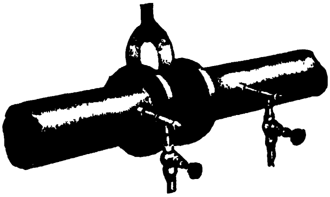


FIG. 101. THIN ORIFICE PLATE AND FITTING
(*Electroflo Meters, Park Royal*)

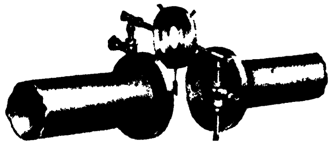


FIG. 102. CARRIER RING ORIFICE PLATE AND FITTING
(*Electroflo Meters, Park Royal*)

Both types of meter may be provided with the carrier ring orifice plate for the pressure difference device. Where this is used it is not necessary to tap the pipe, but allowance must be made between the flanges, for the orifice is carried in a ring, about $1\frac{1}{4}$ in. thick.

For certain locations the carrier ring orifice is rather more accurate than the thin plate type, and is rather less affected by flow disturbances caused by changes in pipe direction and similar factors. Also, the design is such that the pressure differences are averaged over the whole pipe area.

Both types of orifice, and the methods of fitting them, are shown in Figs. 101 and 102.

Since any form of meter working on the pressure drop principle is actually a velocity meter, it follows that any variation in the specific volume of the fluid measured must affect the meter readings. Where steam is the fluid, the specific volume changes with the pressure, and therefore the meter readings are affected by pressure changes in the system. The makers supply correction curves for every meter, so that the proper allowances may be worked out. A typical correction curve is given in Fig. 103.

The accuracy of these meters tends to fall away a little as the reading approaches the ends of the scale, but by careful selection of the range of instrument required, difficulty from this source can be eliminated or at least so reduced as to become unimportant.

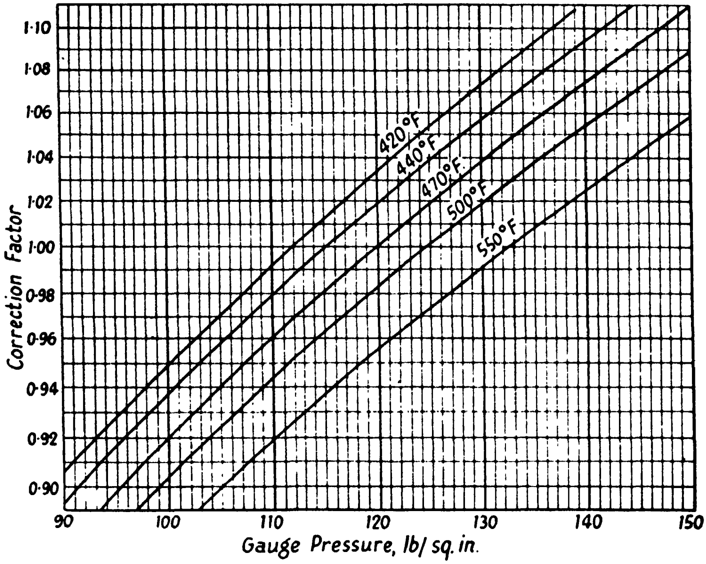


FIG. 103. TYPICAL CORRECTION CURVE
(*Electroflo Meters, Park Royal*)

With regard to the reading instruments themselves, there are a number of patterns available, and the correct choice must obviously be influenced to some extent by the degree of instrumentation of the complete plant. If the steam flow equipment is purchased separately from the other instruments then a simple panel consisting of Indicator, Integrator, and Recorder will no doubt be preferred. Power station practice tends to the large and elaborate control panels shown in Fig. 104 but between these two extremes almost any requirement can be provided.

Water Measurement

It is sometimes considered desirable to measure the boiler feed water. Usually where the feed water is measured steam flow meters are provided as well, because the information which can be provided by feed measurement alone is limited. True, accurate figures will be provided as to the feed input to the boiler over a period, and in certain cases the rate of flow at any instant may be read off.

But it is important to remember that the rate of feed may not be coincident with the steam demand at the time, this being especially true of boilers with large water capacity. With Lancashire and similar boilers, the very large heat storage capacity is utilized when dealing with violent peaks, and very often the pumps are slowed down, and the water level allowed to fall during a short but heavy pull. True, the level must afterwards be made up when the load falls off, but a moment's consideration will show that under such circumstances an entirely false impression would be given by the rate of flow as indicated by the feed water meter. Moreover, the rate of feed can only alter when the pumps are speeded up or slowed down by the operator, which means that the water meter can only vary *after* some action has been taken, and cannot therefore indicate beforehand when such action is necessary. It does not therefore indicate load change in the same way as the steam flow meter. All the same, it is possibly the most useful of all instruments for measuring an overall water consumption figure over extended periods. It should not be used alone for efficiency calculations unless special precautions are taken with regard to blow-down, leakages from pump glands, and the like.

FIG. 104. LARGE POWER STATION CONTROL PANEL
(*Electroflo Meters, Park Royal*)

With certain types, some care is necessary, particularly with regard to choice of position. The accuracy may be affected by the pulsations of the feed pump if this is reciprocating, and the error from this source is particularly distressing, because it is not constant and cannot be corrected with certainty.

A particularly interesting and reliable type of water meter, which is not affected in this way, is shown in Fig. 105. This is the Lea Recorder, which employs the well-known principle of the *V* notch. Usually, the water flows by gravity into the notch tank which is integral with the instrument. The tank is provided with a central partition, in which is fitted a *V* notch, calculated for the particular flow limits. The compartment into which the

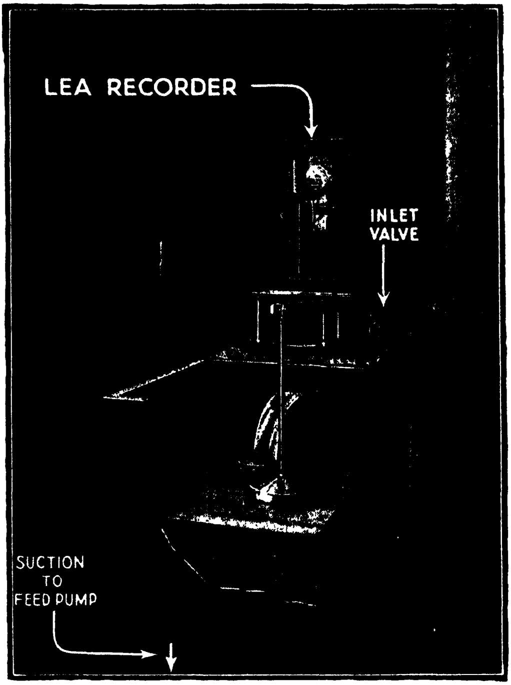


FIG. 105. LEA RECORDER (WATER METER)
(*Lea Recorder Co., Manchester*)

water flows after passing the notch is connected to the suction of the feed pumps, and any variation of the level at this point is transmitted by a float to a valve controlling the inlet to the notch tank. Thus the level of the water before the notch will always be related to the rate at which the pumps are working.

The meter itself is float-operated, and integrates and records the water flowing.

Coal Measurement

For the measurement of the coal burnt, the means employed must depend primarily on the size of the installation, and also on whether the units are hand- or stoker-fired. Where hand-firing is used, there can be only two means. Either the coal must be weighed in bulk before delivery to the firing floor—a method which is likely to give little indication of the performance of individual boilers—or some means must be provided to measure the coal in smaller quantities as fed to each unit.

For bulk-weighing in large quantities, some form of automatic weighing machine, such as the Avery, may be installed. The coal is delivered to the machine as required, weighed and tipped automatically, and the weight recorded by a counter mechanism. In large power stations these machines are frequently fitted to individual boiler units, but although they are without doubt the most accurate of the usual commercial methods of coal measuring, the first cost is comparatively high per machine, so that they are usually considered in connection with large quantities.

With hand-firing, there is no simple method of metering the coal to single units continuously, but it is still necessary that the relationship between water evaporated and coal burned should be known. Snap testing is of little use, and the testing should be as continuous as possible.

Unless some means of continuous weighing to *each* boiler can be devised, the volumetric method must be resorted to. A bottomless box of known capacity should be provided for each boiler, and a shift record kept of the number of times it is emptied. If the box is of suitable size, say 20 cu ft, this does not entail any great hardship on the part of the firemen, and the method is surprisingly accurate. The weight of British coals is about 42 to 45 lb per cu ft, and it is a somewhat startling fact, long ago realized and made public by the manufacturers of the Lea coal meter, that the sizing of the coal makes little or no difference to the volume/weight ratio. In other words, whether the coal be in the form of nuts, beans, or peas, the weight per cu ft of the same class of coal varies but little.

Lea Coal Meter

This fact is the basis on which the Lea coal meter operates. The meter can be fitted to most designs of mechanical stoker, and measures volume, not weight. A chain-grate stoker fitted with one of these meters is shown in Fig. 106. The principle is clearly seen by referring to this and to the diagram in Fig. 107 together.

A specially designed drum cam is driven by chain from the stoker worm shaft or other suitable point, such that there is a fixed relationship between the speed of the chain-grate and the speed of the drum. Engaging with the drum, and so arranged as to be movable along its length, is a small intermediate pinion, which drives a second pinion, equal in length to the drum cam.

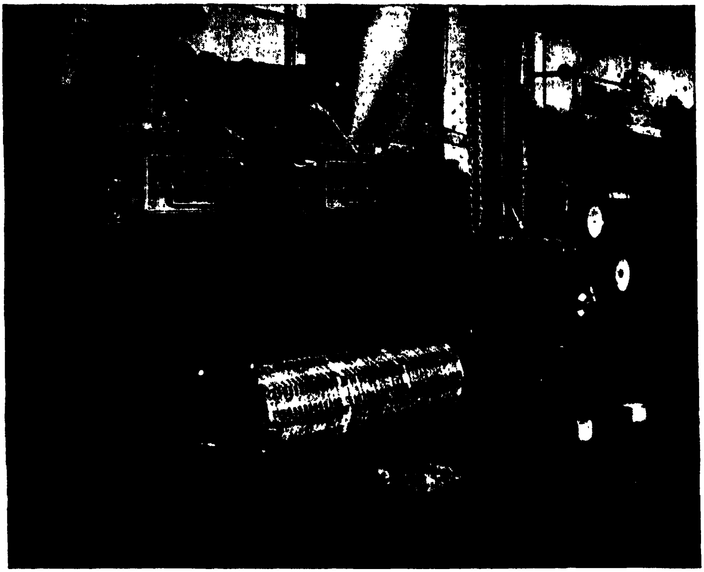


FIG. 106. LEA COAL METER FITTED TO CHAIN-GRATE STOKER
(*Lea Recorder Co., Manchester*)

and toothed along the whole of that length. Thus, whatever the position assumed by the intermediate pinion along the drum cam, it is always in mesh with the second pinion, which is directly connected to the counting mechanism. The intermediate pinion is mounted in a special sliding carriage, which is restrained in one direction by a strong spring. The carriage is directly connected to the guillotine door of the stoker, so that the position of the guillotine determines the position of the carriage in relation to the length of the drum.

The length of the teeth on the drum is graduated, so that the number of teeth engaging the pinion per revolution depends on the position of the pinion along the drum. Thus, taking the two extreme positions, with the guillotine door fully open, the pinion will be at the end of the drum where all the teeth are in mesh, while at the other end, with the fire door fully closed the pinion will be pulled clear of the drum teeth altogether. For any intermediate position, the carefully calculated tooth lengths ensure that the number of teeth engaged shall be exactly proportionate to the door opening. Thus, we have the two variables necessary to measure volume completely provided for. The speed of revolution of the drum is a function of the grate speed, and varies with it, and the number of teeth engaged

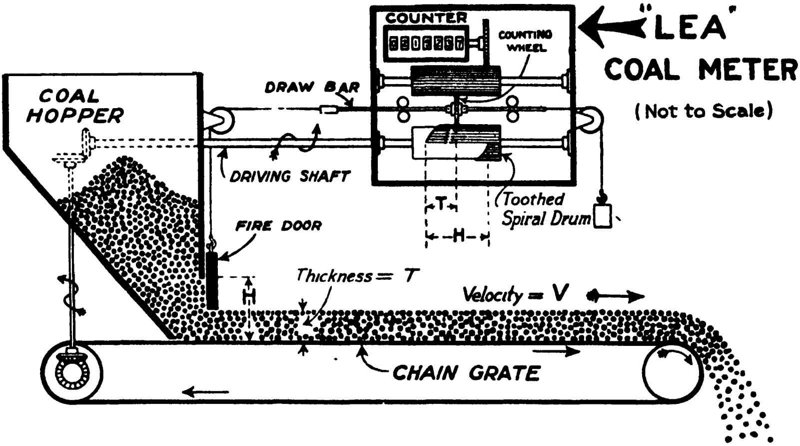


FIG. 107. LEA COAL METER, DIAGRAM OF WORKING
(*Lea Recorder Co., Manchester*)

per revolution varies with the fire thickness, i.e. the height of the guillotine door.

When the meter is fitted to the boiler, a test is taken. A known volume (therefore a known weight) of the coal is put through, and the meter readings are taken at the beginning and end of the test. From this, a simple multiplier is derived from which the coal weight corresponding to any meter reading may be calculated. The same principle is observed whatever the class of stoker, but the methods of driving the meter vary considerably, especially where reciprocating movements have to be dealt with.

Romer-Lea Chute Meter

There is another form of meter which is worthy of attention, for it may form the solution to an otherwise difficult problem. This is the Romer-Lea chute meter, illustrated in Fig. 108. For chutes not more than 30° out of the vertical, this very simple device forms a quite satisfactory means of coal measurement. Again the volume principle is used, and the makers claim that where conditions with regard to verticality and "immersion length" are satisfactory, accuracy to within 2 per cent is obtained.

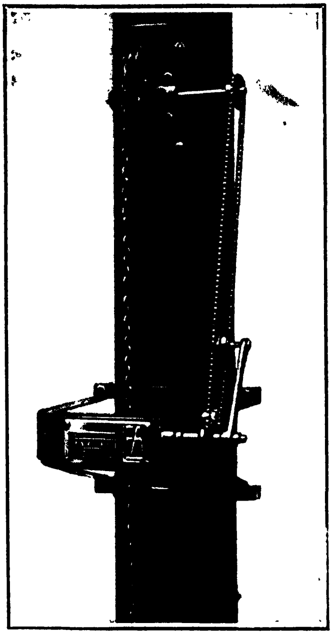


FIG. 108. ROMER-LEA CHUTE METER
(*Lea Recorder Co., Manchester*)

For bulk measurement, the same makers supply the Lea Cubi-Meter. This is constructed on the principle already described for the chain-grate stoker, the coal volume being measured in exactly the same way. The meter is in fact a chain-grate stoker in miniature, the "grate" being in the form of an endless conveyer, fitted with a hopper and door or flap, which trails on the moving surface of the coal. The conveyer may move at a much higher speed than an actual stoker grate, since the apparatus is a measuring device only, and questions of combustion have not to be considered. The device may be used in a variety of ways, and may most conveniently be mounted at the loading end of a conveyer-belt serving a battery of boilers.

With the aid of the two instruments mentioned, i.e. the steam flow meter, and the coal measuring device, either coal meter or bottomless box, we are able to assess directly one vital piece of information, which is the lb of steam evaporated per lb of coal burned, but we cannot without further knowledge calculate the boiler efficiency. For this, we must know the heat input and output of the boiler.

True, the lb of steam per lb of coal may be used to obtain the costs of the *steam* per 1000 lb or per ton, but valuable though this information may be, it is not a criterion of boiler performance.

The boiler heat output is easily obtained by reference to steam Tables.

Example.—A boiler works at 125 lb per sq in. gauge, stop valve temperature 500° F, feed water temperature 160° F.

The heat content per lb of steam from the feed water temperature will be

$$1273 - (160 - 32) = 1145 \text{ B.Th.U. per lb.}$$

If the boiler is evaporating 6000 lb of steam per hour, it is obviously only necessary to multiply by this figure in order to obtain the hourly heat output.

Suppose that at the same evaporation the boiler burns 725 lb of coal per hour, we may at once obtain the basic figure of

$$\frac{6000}{725} = 8{\cdot}3 \text{ lb of steam per lb of coal.}$$

Assessment of Coal Calorific Value

The heat content or calorific value of the coal is needed before further calculation can be made. This figure may be obtained either by the use of a bomb or other form of calorimeter—instruments which are not in the possession of every works, and whose use hardly comes within the province of the works engineer—or by the following simple method.

The apparatus required is a crucible with lid, a Bunsen burner, and a good balance.

The procedure is as follows—

Weigh the crucible without its lid, place in it a small sample of the powdered fuel, and re-weigh. Obtain from this the weight of the fuel, W_1. Place in a dry and evenly warmed place for some hours and re-weigh. Repeat until there is no further loss of weight, the final result being W_2.

Then $$\frac{W_1 - W_2}{W_1} \times 100 = \text{percentage moisture.}$$

Place the crucible on a tripod stand and arrange the lid so as to leave a small gap at the side. Adjust the flame of the Bunsen so that only the tip envelops the crucible. In a few minutes the gases escaping from the side of the lid will ignite. After the flame has died away, continue the heating for another two minutes. Allow to cool in a dessicator or inverted glass, and re-weigh.

Calculate the percentage loss in weight, which gives the percentage volatiles. Set the crucible well down in the Bunsen flame, remove the lid, and heat strongly. When the combustion appears to be complete, cool and re-weigh. Repeat till there is no further loss in weight. The loss per cent is the fixed carbon, and the residue is the ash content.

The calorific value of the coal may now be calculated from the well-known formula of Goutal.

$$\text{B.Th.U. per lb} = 147{\cdot}6C + (1{\cdot}8a \times V)$$

where C = percentage fixed carbon
V = percentage volatiles
a = a constant derived from V_1, and reference to the curve in Fig. 109.

$$V_1 = \frac{100V}{C + V}$$

The formula will give satisfactory results over a fairly wide range of bituminous coals, but the bomb calorimeter test is more accurate for the estimation of calorific value, although it does not furnish information as to the proportions of the coal constituents. During the test detailed, the behaviour of the coal while burning can be observed, and some idea gained as to the best procedure to be adopted in actual practice.

An example is here given of an actual coal.

	Per cent
Moisture	7·98
Ash	5·58
Fixed Carbon	62·17 = C
Volatiles	24·27 = V

Then $$V_1 = \frac{100V}{C + V} = \frac{2427}{86{\cdot}44} = 28{\cdot}1$$

Reference to the curve in Fig. 109 gives a as approximately 100 for the above value of V_1, so that

$$\begin{aligned} \text{B.Th.U. per lb} &= 147{\cdot}6C + (1{\cdot}8a \times V) \\ &= 9170 + (180 \times 24{\cdot}27) \\ &= 13{,}530 \text{ B.Th.U. per lb.} \end{aligned}$$

The calorific value of this coal as obtained by calorimeter was actually 13,820 B.Th.U. per lb, so that the error is only 2 per cent.

We have already ascertained from the readings of the steam flow meter and measurement of the coal input that 8·3 lb of steam are obtained for each lb of coal burned, and that the heat required by the steam from the existing feed temperature is 1145 B.Th.U. per lb.

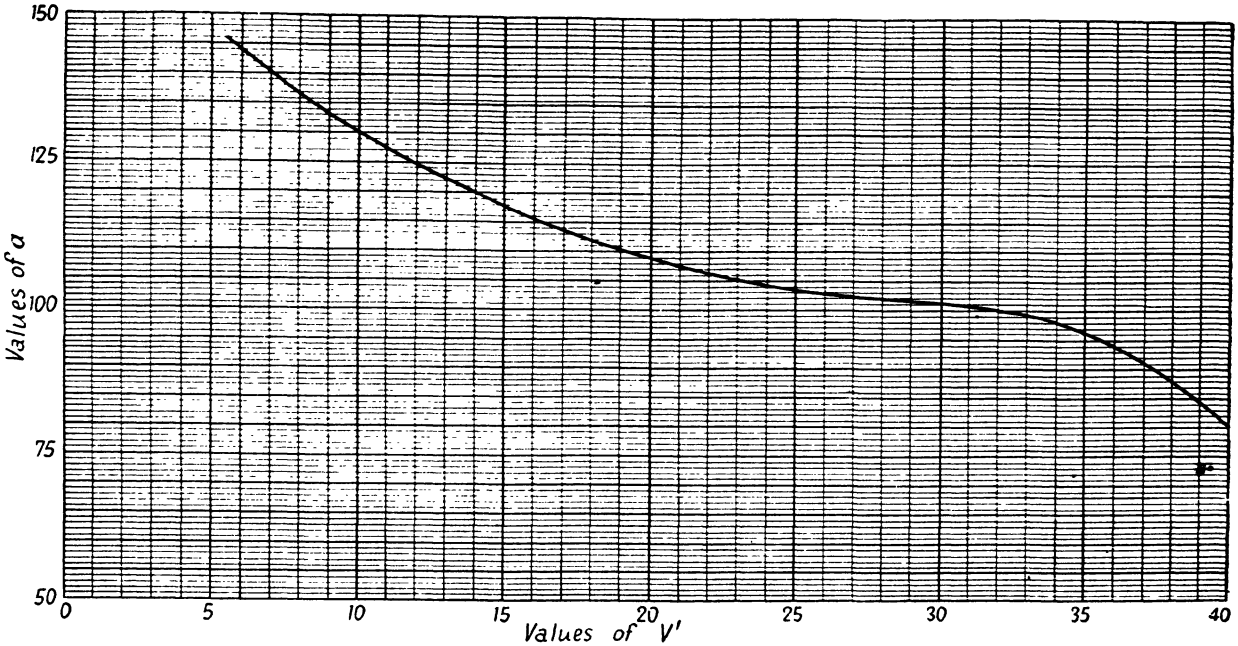


Fig. 109. Curve of Goutal's Factors

Therefore the efficiency of the boiler which *is* the criterion of performance may now be calculated thus—

$$\frac{1145 \times 8{\cdot}3 \times 100}{13{,}530} = 70{\cdot}25 \text{ per cent.}$$

This figure, informative though it may be, simply indicates that the plant has attained a certain standard of performance, but from the *operational* standpoint, it has little value. In other words, the measurement of coal and steam, though necessary, merely checks an accomplished result, but does not provide the type of information which enables a desired result to be achieved.

Operational Instruments

The conditions of combustion must be indicated or recorded in such a way that the proportions of fuel and air may be either observed directly or deduced from the information given, and the appropriate action taken by the operative in sufficient time for it to be effective. It is important to realize that only by corrective action carried out immediately there is a deviation from the best conditions can the optimum be maintained.

CO_2 Recorder

Most important of what may be termed the operational instruments is undoubtedly the CO_2 recorder. This is at once the most valuable and the most delicate of boiler house instruments; delicate, not because of inherent fragility of structure, but because the very nature of the function carried out by the whole assembly of sampling pipes, aspirators, etc., and the small forces normally available to operate the recording instrument, make the system somewhat liable to derangement if care is not used, both in installation and use. Much ingenuity has been exercised by the makers on the design of these instruments.

Principles of Combustion

Three principles will be briefly described. One of the most popular instruments is shown in Fig. 110, which shows the recording instrument itself, and Fig. 111, which shows the combined aspirator and meter. The instrument is electrically operated, and in its essentials is a Wheatstone Bridge, one-half of which is surrounded by the flue gas, and the other by pure air. A small steady potential is applied to the ends of the bridge, the elements of which lose heat at a rate corresponding to the thermal conductivity of the surrounding gases or air. Thermal equilibrium is therefore reached with the two parts of the bridge at differing temperatures. But the electrical conductivity of the wire composing the elements varies with the temperature, so that there is an out-of-balance on the system, which is arranged to affect a sensitive galvanometer calibrated in percentage CO_2.

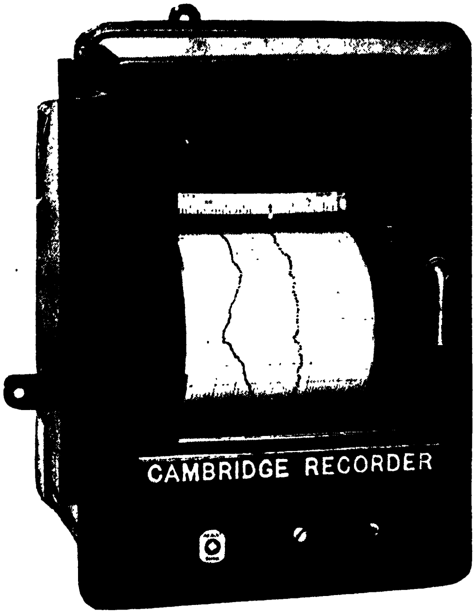


FIG. 110. CAMBRIDGE CO_2 RECORDER
(Cambridge Instrument Co.)

This particular instrument has the great advantage that no chemical reagents are used, and it therefore needs less routine attention than instruments working on other principles. The small potential required may be supplied either from a specially designed mains unit or from a battery. A potentiometer is provided at the reading instrument, by means of which the potential may be adjusted at need, to give a precise full-scale reading.

A water supply, 4 to 5 gallons per hour, at a head of from 2 to 10 ft, must be available to work the aspirator. It is usual to fit a small overhead tank kept to a constant head by a ball-cock for this service.

The reading instrument is fitted in a strong dust-proof case, and is quite robust enough to mount on a panel in the boiler house, where it can easily be seen by the operatives. If desired, an illuminated dial instrument, readable from a distance, can be supplied in addition to the recorder itself.

In the chemical type of instrument the gases are drawn continuously or at controlled intervals, through some substance which has the property of absorbing CO_2. The pressure difference so set up operates the instrument, which thus reads in proportion to the amount of CO_2 in the gases.

Yet another type of instrument makes use of the difference in the densities of flue gas and air. Two fans revolve rapidly in separate chambers, which are filled respectively with air and flue gas. In each chamber, closely adjacent to the fan, is a

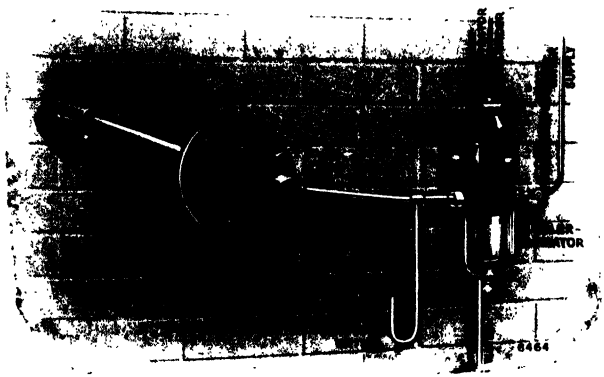


FIG. 111. SAMPLING PIPE AND ASPIRATOR OF CAMBRIDGE CO_2 RECORDER
(*Cambridge Instrument Co.*)

vaned disc, so mounted that it tends to rotate with the friction of the gas or air set in motion by its fan. The torque thus set up varies with the density of the gases, and the square of the speed of the impeller. The torque difference is mechanically arranged to operate the reading instrument.

Whichever type of instrument is finally chosen, satisfactory operation will largely depend on the care taken in the initial installation and choice of position.

The pipe system between the sampling pipe and the instrument should be as short and simple as possible, since on this will depend the speed of response in operation. The position of the sampling pipe is of vital importance. The general principle is that at the point selected, the combustion must be as complete as the conditions will allow, and where flue filters are used, the temperature should be high enough to burn off any deposits that may form. In the Lancashire boiler these conditions are found in front of the side flue dampers, but the pipe must be fitted sufficiently far away not to be affected by the turbulence set up by the contraction of the gas path, which may cause difficulties with the aspiration.

In water tube boilers there is much more scope for choice, and some experiment may be necessary before a suitable position is found. Much depends on the individual setting, and a position which is quite satisfactory on one boiler may be useless on another.

The arrangement of the tube tiles has a great influence on the suitability or otherwise of a position. The sampling pipe must be given a good slope towards the meter, and an ample water trap should be fitted. The makers' advice on this should always be followed to the letter. The pipe must be easily dismantled, and for this purpose should be installed in a special gland bricked into the setting. Even under the best conditions hard deposits are likely to form in the pipe, and it is a troublesome business to remove these unless the work can be done on the bench. Very often it is cheaper to scrap the choked pipe and fit a new one.

In order to understand the importance of the part played by the CO_2 recorder in boiler operation it is necessary to know something of the mechanics of combustion. It is at least necessary to grasp the significance of flue gas composition.

Bituminous coal is composed of carbon, and varying amounts of hydrogen, oxygen, nitrogen, and sulphur, together with incombustible minerals. When heated in sufficient air, the carbon content combines with the available oxygen to form CO_2, releasing 14,500 B.Th.U. for each lb of carbon in the process. The hydrogen also combines with oxygen, forming water vapour. If the air supply is not sufficient, the carbon is not completely oxidized, but CO is formed, and passes away into the flues. In the conversion to CO, only 4300 B.Th.U. are released, which means a loss of over 10,000 B.Th.U. for every lb of carbon so converted. When this fact is realized, the value of the CO_2 recorder is at once apparent, and it is seen that the ideal condition would be to supply just sufficient air to the furnace to convert all the carbon to CO_2 without the formation of CO.

Unfortunately, the theoretically correct amount of air can never be supplied in practice, at least on stoker- or hand-fired boilers, because there must always be sufficient excess air completely to surround and combust every particle of the fuel. Too much excess air on the other hand leads to very serious losses indeed, due to the heat carried away to waste by large volumes of air, which have had to be drawn through the fires, absorbing valuable heat on the way. Fig. 112 shows the loss of sensible heat at various flue temperatures and CO_2 readings.

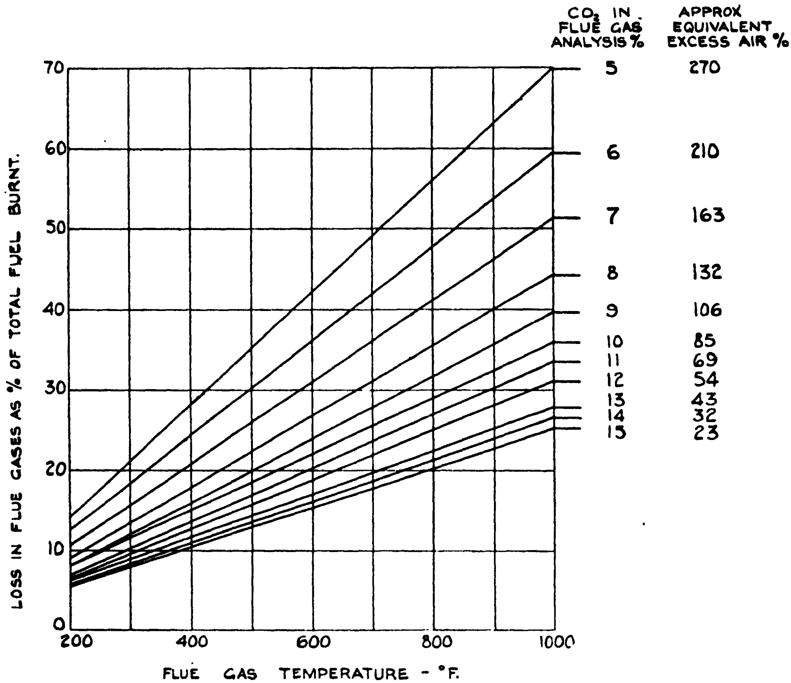


FIG. 112. CURVES OF FLUE LOSSES AT VARIOUS CO_2 READINGS AND FLUE TEMPERATURES

The CO_2 recorder gives a direct indication of combustion conditions by showing, from minute to minute, the composition of the flue gases.

Herein lies the importance of the instrument; but in addition a knowledge of the exit temperatures, as well as the CO_2 percentage, enables a direct calculation to be made of the greatest single loss in the whole plant. This is the sensible heat carried away in the flue gases. Where sensible heat is involved, any formulae that may be used must include terms which take into account the weight of the gases, or their composition (from which the weight may be calculated), the specific heat, and the temperature rise above that of the inlet air.

Calculation of Losses

Where a complete analysis is possible the more orthodox method is to calculate the weight, and compute the loss directly from

$$\text{Heat loss in gases} = W(T - t) \times 0{\cdot}241$$

where W = weight of gases generated per lb of coal burnt, calculated from the analysis.
T = exit temperature of the gases.
t = temperature of the air admitted to the furnace.
$0{\cdot}241$ = specific heat of the gases between the temperature limits involved.

The result will be shown as B.Th.U. lost per lb of coal burned. In factory practice, it is seldom that a complete gas analysis can be made with sufficient frequency to be of practical use, except perhaps in the case of some particularly important or special test, for which the services of the laboratory can be temporarily enlisted.

The Orsat apparatus may be used of course, but unless the sampling is extended over a considerable period the analysis so obtained is likely to bear little relation to reality, not because of inherent defects in the apparatus, but because of the difficulty of maintaining sufficiently constant conditions on the plant. But with the aid of a reliable flue gas thermometer, and the recorded CO_2 readings, and without resorting either to the laboratory or to the Orsat apparatus, the flue loss may be calculated by the formula already introduced in the chapter on Economizers.

$$\text{Flue loss percentage} = \frac{K(T - t)}{CO_2 \text{ percentage}}$$

As before, T is the final temperature of the flue gases, and t the temperature of the incoming air. K is a constant derived from the specific heat, and the relationship between the weight of CO_2, and that of the remaining constituents.

For the conditions met within boiler house practice, and for a wide range of bituminous coals, K may safely be taken as 0·35.

Refer now to the example quoted.

Calorific value of the coal .	13,530 B.Th.U. per lb
Heat absorbed by the steam.	1145 B.Th.U. per lb
Lb steam per lb coal . .	8·3

From this we deduced the efficiency as 70·25 per cent.

With the information made available by the CO_2 recorder and the flue gas temperature, we are now able to isolate some of the losses and calculate them separately.

EXAMPLE.—Assume the temperature at the flue exit to be 450° F and the CO_2 percentage to be 9.

$$\text{Then} \qquad \text{Flue loss percentage} = \frac{0{\cdot}35\,(450 - 70)}{9}$$

$$= 14{\cdot}8 \text{ per cent.}$$

Therefore, of the total loss of 29·75 per cent, 14·8 per cent is the flue loss, and 14·95 per cent remains to be accounted for. Reference to the proximate analysis of the coal gives the moisture content as 7·98 per cent, that is to say, each lb of coal contains 0·0798 lb of water, which must be converted into steam, and superheated to the temperature of the furnace. During its passage through the boiler passes some of the heat is given back to the boiler heating surfaces, so that it is only necessary to calculate the superheat *to the temperature of the exit gases.*

The expression used must include weight, the specific heat of superheated steam within the limits of the temperatures involved, and the latent heat of evaporation at atmospheric pressure. The heat taken up by the water to 212° F, will obviously be $(212 - t)$ B.Th.U. per lb. To convert the whole into steam at atmospheric pressure the latent heat of evaporation must be added. Reference to steam tables gives this as 970·4, so that the expression to the point at which one lb of water is converted into steam is $(212 - t) + 970{\cdot}4$.

We must now consider the heat required to superheat this steam to the temperature of the exit gases. This must be $(T - 212)$ multiplied by the specific heat of superheated steam, which is 0·48.

We now have an expression in two parts, the first part dealing with the conversion of one pound of water into steam, and the second part with the extra heat required to superheat that steam to a known temperature. The complete figure for the heat absorbed by the moisture will be the sum of these two quantities.

$$(212 - 70) + 970{\cdot}4 + 0{\cdot}48(450 - 212)$$

$$= 1226{\cdot}4 \text{ B.Th.U.}$$

To obtain the heat loss *per lb of coal* it is only necessary to multiply by the known weight of the moisture per lb.

$$0{\cdot}0798 \times 1226{\cdot}4 = 98 \text{ B.Th.U.}$$

Expressed as a percentage on the heat content of the coal this is 0·725 per cent. There is still another loss which may be calculated with the information available.

It was stated earlier that some of the hydrogen content of the coal combined with the oxygen to form water vapour. 1 lb of hydrogen burned will result in the formation of 9 lb of water, so

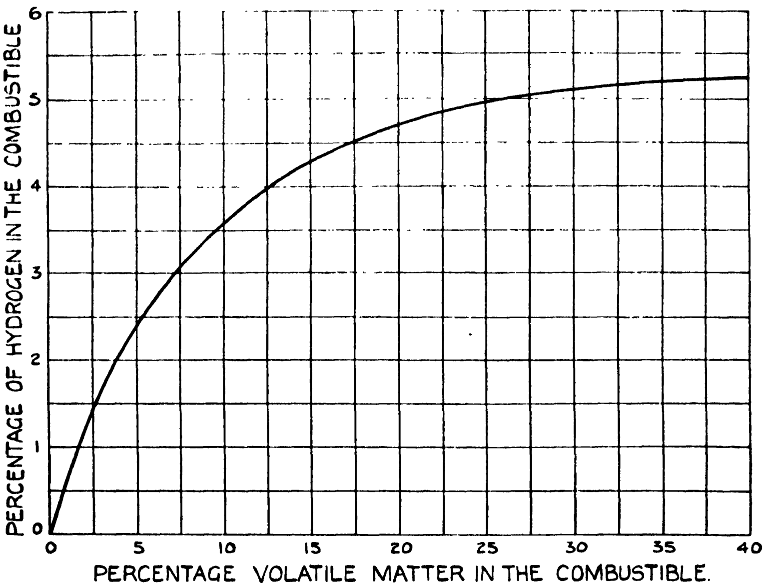


FIG. 113. CURVE SHOWING RELATIONSHIP OF H TO THE VOLATILE CONTENT IN BITUMINOUS COAL

that if the weight of hydrogen per lb of coal can be estimated, the moisture formed can be found very simply.

The curve in Fig. 113 gives the approximate relationship of the H content to the volatiles in bituminous coal. From the analysis, the volatile content of the coal is given as 27·42, and from this the curve gives the corresponding value of H as 4·9 per cent.

Then the weight of H per lb of coal will be 0·049 lb, and the water formed during combustion will be $0{\cdot}049 \times 9 = 0{\cdot}441$ lb.

In the previous calculation the heat absorbed by one pound of water under the conditions existing in the furnace is shown to be 1226·4 B.Th.U., and therefore the loss per lb of coal due to the moisture formed by the H is $1226{\cdot}4 \times 0{\cdot}441 = 540$ B.Th.U. or 4·0 per cent on the calorific value. The loss due to the ash may be written down straight away from the proximate analysis as 5·58 per cent.

Summarizing, it is seen that 95·335 per cent of the heat supplied to the boiler is now accounted for.

There is still a further loss, not yet considered, which is the unburned carbon in the ash. Before the calculation can be carried out the weight of this carbon per lb of ash must be computed by taking a sample of the boiler refuse from the ash heap under actual working conditions, strongly heating a dried and weighed sample. and obtaining the combustible from the loss in weight. It is assumed that the whole of this combustible is pure carbon, giving 14,500 B.Th.U. when burned.

Let the ash obtained from actual firing conditions be 6·0 per cent, and the loss of weight after strongly heating 30 per cent.

Then the loss per lb of coal due to the carbon in the ash is

$$\frac{0{\cdot}06 \times 30}{100} \times 14{,}500$$

$$= 261 \text{ B.Th.U.} = 2 \text{ per cent.}$$

The remaining losses amounting to 2·645 per cent are made up of those due to incomplete combustion of carbon, radiation, and "unaccounted for." Thus, a fairly complete heat balance has been achieved with a minimum of instrumentation.

Another loss, usually neglected but none the less real, is concerned with the loss of power, or rather the excess power required to deal with the larger gas volumes produced as a result of low CO_2 and consequent high flue temperatures.

Where induced draught is fitted, the load on the motor is of course dependent on the work done by the fan. The cubic feet of air per minute passing through the fan is proportional to the speed, but the power absorbed is proportional to the cube of the speed.

The graph in Fig. 26 shows that for a CO_2 reading of 12 per cent, 16·7 lb of flue gas are given off for each lb of fuel burned.

Therefore, if 725 lb of coal per hour are consumed, $\frac{725 \times 16{\cdot}7}{60} = 202$ lb per minute are evolved. The temperature of the gases with this CO_2 reading may be taken as 329° F, and as the weight at this temperature is 0·0509 lb per cu ft, the fan would be dealing with 3960 cu ft per minute. At this load the speed might be 500 r.p.m., and the b.h.p. about 2.

Assume now that the excess air is increased so that the CO_2 falls to 9 per cent. The gas now produced amounts to 22 lb per lb of coal, and the weight is 265 lb per minute. The temperature under the new conditions would rise to 400° F, at which the weight is 0·046 lb per cu ft. The fan must now deal with 5800 cu ft per minute, and for this the speed must be increased to 730 r.p.m.

Since the power absorbed increases as the cube of the speed,

$$\left(\frac{730}{500}\right)^3 \times 2 = 6{\cdot}0 \text{ b.h.p.}$$

Where constant speed fans are used, and the necessary adjustments carried out by the dampers, the power input varies as the square of the volume, but as the fan must be run at all times at the speed necessary to cope with full-load conditions, the saving due to the reduction of the air supply is not so marked.

In the example given, it has been necessary to assume temperatures at various points in the plant; for instance, the total temperature of the steam at the stop valve, and the temperature of the feed water are both shown to be essential in order to calculate the heat given up to the steam. The temperature of the outgoing flue gases in conjunction with the CO_2 reading enables some of the losses to be separated and identified, and a reasonable loss balance made out. It is desirable, however, that some knowledge should be available of the heat distribution throughout the plant, and to obtain this a little more instrumentation is necessary. This should consist of three more thermometers, and at least one draught gauge, of which more will be said presently.

We already have the total steam temperature, and as the temperature of saturated steam at the working pressure can be read off direct from the steam Tables, we are in a position to calculate the work done by boiler alone, and the superheater alone. If the temperature of the feed water can be taken as the temperature of the water entering the economizers, one additional thermometer, giving the water temperature on the outlet side, will in the same way give us the necessary data for the heat absorbed by the economizers. Very valuable as a check, though not strictly essential, is a thermometer giving the flue gas temperature entering the economizer. It is always as well to fit this instrument, for, even if it is not absolutely necessary for routine operation, it *is* essential for tests, and the aim should always be to run the plant as nearly as possible under conditions of continuous test.

Thermometers

Here a word must be said as to the type of thermometer best fitted for the duty. It is useless to fit an instrument from which frequent readings are required in any position from which it

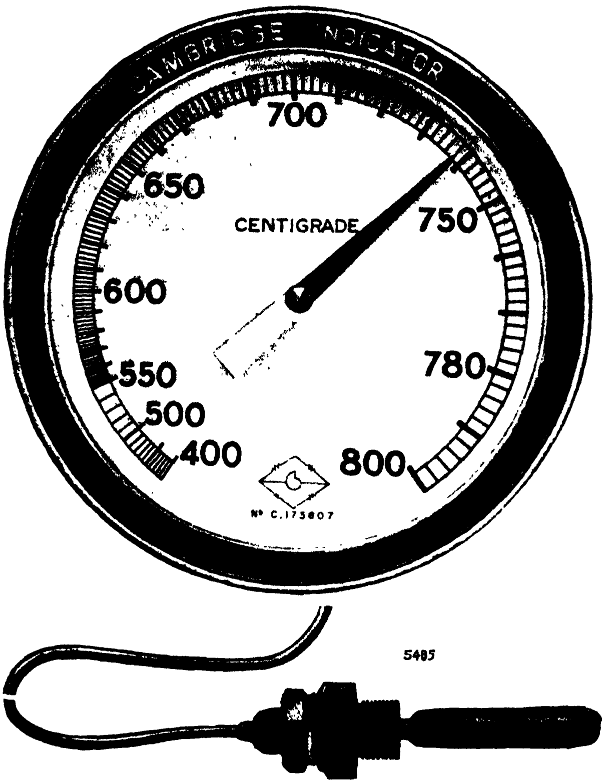


FIG. 114. CAMBRIDGE DIAL TYPE, MERCURY IN STEEL THERMOMETER
(*Cambridge Instrument Co.*)

cannot be clearly *and easily* seen from the firing floor. If the instrumentation of the plant is to be fully exploited, it is necessary that readings should be taken at least every half hour unless every instrument is a recorder, and an installation as elaborate as this may be too expensive for the great majority of commercial boiler houses.

Visibility, with the minimum of trouble for a simple reading instrument, is therefore of the utmost importance; a superheat thermometer, to reach which it is necessary to climb up an iron ladder and crawl under a succession of hot steam pipes, will simply not be read. The ordinary type of mercury column thermometer has therefore a very limited field of use for boiler work, although it is very often put in for the sake of cheapness.

It is far better to install at once a distance-type dial thermometer, an excellent example of which is shown in Fig. 114. The dial of this instrument can be installed as far as 60 ft from the bulb, and the bulb itself is susceptible to great flexibility of design. This, coupled with the fact that a variety of ranges are catered for, makes this design particularly suitable for boiler work. Moreover, the instalment of a number of similar instruments makes it possible to group all the dials, together with the CO_2 recorder and draught gauges, in close proximity, so that the complete assembly will tell the whole story of the plant at all times.

FIG. 115. CAMBRIDGE MULTI-POINT PYROMETER
(*Cambridge Instrument Co.*)

Pyrometer

For larger plants, the electrical pyrometer will perhaps be favoured, the type of instrument as shown in Fig. 115. Here, a number of points are connected to the same reading instrument, it being possible to select any one point by the selector switch. This very neat arrangement is favoured by many engineers, but there is a good deal to be said for the multi-dial system, on the grounds that the operatives get used to seeing the indicating fingers in certain positions relative to each other, and can at once detect an abnormality. This is a valuable operational asset, especially if the dials can be read from a distance.

Heat Distribution

Starting from the data we already know, let us now work out the heat distribution for the plant.

Steam pressure . . . 125 lb per sq in. gauge

Steam stop valve temperature . 500° F
Feed water temperature . . . 160° F
Temperature of water leaving economizers . . . . 300° F
From previous information, 8·3 lb steam per lb coal
Calorific value of coal . . 13,530 B.Th.U. per lb

Work done by the boiler,

$$Eb = \frac{W(H-h)\,100}{C}$$

where W = weight of water evaporated per lb of coal
H = total heat of the steam at the working pressure (saturated)
h = heat in the feed water entering boiler from the economizer
C = calorific value of the coal.

Then $$Eb = \frac{8 \cdot 3[1192 - (300 - 32)] \times 100}{13{,}530} = 56 \cdot 5 \text{ per cent.}$$

Work done by economizers,

$$Ee = \frac{8 \cdot 3(t - t')100}{13{,}530}$$

where t = temperature of water leaving economizer
t' = temperature of water entering.

Then $$Ee = \frac{8 \cdot 3(300 - 160)100}{13{,}530} = 8 \cdot 65 \text{ per cent.}$$

Work done by superheater,

$$Es = \frac{8 \cdot 3 \times 0 \cdot 48(Ts - ts)100}{13{,}530}$$

where 0·48 = specific heat of steam
Ts = stop valve temperature (superheater outlet)
ts = temperature of saturated steam at the working pressure (steam Tables).

Then $$Es = \frac{8 \cdot 3 \times 0 \cdot 48(500 - 353)100}{13{,}530} = 5 \cdot 1 \text{ per cent.}$$

Totalling all the above, we obtain the efficiency originally deduced, 70·25 per cent. Thus, by the minimum of instrumentation, we have not only been able to separate the losses on the plant, but also to assess the work done by the separate plant components.

Draught Gauges

Closely allied to the CO_2 recorder in an operational role is the draught gauge. In its simplest form this may consist of a *U* tube, calibrated as a rule in tenths of one inch, and connected at one end to some suitable point in the path of the flue gases. So connected it reads the difference in pressure, in inches of water, between the atmosphere, and the point in the flue to which it is connected. This difference in pressure depends upon a number of factors, all of which influence the reading. Of themselves, the readings are not quantitative, and alone, with no other information available, the gauge is little more than a tell-tale, informing the fireman of the effect of a particular damper movement, or change of fan speed. By itself therefore, the draught reading gives no clue whatever as to whether the amount of air supplied to the furnace is correct or otherwise. If, however, the amount of gas flowing can be calculated by other means, a relationship can be established between the actual gauge reading and the gas flow. This relationship will remain true, *so long as the resistance of the gas path does not alter*.

During normal working, however, certain changes in the character of the gas path must be regarded as inevitable, and not all of these are under the control of the operator. An increase in the thickness of the fuel bed will cause an increase of resistance which may be termed a voluntary change, but the gradual decrease in the area of the gas path due to deposits of flue dust, or the choking of tubes, or the infiltration of air from the formation of leaks in the setting, are involuntary, but may nevertheless be detected if the instruments are properly interpreted.

It remains now to consider the best practical means of setting up what, for want of a better term, may be called the "basis state" of the particular plant.

Assume a water tube boiler, fitted with induced draught, economizers and chain-grate stoker. The CO_2 recorder is fitted in a position which has been found, by experiment and trial with the Orsat apparatus, to give the truest indication of combustion conditions. The draught gauge is connected in the last pass, immediately before the boiler outlet. A thermometer is inserted in the flue close to the gauge.

It is essential that before taking the basic readings the plant should be completely flued and scaled, and, above all, the setting carefully examined, and all leaks made good. If this is not done, the whole relationship will be false, and the test useless.

Under the above conditions, and steaming at 6000 lb per hour, the CO_2 is 12 per cent and the draught gauge reads 0·25 in. WG. The fire thickness is 4 in. The normal routine readings of steam flow meters and coal meters show that 7·5 lb of steam per lb of coal are being evaporated, and as the flue temperature at the outlet is 380° F the flue loss is round about 9·5 per cent, so that for the type of plant it may be said the efficiency is reasonable.

The coal, from the above figures, is being fed to the boiler at 800 lb per hour. Referring now to Fig. 26, it is seen that for a CO_2 of 12 per cent, 16·7 lb of flue gas per lb of coal are generated, so that in the present case, the weight of gas amounts to 223 lb per minute. If the temperature in the flue near the draught gauge is 500° F, the density of the gas is approximately 24 cu ft per lb, so that the volume is 5350 cu ft per minute. This, then, corresponds to a reading of 0·25 in. WG, *but only for the particular conditions of gas path resistance obtaining at the time of the test.*

It is now possible to calculate the approximate datum reading over a range of gas flows under those same conditions, by using the square root of the readings.

Thus, for a gas flow of say 8000 cu ft per minute,

$$\sqrt{r} = \sqrt{0{\cdot}25} \times \frac{8000}{5350}$$

$$\sqrt{r} = 0{\cdot}746$$

$$r = 0{\cdot}56 \text{ approx. in. WG.}$$

Using this principle, and calculating for a suitable range of gas flows, a curve may be constructed, as shown in Fig. 116, and kept for checking purposes. It is seen that the gas volumes are calculated quite easily by referring to the curve in Fig. 26, and estimating the combustion rates from the routine figures of the plant.

The correction for temperature should always be made by taking the reading of a thermometer specially placed near the draught gauge for the purpose. Once the datum is established, it is sufficient to check up on the boiler once a week. When the check is being carried out, it goes without saying that every effort should be made to see that the controllable factors are as nearly as possible the same as when the basic test was taken.

The firebed should be of the same thickness, well covered and free from holes. Since the gas flow can be ascertained, the load is not important, so long as there is enough information to calculate the combustion rate, and provided that the boiler is neither forced nor badly underloaded. In general, it will be found that increases in resistance, upstream from the draught gauge, will give higher readings than normal for a given gas

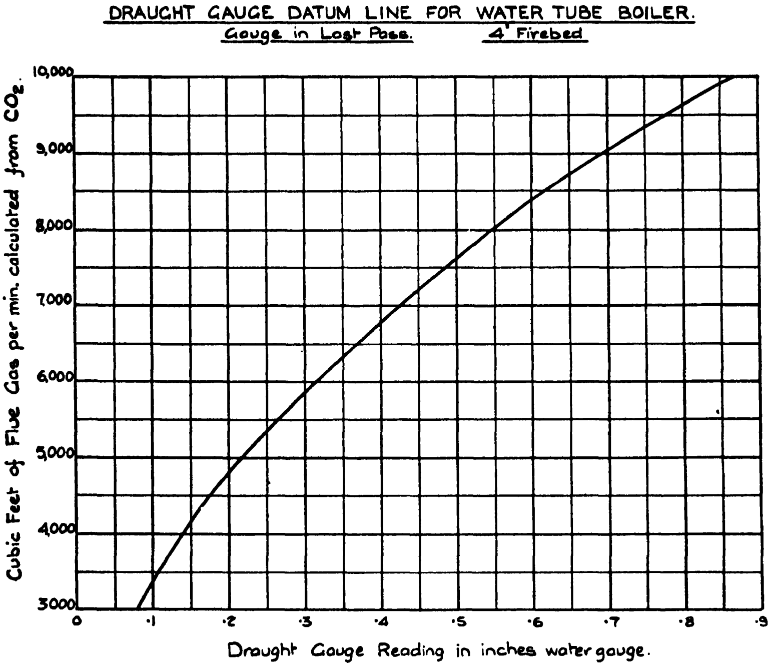


FIG. 116. CURVE OF GAS VOLUMES BY DRAUGHT GAUGE READINGS RESULTING FROM BASIC TEST

flow, so that obstructions due to tube fouling will be indicated by a set of persistently high readings.

Leaks before the draught gauge may not show up unless they are very bad indeed, although actually they are tantamount to a decrease in resistance, but the dilution will cause a drop in CO_2 from which the seriousness of the infiltration can be gauged, again by referring to Fig. 26. Each plant will have its own characteristics, which continued observation will make clear, but it is quite obvious that no hard and fast rule for general application can be laid down.

Even the datum for the individual plant must be regarded as somewhat flexible, and its value lies in the fact that it establishes a standard, however tolerant, from which definite departures from the normal can be detected. Moreover, the very existence of such means of comparison encourages observation and interest.

The elementary form of *U* tube is of little use for the type of continuous test envisaged. Reference to the curve in Fig. 116

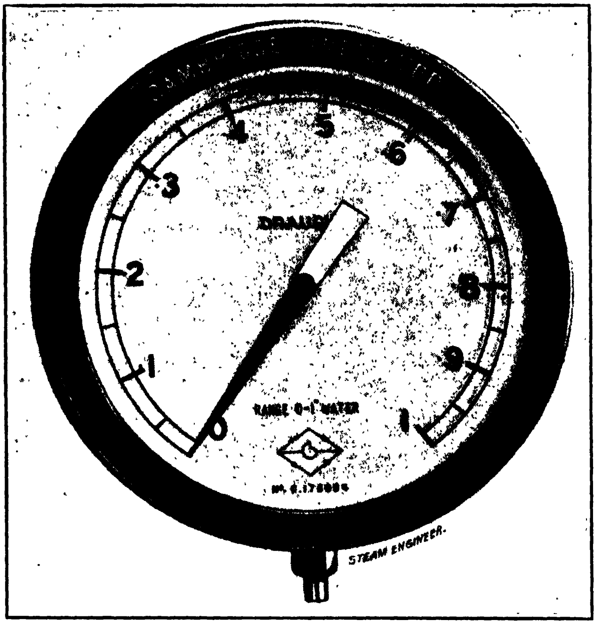


FIG. 117. CAMBRIDGE DIAL TYPE DRAUGHT GAUGE
(*Cambridge Instrument Co.*)

shows the very considerable variation in gas flow required to give a difference of only 0·1 in. in the reading, so that the limitations of the *U* tube are quite clear. It is however of some use for snap tests, and also to ascertain the limits of range for a particular working position, before ordering a more sensitive appliance. For commercial working in permanent positions, it is more usual to fit one or other of the instruments illustrated in Figs. 117 and 118. Both these types, dial or quadrant, have their adherents among engineers. In large installations, where more than one draught gauge is necessary on a single boiler, the quadrant type is often favoured because less space is taken up on what may already be a crowded panel. Also, it is possible so to arrange the ranges that, when working conditions are correct the indicating fingers form one straight line across the faces of the instruments, so that deviations are at once seen.

The single-gauge installation, described in detail, is applicable to boilers with induced or natural draught, an arrangement which is possibly more common than any other in industry.

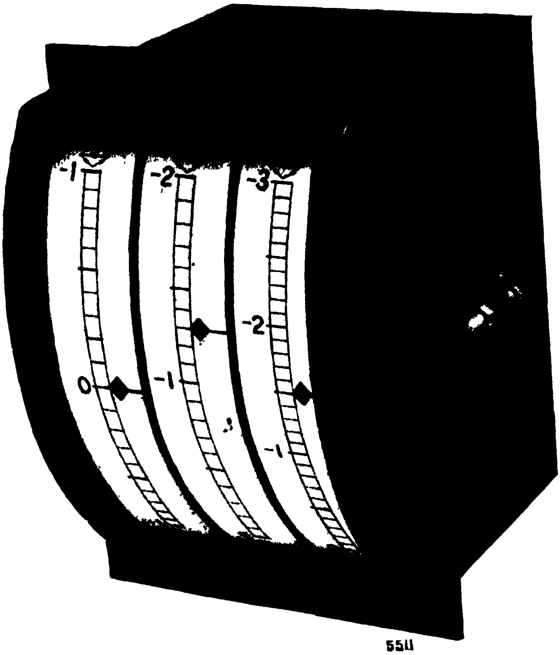


FIG. 118. CAMBRIDGE QUADRANT TYPE DRAUGHT GAUGES
(*Cambridge Instrument Co.*)

Where the boilers are fitted with forced or balanced draught, at least two gauges are required, one reading the total draught at the chimney base, and the other, usually a centre-zero instrument, reading the draught over the fires. For Lancashire boilers and similar two-flued arrangements, the practice is to provide a centre-zero gauge over each of the fires, and another instrument reading the total draught at the chimney base. In this case, where both forced and induced draught are fitted, a perfect balance can be achieved for every load. The fans or dampers are adjusted so as to give the slightest possible suction over the fires when the induced draught is providing the appropriate total draught for the prevailing conditions.

TABLE IX

TABLE OF VOLUMES AND WEIGHTS OF AIR AT ATMOSPHERIC PRESSURE AND VARIOUS TEMPERATURES

Temperature, degrees F	Volume, cu ft per lb	Weight, lb per cu ft
32	12·40	0·0807
50	12·85	0·0778
55	12·98	0·0771
60	13·10	0·0763
65	13·23	0·0756
70	13·35	0·0749
75	13·48	0·0742
80	13·61	0·0735
85	13·73	0·0728
90	13·86	0·0722
95	13·98	0·0715
100	14·11	0·0709
110	14·36	0·0696
120	14·61	0·0684
130	14·87	0·0672
140	15·12	0·0662
150	15·37	0·0651
160	15·62	0·0640
170	15·87	0·0630
180	16·13	0·0620
190	16·38	0·0611
200	16·63	0·0601
210	16·88	0·0592
212	16·93	0·0591
220	17·13	0·0584
230	17·39	0·0575
240	17·64	0·0567
250	17·89	0·0559
260	18·14	0·0551
270	18·39	0·0544
280	18·65	0·0536
290	18·90	0·0529
300	19·15	0·0522
320	19·65	0·0509
340	20·16	0·0496
360	20·66	0·0484
380	21·17	0·0472
400	21·67	0·0462
425	22·30	0·0448
450	22·93	0·0436
475	23·56	0·0424
500	24·19	0·0413
525	24·82	0·0403
550	25·45	0·0393
575	26·08	0·0383
600	26·72	0·0374
650	27·98	0·0357
700	29·24	0·0342
750	30·50	0·0328
800	31·76	0·0315
850	33·02	0·0303

CHAPTER IX

PRIME MOVERS AND PROCESS STEAM

ALTHOUGH the benefits to be derived from the utilization of the exhaust steam from prime movers, either reciprocating engine or turbine, have been known for many years, there still seems to be considerable reluctance on the part of managements to adopt the back pressure or pass-out systems, even when the particular industry is such that very great economies could not fail to be realized.

Some of this reluctance is due to widespread misconception as to the pressures necessary to carry out various processes, a misconception which leads very naturally to hesitancy and unwillingness to alter methods which may have proved successful and trouble-free over a period of years. Some difficulty is also introduced by lack of appreciation of the influence of a drop in steam pressure upon steam volumes, and of the fact that a drop in pressure must be accompanied by an appropriate

TABLE X

SUITABLE STEAM PRESSURES FOR VARIOUS INDUSTRIES

	Lb per sq in.		*Lb per sq in.*
Baths	5	Margarine works	7
Brick works	2 to 5	Nitrate oficinas	40 to 60
Calico printing	7 to 15	Oil refineries	2
Cement works	5 to 10	Pottery works	3 to 5
Chemical works	10 to 70	Soap works	5
Clay works	2 to 5	Starch works	10
Cotton mills	2 to 20	Sugar estates	10 to 15
Distilleries	7 to 30	Sugar refineries	15 to 100
Flax spinning	35	Tanneries	3 to 10
Gas works	7 to 10	Woollen mills	5 to 30
Gelatine works	5 to 7		

enlargement of process pipes, in order that the same *weight* of steam per hour may flow to the point of utilization.

Choice of Pressure

It is not unknown for managements to drop pressure experimentally (without the necessary pipe alteration), and to base their findings as to the possibility of using low pressure steam upon a procedure doomed to failure from the start. For this reason perhaps more than any other, it is possible to find in factories, even to-day, ovens in which the drying temperature is no more than 180° F, with ridiculously small heating surfaces, and supplied with steam at pressures of 50, 60, and even 100 lb per sq in. Worse still, open boiling vats are still supplied with steam at similar pressures, notwithstanding the fact that their temperature cannot exceed 212° F.

It is difficult to imagine anything more wasteful, especially when the boiling is done by direct contact, in which case large quantities of practically uncondensed steam bubble straight up to the surface as the boiling point is reached, giving up their heat content uselessly to the atmosphere.

Using the methods outlined in Chapter V, consider for a moment the necessary alterations.

A drying oven situated 100 yd from the boilers requires 350 lb of steam per hour, which is supplied at 100 lb per sq in. through a 1 in. bore pipe.

Starting with the fundamental Box formula,

$$d = \sqrt[5]{\frac{C^2 \times L}{H}} \div 3{\cdot}7,$$

with symbols as in Chapter V, we can transpose to get the pressure drop,

$$H = \frac{C^2 \times L}{(3{\cdot}7d)^5}$$

Reference to the steam Tables gives the volume of steam at 100 lb per sq in. saturated as 3·88 cu ft per lb, so that the flow is $\frac{3{\cdot}88 \times 350}{60} = 22{\cdot}6$ cu ft per minute.

Then $$H = \frac{22{\cdot}6^2 \times 100}{3{\cdot}7^5} = \frac{52{,}000}{690}$$

$$H = 75{\cdot}5 \text{ inches of water.}$$

But 1 lb per sq in. equals 27 in. of water, so the pressure drop from end to end of the pipe, when the full quantity is being carried, is 2·8 lb.

It is decided to supply this process with steam at 5 lb per sq in. The pressure drop of 2·8 lb, negligible for an inlet pressure of 100 lb, could not be permitted at the proposed low pressure, and a drop of not more than 0·5 lb, or 13·5 in. WG, is all that could be allowed.

But a comparison of the latent heat of steam at the two pressures shows at once the first economy, by the simple reduction of the pressure alone. The latent heat of steam at 100 lb per sq in. is 879 B.Th.U. per lb, as compared with

960 B.Th.U. per lb for the lower pressure. Therefore, the weight of steam required to do the same amount of work at the reduced pressure may at once be corrected in the ratio of the latent heats.

$$\text{Weight required} = \frac{350 \times 879}{960} = 320 \text{ lb per hour.}$$

From the original form of the Box formula, and using the new pressure drop, with the specific volume 20·8 cu ft per lb—

$$d = \sqrt[5]{\frac{(20{\cdot}8 \times 5{\cdot}3)^2 \times 100}{13{\cdot}5}} \div 3{\cdot}7$$

$$= \sqrt[5]{90{,}000} \div 3{\cdot}7$$

$$= \frac{9{\cdot}75}{3{\cdot}7} = 2{\cdot}64 \text{ in.}$$

It would pay in this case to put in the nearest size larger, which would mean a 3 in. pipe.

It has been shown that the use of steam at low pressures results in an increased availability of latent heat, so that it may be stated as an axiom that, for process work, the lowest practicable pressures should be used. The governing criteria should be the temperature actually necessary efficiently to carry out the process, and the employment of a reasonably economical pipe size.

Where it is found that the processes will in fact permit the use of low pressure steam, the possibility of generating at a higher pressure, and producing power by first passing through some form of prime mover, exhausting to the process pressure, immediately becomes attractive. Because the work which can be done in an engine by one lb of steam is a function of the heat drop from inlet to exhaust, it follows that for any given inlet condition, the lower the exhaust pressure the greater the output from the engine from a given weight of steam.

Careful consideration must therefore be given to every case, both with regard to the power required, and the steam available.

Comparison of Turbines and Reciprocating Engines

Where the power demand is small in relation to the quantities of process steam, it may be permissible or even desirable to select a higher exhaust pressure, so that all or nearly all the steam may be passed through the engine while working at an economical load for the greatest possible proportion of the working time. This may, in certain cases, be preferable to the alternative of running the engine to a lower back pressure, therefore with less steam, and supplying the greater part of the process demand with live steam reduced through a valve from the boilers.

The steam state at exhaust, which may be of the utmost importance, is influenced both by the heat drop through the engine and by the inlet pressure and condition. Where the boilers are already installed, the inlet pressure may be fixed by the permissible pressure which can be carried, so that the heat drop through the engine, and therefore the work available, may well be limited by some necessary exhaust condition, not perhaps of pressure but of dryness. It is common practice to run back pressure or pass-out reciprocating engines, with non-lubricated cylinders, so that the steam shall not be contaminated with oil, and this precludes the use of all but low degrees of superheat. It is possible to run with superheat, with lubricated cylinders, by fitting an efficient oil separator in the exhaust line, and this is very frequently done, but where foodstuffs are treated by direct contact with the steam, even this is not permissible.

With turbines there is no such limitation, and even when the pressure of existing boilers cannot be raised from considerations of safety, it is often possible to provide for the desired quality of steam at exhaust or pass-out point, by arranging the necessary superheat at inlet. Reheating the exhaust is sometimes resorted to, but it is not common practice.

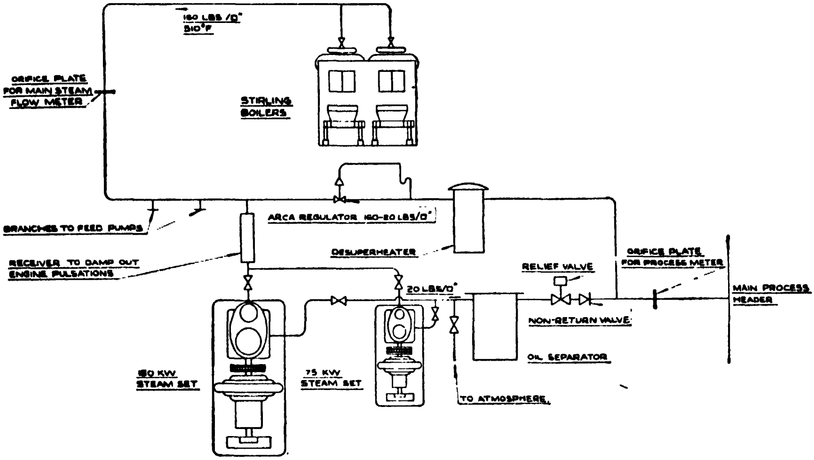
Back Pressure Sets

Where the power required is large in proportion to the steam demand, consideration should first be given to the possibility of lowering the process pressure to the point at which the engine can supply all the power needs from the steam available. If this cannot be done, either a pass-out set arranged to supply the process needs from some point in the steam cycle before exhaust, and passing the excess power steam to condenser, or a small back pressure set, making full use of the process steam, may be installed, the excess power in the latter case being taken care of by a separate condensing set, or purchased.

When the process and power demands do not coincide, a similar scheme might be considered, but if the steam quantities are not large and the "out-of-balance periods" short, such elaboration might not be justified. With low exhaust pressures excess steam may be exhausted to atmosphere for short peak load, or sudden falls in process demand.

A simple schematic lay-out, in which provision is made for exhausting peak load steam to atmosphere, and for supplying make-up steam for heavy process demands or low engine loads, is shown in Fig. 119.

The economies resulting from the adoption of the system



ARRGT OF TWO VERTICAL ENGINES AS BACK PRESSURE SET

FIG. 119. SIMPLE SCHEMATIC DIAGRAM OF LAY-OUT FOR BACK PRESSURE ENGINE
("*Cheap Steam*")

may be very simply demonstrated by reference to the Mollier diagram (Fig. 120), and a comparison of two hypothetical engines.

Assume the inlet pressure to be 200 lb per sq in., absolute, and saturated. In the first case, the engine exhausts to a condenser. to a vacuum of 25 in. mercury, or $2\frac{1}{2}$ lb per sq in. absolute.

Select on the Mollier diagram the point at which the 200 lb pressure line intersects the saturation line. From this point, draw vertically downwards till the 2·5 lb pressure line is reached. The length of this line represents the heat drop which would result in an engine working on the ideal Rankine cycle with adiabatic expansion. By projecting horizontally to the total heat ordinate, the total heat of the steam at inlet and exhaust may be read off directly, and the heat drop obtained by subtraction.

Total heat at 200 lb absolute, saturated	1198	B.Th.U./lb
Total heat at 2·5 lb and 79·5 per cent dry	910	,,
Heat drop	288	B.Th.U./lb

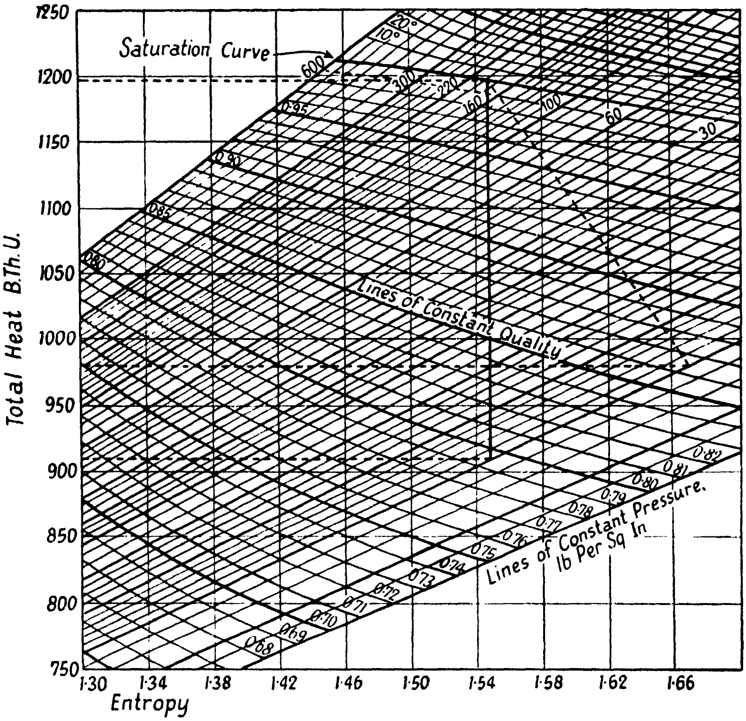


FIG. 120. MOLLIER DIAGRAM APPLIED TO CONDENSING ENGINE

The ideal cycle cannot, however, be realized in practice. First, the pressure actually applied to the piston at the commencement of the stroke must be some pressure less than that applied to the engine, by some difference necessary to ensure steady steam flow. As the inlet valves open during the admission period, there will be a further pressure drop, causing a slight fall in the inlet line, and a further rounding of the diagram will occur as the admission valves close, and again as the exhaust valves open. Also, the expansion after cut-off is not truly adiabatic, but complex in character, it being affected by condensation and re-evaporation from the cylinder walls at various parts of the cycle. The comparative indicator diagrams, ideal and actual, are shown superimposed in Fig. 121.

The actual is less in area than the ideal by some figure peculiar to the particular engine, and is influenced by the design features, to some extent by the size, and by the running conditions.

For the present purpose a purely arbitrary figure must be assumed. Say that the area of actual diagram of the engine

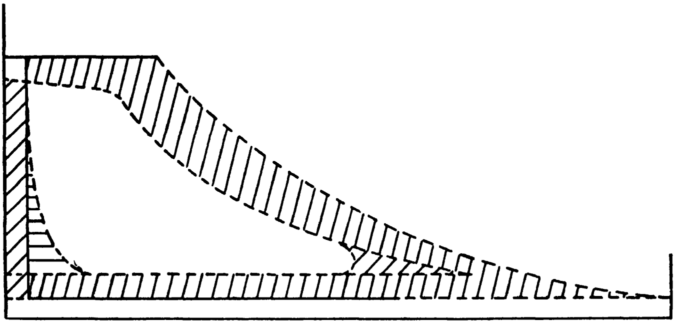


FIG. 121. COMPARISON OF ACTUAL WITH THE IDEAL INDICATOR DIAGRAM

The losses due to condensation leakage, wire-drawing, clearance compression, and incomplete expansion are indicated by the shaded areas.

under discussion is less than the ideal by 25 per cent. Set off from the lower end of the vertical line drawn on the Mollier diagram, a distance equal to 25 per cent of its length (or calculate 75 per cent of the adiabatic heat drop), and project horizontally to the ordinate as before. By projecting horizontally to the *right* of the vertical till the projection meets the 2·5 lb pressure line, the steam state at exhaust can be read off. The nearest state line to the point of intersection gives a dryness fraction of 0·866, so, put in another way, the steam at exhaust is 86·6 dry.

From the ordinate, or by calculation, the total heat at exhaust is 980 B.Th.U. per lb, and the heat drop in the actual engine is $1198 - 980 = 218$ B.Th.U.

By dividing this figure into the B.Th.U. equivalent of h.p., the weight of steam per indicated horse power may be obtained.

Thus $\quad \dfrac{2545}{218} = 11{\cdot}65$ lb steam per i.h.p. hour.

If it is desired to express the steam consumption in terms of brake h.p. the mechanical efficiency of the engine (say 0·86) must be brought into the calculation

$$\text{Steam per b.h.p. hour} = \frac{2545}{218 \times 0{\cdot}86} = 13{\cdot}6 \text{ lb.}$$

Similarly, where the engine is directly coupled to an electrical generator, the steam consumption per kWh may be expressed

$$\frac{3415}{218 \times 0{\cdot}86 \times 0{\cdot}90} = 20{\cdot}2 \text{ lb,}$$

where 3415 is the B.Th.U. equivalent of 1 kWh, and 0·90 the efficiency of the electrical generator.

This consumption might be reasonably expected from a small set, giving, say, 200 kW, so that at full load,

$$\text{Steam passed through engine} = 200 \times 20{\cdot}2 = 4040 \text{ lb per hour.}$$

Referring now to the figures obtained from the Mollier diagram,

Heat input . .	4040 × 1198 =	4,840,000 B.Th.U.
Rejected at Exhaust	4040 × 980 =	3,960,000 ,,
Difference		880,000 ,,
Converted to kW .	3415 × 200 =	683,000 ,,
Electrical and mechanical losses .		195,000 ,,

It is seen that by far the greatest loss is due to the heat rejected from the engine exhaust, and that this amounts to about 82 per cent of the heat input. The other losses, electrical and mechanical, are very small indeed. The rejected heat is destroyed in the condenser, and cannot be used for process or any other purpose. Moreover, additional power is required to drive the circulating or other pumps which supply the necessary cooling water. The point to be emphasized is that, although the provision of a condenser enables more work to be performed by each pound of steam passing through the engine, this additional work can only be extracted by the sacrifice of latent heat. It has already been demonstrated that the latent heat forms by far the greater proportion of the total heat in every pound of steam, so that any system which involves its destruction must necessarily work at very low thermal efficiencies. The Sankey diagram in Fig. 122 shows clearly the proportions of the used to the rejected heat.

Assume now an engine similar to the first example, taking steam at 200 lb per sq in. absolute, saturated as before, but

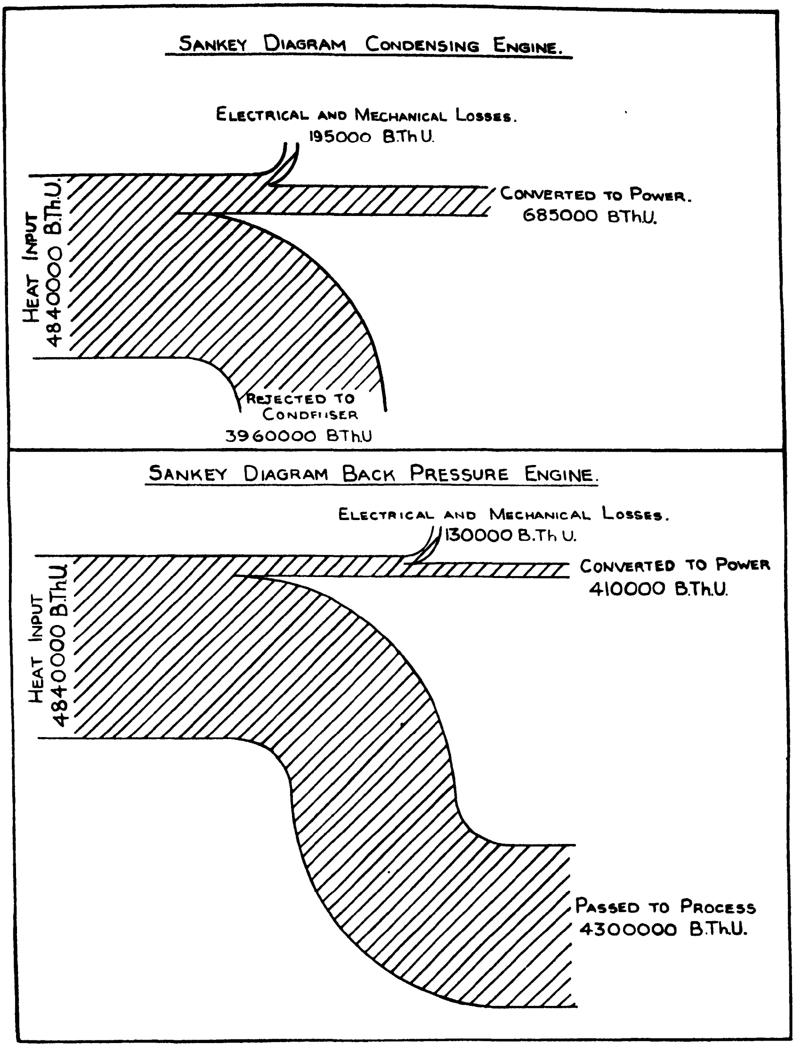


FIG. 122. COMPARATIVE SANKEY DIAGRAMS FOR CONDENSING AND BACK PRESSURE ENGINES

instead of exhausting to a condenser, the whole of the exhaust steam is used in the factory at a pressure of 5 lb per sq in. gauge (20 lb absolute).

The construction on the Mollier diagram is exactly the same as before, and for the sake of comparison we may assume the same deviation of the actual from the ideal cycle.

Then

Total heat of steam at 200 lb absolute, saturated	1198 B.Th.U.
Total heat of steam at 20 lb absolute, and 86·7 dry	1026 ,,
Adiabatic heat drop	172 ,,
But for actual cycle, total heat at inlet .	1198 ,,
Total heat at exhaust steam is 91 per cent dry	1066 ,,
Heat drop	132 ,,

Then steam consumption per kWh will be

$$\frac{3415}{132 \times 0{\cdot}86 \times 0{\cdot}90} = 33{\cdot}4 \text{ lb}$$

On the basis of the 4040 lb of steam per hour taken by the first engine, the present example would produce 120 kW.

Then

Heat input .	4040 × 1198 =	4,840,000 B.Th.U./lb
Passed to process	4040 × 1066 =	4,300,000 ,,
Difference		540,000 ,,
Converted to kW	3415 × 120 =	410,000 ,,
Electrical and mechanical losses .		130,000 ,,

Compare this with the previous result. The engine, it is true, produces less power for the same steam throughput, but because the exhaust steam is available for further use in the factory, the overall efficiency of the installation is very much greater. Up to the engine exhaust, the only losses are very small, electrical and mechanical, amounting in heat equivalent to about 3 per cent of the heat input. The power is, in fact, produced as a complete by-product of the process steam demand. This is the best possible case, and during periods of steady demand, such that the engine can work at or near its full load, while supplying the factory with all the steam it needs, the power is produced for the cost of the heat drop through the engine.

If the boilers are producing 7 lb of steam per lb of coal, with coal at 40s. per ton (0·214d. per lb),

7 × 1198 B.Th.U. cost 0·214d.
8386 B.Th.U. cost 0·214d.

The difference between the heat input to the engine and the heat passed to process must all be debited against power, so

$$\text{cost of power, 120 kWh} = \frac{0{\cdot}214\text{d.} \times 540{,}000}{8386}$$
$$= 12{\cdot}9\text{d.}$$
$$= 0{\cdot}1075\text{d. per kWh.}$$

Contrast this with the engine passing all its steam to condenser, and consuming 20·2 lb steam per kWh. In this case all

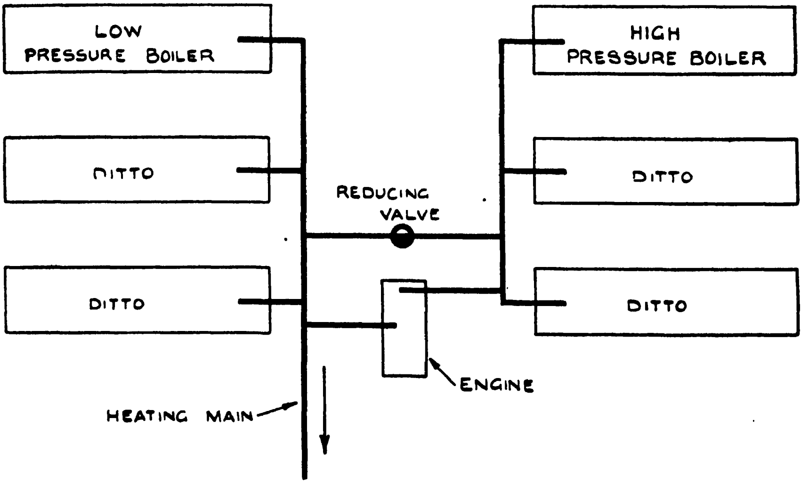


FIG. 123. HIGH- AND LOW-PRESSURE BOILERS IN CONJUNCTION WITH BACK PRESSURE ENGINE
(Bellis & Morcom, Birmingham)

the steam produced must be debited against power, since none is used for process, and the cost becomes,

$$\frac{20{\cdot}2 \times 0{\cdot}214}{7} = 0{\cdot}62\text{d. per kWh.}$$

The figure given is perfectly attainable in practice, where the process steam demands are at all times in excess of the steam required for power, that is to say, when the whole of the exhaust steam can be absorbed in the factory, but should it be necessary at any time to exhaust to atmosphere, the value of the steam so wasted must be costed against power. Under certain circumstances it is possible to employ steam accumulators to store this steam, or the arrangement shown in Fig. 123 may be advantageous. Here, there are two sets of boilers, one high pressure, the other working at the exhaust pressure. The engine exhaust is directly connected to the low-pressure boilers, the dampers of which are regulated to maintain that pressure correctly.

Although, as has been shown, the power obtainable from a given steam quantity increases with the pressure difference between inlet and exhaust, and low exhaust pressures are desirable on this account, it does not follow that because the

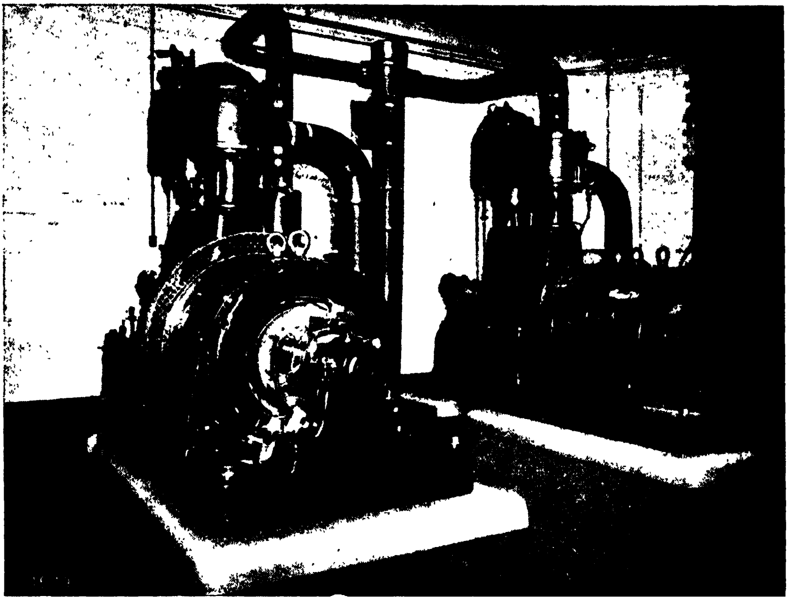


FIG. 124. TYPICAL BACK PRESSURE ENGINE SETS
(Bellis & Morcom, Birmingham)

process temperatures may fix the lower limit of pressure to some figure higher than those so far discussed, the use of back pressure or extracted steam is precluded.

There are instances of engines by well-known makers working with inlet pressures of 350 lb per sq in., and exhausting to 190 lb. Table XI gives the approximate power in b.h.p. which may be obtained from 10,000 lb of steam per hour at various inlet and exhaust pressures.

Typical Back Pressure Set

A type of engine used for back pressure work is illustrated in Fig. 124. It is totally enclosed, and force lubricated to all parts

TABLE XI—APPROXIMATE B.H.P. OBTAINABLE FROM RECIPROCATING ENGINES (E.) AND TURBINES (T.) PASSING 10,000 LB OF DRY STEAM PER HOUR AT VARIOUS INLET AND EXHAUST PRESSURES

	Vacuum															
Exhaust pressure*	20 in.	15 in.	10 in.	5 in.	0	5	10	20	30	40	50	60	70	80	100	125
Temp. degrees F, dry	161	179	192	203	212	228	240	259	274	287	298	308	316	324	338	353
Total heat†	1129	1137	1143	1147	1151	1157	1161	1169	1174	1178	1182	1185	1187	1189	1193	1197
Latent heat	1000	990	983	976	971	962	954	941	931	922	915	908	901	895	885	872
Cu ft per lb	74·86	51·22	39·1	31·74	26·77	20·4	16·4	12·0	9·4	7·8	6·6	5·8	5·2	4·7	3·9	3·25
Inlet pressure, lb; gauge pressure, lb																
100 E.	440	425	410	395	380	330	280	225	175	130	—	—	—	—	—	—
100 T.	575	470	420	365	315	265	220	165	125	85	50	—	—	—	—	—
125 E.	485	470	450	435	420	380	330	260	215	175	140	—	—	—	—	—
125 T.	585	490	440	390	340	295	250	195	150	110	80	50	—	—	—	—
150 E.	505	490	470	455	440	410	370	310	250	210	170	145	125	—	—	—
150 T.	600	510	460	420	370	330	285	225	175	140	105	80	50	—	—	—
175 E.	530	510	490	475	460	430	400	340	280	235	205	180	150	130	—	—
175 T.	615	530	475	435	390	345	300	245	200	165	130	105	80	—	—	—
200 E.	540	520	500	485	470	445	420	365	320	275	235	205	180	155	120	—
200 T.	635	545	495	450	410	360	315	265	225	190	160	130	110	90	—	—
225 E.	550	530	510	495	480	455	430	380	340	305	265	225	205	180	140	105
225 T.	650	560	510	470	430	385	330	285	245	205	175	140	125	105	—	—
250 E.	—	—	—	—	490	465	440	390	360	325	285	250	225	200	160	120
250 T.	660	575	525	485	445	400	350	300	260	220	190	160	140	120	—	—
300 E.	—	—	—	—	—	—	460	420	385	355	325	285	255	225	190	155
300 T.	675	600	550	510	475	430	390	330	285	245	215	180	160	135	—	—
350 E.	—	—	—	—	—	—	480	440	405	375	345	310	275	250	215	180
350 T.	700	625	580	530	495	450	410	345	305	265	235	205	185	160	—	—
400 E.	—	—	—	—	—	—	—	460	430	400	370	330	300	270	235	200
400 T.	720	650	600	550	510	470	425	365	320	285	255	230	210	180	140	—

* Inches Hg and gauge pressure, lb. † B.Th.U. per lb of dry steam from 32° F.

Approximate increase of power by superheating: 100° 17½ per cent, 150° 25 per cent.

The b.h.p. figures apply to one unit requiring the amount of steam stated (10,000 lb. per hour) at full load, the power plant being of a type and size best suited to the particular inlet and exhaust conditions.

If two sets were installed for the same total quantity of steam, i.e. 5000 lb per hour each, the collective output would be about 5 per cent less than stated above, due to their lower thermal efficiency. The reverse would, of course, apply to units requiring more than 10,000 lb per hour.

except the cylinders, which work without oil, the engine running on saturated steam. The exhaust steam is therefore entirely oil-free, and may be used for all purposes, even for direct contact heating, without fear of contaminating the product. Operation of these engines is simplicity itself, and the system is perfectly stable.

Fluctuations of steam demand are taken up by the governor, and do not cause hunting. Should the exhaust pressure fall, due to a sudden call for process steam, or a falling off of the electrical load, the make-up required to maintain the pressure is supplied direct from the boilers through a reducing valve. It is necessary that this valve should be particularly responsive to pressure changes on the outlet side, and it is usually of the Arca relay type, illustrated in Fig. 125. When for any reason the engine is shut down, all the process steam is supplied through this valve, which must therefore be designed for the maximum capacity required.

On the other hand, power steam in excess of that taken by the process is released to atmosphere (or accumulator) through a relief valve, set to a lb or two above the predetermined pressure. Thus the whole system, in its simplest form, is very largely automatic, only the electrical load needing to be regulated by hand.

Where large "out-of-balance" exists, or where the power steam is constantly in excess of the process demand, either a steam extraction set, or one condensing and one back pressure set, might be considered. The decision could only be made after detailed examination of all the factors involved, particularly with regard to the pressure at exhaust, the pressure and steam state at inlet, and the proportion of the total to the pass-out steam.

In effect, an extraction engine is a compound engine, in which steam is withdrawn from a receiver between the high- and low-pressure cylinders. Because of the starving of the low-pressure cylinder by this withdrawal, the cut-off is frequently so small that the low pressure side is not worked to its full capacity. Also, a decrease in the receiver pressure leads to increased piston loading on the high-pressure side, and a corresponding decrease in the work done by the low-pressure engine, so that the load distribution between the cylinders for all conditions becomes a matter of some difficulty. Cylinder ratios are higher in extraction engines than in normal compounds, so as to make as much use as possible of the small quantities of steam available to the low-pressure side, after extraction. It will be realized, however, that although such an

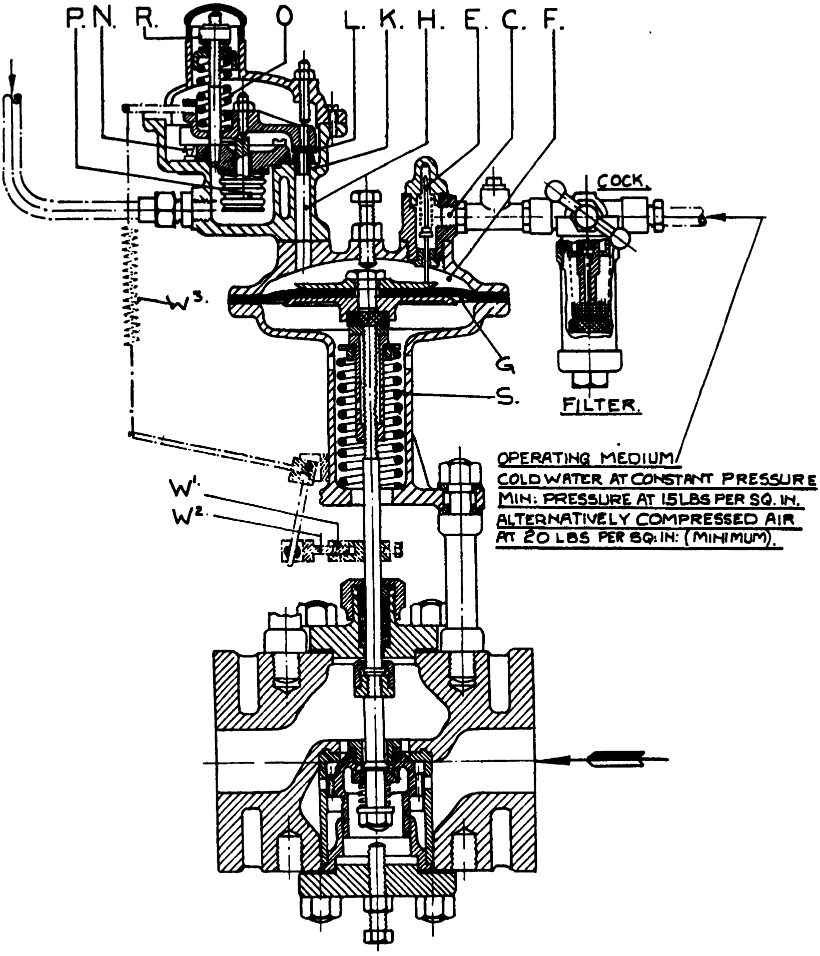


Fig. 125. Arca Regulator for Supplying Live Make-up Steam to an Exhaust System
(*Arca Regulator Co.*)

engine can be designed to run perfectly well and smoothly at one particular set of conditions, any departure from those conditions might tend to rough-running due to maldistribution of piston loading. For this reason the tandem engine is rather better, fundamentally, than the cross compound for extraction conditions, and its running is somewhat less affected by fluctuations of load and extraction quantities.

The steam turbine suffers from none of these disabilities, and for this reason, as well as for its inherent simplicity, it is frequently chosen, in spite of the fact that the reciprocating engine makes rather better use of the steam in the high pressure stages. Where process steam at two pressures is required, a simple arrangement can be made, so that the turbine can provide extracted steam at the higher pressure, and exhaust to the lower. Alternatively, two extraction points can be provided, any remaining steam necessary to produce extra power being exhausted to condenser.

Pass-out Sets

For critical pass-out pressures, nozzle or diaphragm control through the medium of oil-operated relays can be provided to keep the pass-out pressure constant within fairly wide limits of variation of steam demand or mechanical load. The oil cylinders of the relays can be seen at the side of the set in Fig. 126.

Where the pass-out pressures may be allowed to vary, or where both the steam demand and the load on the set may be kept steady within reasonable limits, a simple "bleeder" turbine may be put forward. The arrangement is frequently favoured where possible, on account of its simplicity and lower cost as compared with an extraction machine with controlled pressures. Steam is bled away at the appropriate stage of the turbine, and the pressure available varies both with the load and the pass-out demand. The limits of the pressure obtainable throughout the full range of steam demand and load are determined by the stage from which the steam is extracted, and the design characteristics of the machine. Usually, some modification is necessary to the design as compared with that which would be put forward for a straight condensing machine, the actual form of the modification being influenced by the pressure drop required between the inlet, and the extracted pressure.

A pure impulse wheel may be provided in the first stage, instead of the more usual velocity wheel, as in the set shown in Fig. 127.

This set is a 225 kW geared turbo-generator, provided with a simple "bleeder" connection after the first wheel. There are

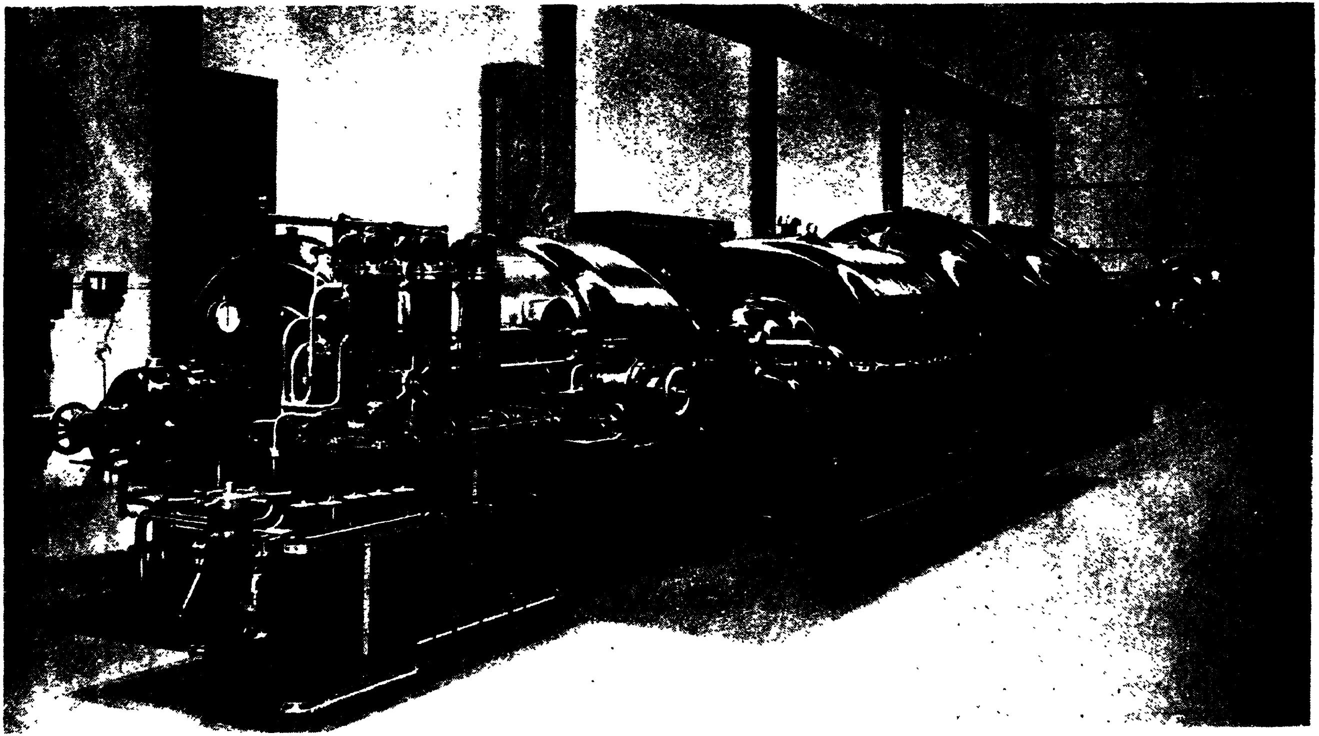


Fig. 126. Typical Pass-out Turbo Set Showing Relays
(Fraser & Chalmers, Erith, Kent)

eight stages in all. There is no automatic control of the pass-out pressure, which therefore fluctuates with steam demand and load. When bleeding at 2000 lb of steam per hour, the pass-out pressure varies between about 12 lb per sq in. at 0·5 load, to 30 lb per sq in. at full load.

Heat Cycles

In the same way as the indicator diagram of the reciprocating engine is less in area than the hypothetical diagram of the ideal

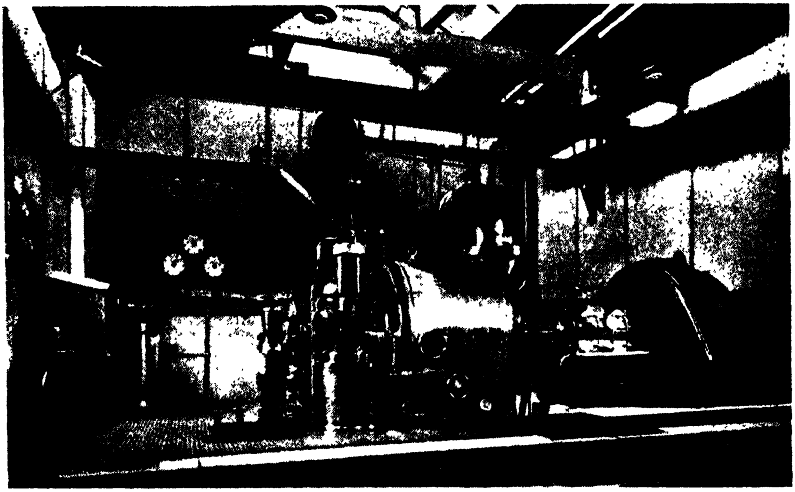


FIG. 127. SIMPLE "BLEEDER" TURBO SET
(D.P. Battery Co., Bakewell)

Rankine cycle, so the expansion of steam through a turbine gives up less heat than the quantity calculated from adiabatic expansion. The steam discharge from any stage is always drier in quality than would result from a pure adiabatic expansion through the same stage. Because of frictional leakage, vane, and other losses, the steam is said to be reheated. The amount of reheat is influenced by the steam condition at inlet, and by the stage efficiency. This is an important factor in blade design, since the quality of the steam admitted to each successive stage is affected, but for the moment an overall stage efficiency may be assumed for the calculation of the thermal figures for a turbine to supply steam at two pressures to a factory, as well as the electrical load.

The factory requires 4000 lb of steam per hour, at 100 lb per sq in. gauge, 15,000 lb per hour at 15 lb per sq in. gauge, and 1000 kW of electrical load. The electrical and steam loads are not fully coincident, and the set is exhausted to a condenser at 28 in. Hg (1 lb absolute, approx.). The existing boilers are built for 325 lb per sq in. and 150° F superheat.

Use of Mollier Diagram for Construction of the Turbine "Condition Line"

Turn now to the Mollier diagram in Fig. 128. We may assume that at full load the throttle is fully open, so that for this calculation the pressure drop at constant total heat across the throttle may be neglected. The point on the diagram corresponding to 340 lb absolute, and 150° superheat, may therefore be located at once, and the adiabatic expansion line drawn vertically to the exhaust pressure line.

The total heat of the steam at inlet is shown to be 1300 B.Th.U. per lb, and at exhaust, 885 B.Th.U. The adiabatic heat drop through the machine is, therefore, 1300 — 855 = 445 B.Th.U. per lb.

Assume an eight-stage machine, with equal heat drop per stage, then $\frac{445}{8} = 55{\cdot}5$ B.Th.U., is the stage heat drop

From the total heat scale, set off in dividers, the distance equal to the stage heat drop, and transfer to the adiabatic line, starting at the inlet condition point. The new position so located gives the pressure, but not the condition, of the steam at the exhaust of the first stage. Assume a constant stage efficiency throughout the machine of 0·65. Set off from the total heat scale as before, a distance equal to $55{\cdot}5 \times 0{\cdot}65 = 36{\cdot}1$ B.Th.U., and transfer to the adiabatic line. From this point, project horizontally to intersect the stage exhaust pressure line. The intersection is the true condition point at the inlet to the next stage. Project vertically from the intersection a distance equal to the adiabatic heat drop of the next stage, set off 0·65, and project horizontally to the exhaust pressure line as before, thus locating the condition point for that stage.

Continue the procedure until at the last stage exhaust, the condition point should fall on the predetermined pressure line. A free curve drawn through all the condition points is the condition curve for the machine.

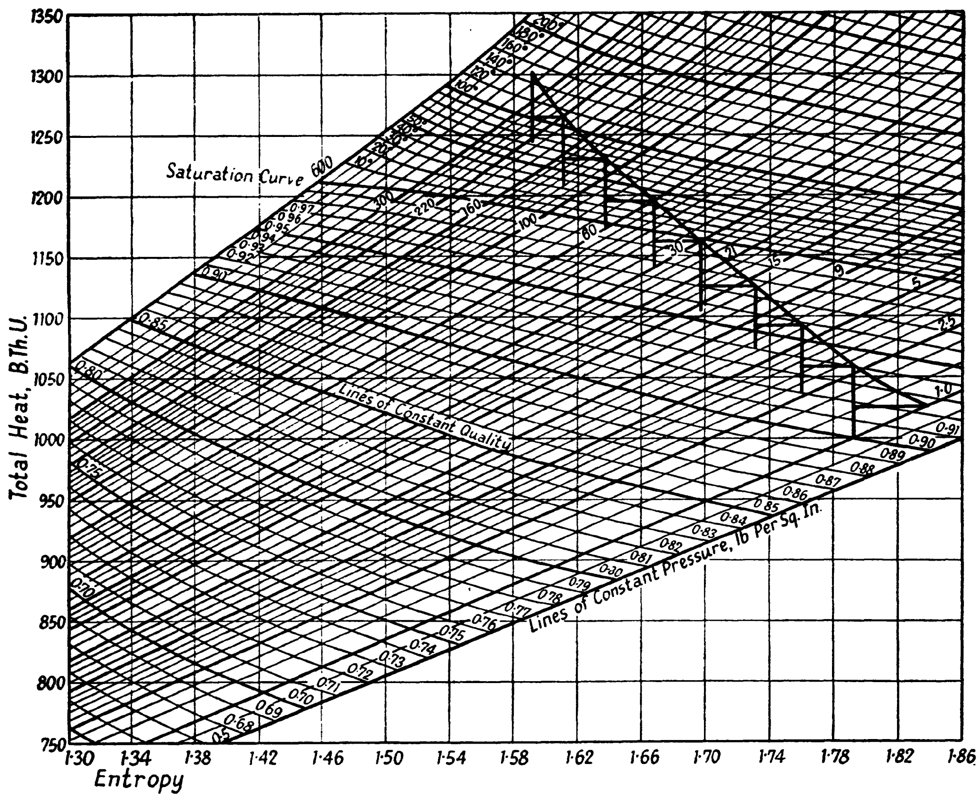


FIG. 128. MOLLIER DIAGRAM APPLIED TO HYPOTHETICAL MULTI-STAGE TURBINE

By inspection, it can be seen that the 115 lb absolute extraction point falls at the exhaust of the second stage, and that the steam still carries about 70° of superheat. The 30 lb absolute pressure line intersects the exhaust point of the fourth stage, and the steam is as nearly as possible dry-saturated.

Assessment of Steam Consumption

By treating the set as three separate turbines an assessment of the probable steam consumption of each section may be made, so that, because the extraction quantities are known, the excess steam to condenser, in order to obtain the required power output, may be computed.

Heat drop from inlet to second stage exhaust

$$1300 - 1230 = 70 \text{ B.Th.U.}$$

$$\text{Lb per kW} = \frac{3415}{HD \times Nt \times Ng}$$

where HD = heat drop
Nt = efficiency (mechanical) of turbine = 0·88
Ng = efficiency of generator = 0·92

$$\text{Then steam per kW} = \frac{3415}{70 \times 0{\cdot}88 \times 0{\cdot}92}$$
$$= 60 \text{ lb per kW.}$$

But 4000 lb of steam per hour are required from this extraction point, so that at full load $\frac{4000}{60} = 67$ kWh.

Similarly, at the lower extraction stage, the heat drop is

$$1300 - 1160 = 140 \text{ B.Th.U.}$$

$$\text{Lb steam per kWh} = \frac{3415}{140 \times 0{\cdot}88 \times 0{\cdot}92}$$
$$= 30 \text{ lb per kWh.}$$

But 15,000 lb of steam per hour are needed at this point, so that

$$\frac{15{,}000}{30} = 500 \text{ kWh are produced.}$$

Then a total of 567 kWh are produced by the extracted steam, leaving 433 to make up the required electrical load by steam passed to condenser.

The heat drop from inlet to the exhaust pressure is (from the condition curve) 1300 — 1025 = 275 B.Th.U.

$$\text{Lb steam per kWh} = \frac{3415}{275 \times 0{\cdot}88 \times 0{\cdot}92}$$
$$= 15{\cdot}4.$$

Then the steam to condenser = 15·4 × 433 = 6650 lb per hour.

Summarizing the above,

Heat input (4000 + 15,000 + 6650) × 1300	= 33,400,000	B.Th.U./hour	
Passed to process at 100 lb gauge, 4000 × 1230	= 4,920,000	,,	,,
Passed to process at 15 lb gauge, 15,000 × 1160	= 17,350,000	,,	,,
Total to process	= 22,270,000	,,	,,
Remainder after subtracting from input	= 11,130,000	,,	,,
Converted to kW, 3415 × 1000	= 3,415,000	,,	,,
Remainder	= 7,715,000	,,	,,
Passed to condenser, 6650 × 1025	= 6,825,000	,,	,,
Other losses	= 890,000	,,	,,

The Sankey diagram for the set is given in Fig. 129. The gains realized may be visualized by comparing the diagram of the pass-out set with that of the same or a similar set, running with the pass-out valves closed, and all steam passing to condenser.

In this case we may use the same condition line and the same heat drop, so that the steam consumption will be 15·4 lb per kWh, and when running on full load the steam throughout will be

$$1000 \times 15{\cdot}4 = 15{,}400 \text{ lb per hour.}$$

Without making a full analysis it may be said at once that of the total heat input of 15,400 × 1300 = 20,000,000, the only heat put to useful purpose is that directly converted to kWh. This, as has been demonstrated, accounts only for the comparatively small amount of 3,415,000 B.Th.U., and *all* the remainder is wasted in one form or another. But the point to be stressed is that only a very small proportion of this wasted heat can be classed as irrecoverable, as the examples of back pressure and pass-out sets have shown. When the almost limitless flexibility of design, made possible by the manipulation of inlet pressures and superheats on the one hand, and extraction pressures and machine loads on the other, is taken into account,

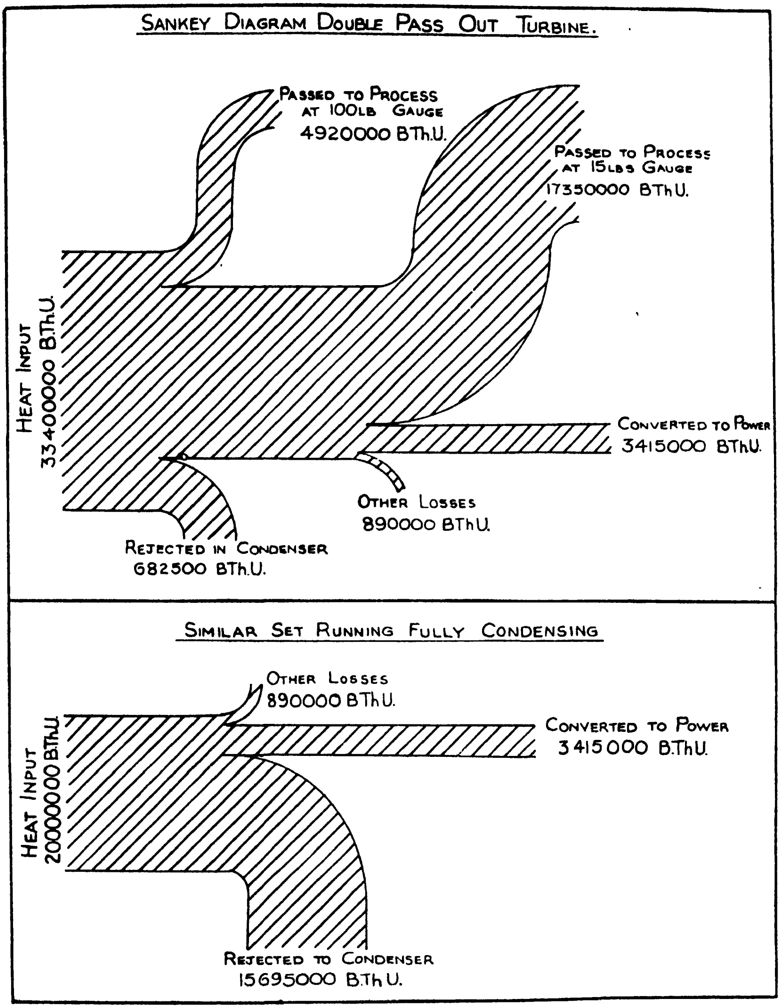


FIG. 129. SANKEY DIAGRAMS FOR DOUBLE PASS-OUT TURBINE, RUNNING PASS-OUT AND FULLY CONDENSING

there seems little excuse for the continued wastage of valuable latent heat. At all events, no new factory requiring quantities of steam for its processes can afford to ignore the possibilities. That quite small schemes can be successfully and economically worked is evidenced by the many small laundries, sometimes requiring no more than say 75 kW, or even less, of power. These little engines work with little more maintenance than an

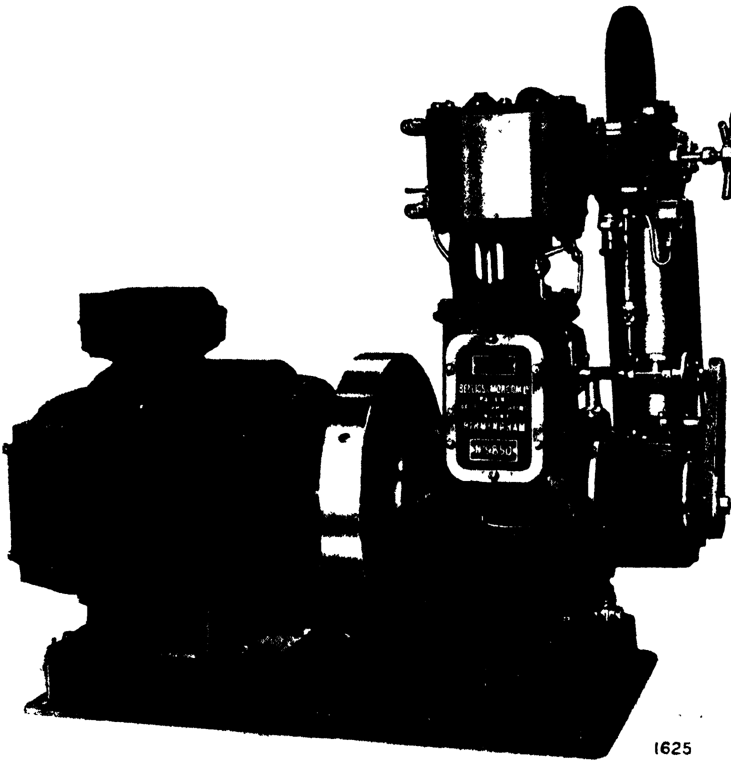


FIG. 130. SMALL SINGLE CYLINDER BACK PRESSURE ENGINE FOR HIGH BACK PRESSURES. "S" CLASS

(Bellis & Morcom, Birmingham)

electric motor, and certainly with no more than that required for other machines in the laundry. They may be designed single cylinder for quite high back pressures, and a great deal of scope is possible in defining the conditions under which they can be made to work. Such an engine is illustrated in Fig. 130.

Revert for a moment to the example of the double pass-out turbine already cited.

Without the pass-out system the factory would need to produce 19,000 lb of steam per hour at 100 lb per sq in. for process work (it is assumed that steam would be produced at the higher pressure and reduced to the 15 lb range), and 15,400 lb per hour for power, or its equivalent in purchased electricity. The total amounts to 34,400 lb per hour. With the pass-out set, the total steam required for *all* factory needs is 25,650 lb per hour, a saving of 8750 lb—the output of a 30 ft × 8 ft 6 in. Lancashire boiler.

Collection of Data

It must be realized that in all the above calculations no consideration has been given to partial load conditions, which affect materially the economics of the system considered over a period. Before putting forward a detailed proposition, the most accurate possible data must be gathered and plotted by the methods discussed in previous chapters, but in addition to this, the characteristics of the electrical (or mechanical) load on the proposed set must be studied as well. On the combined findings of the two investigations the final selection of the type of plant must rest.

While the general average relationship of the two loads (steam and electrical) may point broadly to, say, two types, only a detailed study of the duration and magnitude of the peaks, together with some consideration of the required process pressure, can assist the final determination as to whether the set may be allowed to exhaust to atmosphere, or whether the more elaborate pass-out equipment must be put in.

Again, if the pass-out set be chosen, the maximum and minimum figures for the bled steam affect the condenser design. If the steam demand will never fall below a predetermined figure the condenser may be made smaller, but if on the other hand the pass-out steam is likely at any time to fall to zero, then the set will at some time or other be working fully condensing, and provision must be made accordingly.

Nozzle areas and blading, in the case of turbines, are also affected.

The choice of reciprocating engine or turbine is not easy, and must depend to a great extent on particular conditions. For back pressure requirements, where the requisite powers and exhaust pressures can be obtained without superheat, the reciprocating engine is favoured as being somewhat more economical, at least for powers up to about 500 kW or slightly higher. Where the needs can only be met by the employment of superheat, necessitating lubricated cylinders, if a reciprocating engine were used, the turbine becomes obligatory if clean steam is vital at exhaust.

For pass-out conditions, reciprocating engines of the compound tandem or uniflow type are rather more satisfactory than cross compounds, for the reasons involving piston loading already given.

Cross compounds can be made to run satisfactorily while extracting steam, but must not be expected to work well where the extracted quantities fluctuate greatly. The turbine is generally somewhat cheaper than the reciprocating engine for extraction work, it is simple, and its running (as against its economy) is not adversely affected by fluctuation.

CHAPTER X

FEED WATER TREATMENT

IT may be said at once that feed water treatment is a specialist's business. The problems involved are so complex in themselves, and so varied in character, that it is not possible nor would it be desirable to deal with them in a work of this nature, except in the most general terms.

In recent years, the importance of feeding pure water to steam-raising plant has very greatly increased, largely by reason of the general trend of boiler design, and the demand for high efficiency self-contained units, capable of high outputs in small space. High output in small space usually means a tubular boiler of some description, either smoke or water tube, designed for high rates of heat transfer per unit area of surface.

Effect of Scale upon Boiler Heating Surfaces

Much has been written upon the effect of scale upon boiler efficiency. That it must have some effect is certain, but it is now established beyond reasonable doubt that the important factor is not so much the question of efficiency as the possibility of damage to the boiler. Take the case of a multi-tubular boiler as an instance. The highest rate of scale formation will take place in the hottest bank of tubes, so that as these scale up the heat which should be transferred through them becomes available for tubes in more remote regions of the boiler, away from the normal hot zone. As these in their turn scale up, there will be a tendency for the temperature of the flue gases to rise as the first indication of loss of efficiency, but long before that actually happens, rupture would almost certainly have occurred in the first (hot) tube bank.

This is true of boilers with complex heating surfaces, or where economizers and kindred apparatus are fitted. In the case of properly instrumented plant, some indication would be given long before actual loss of efficiency takes place.

Tubes exposed to the radiant heat of the furnace are particularly susceptible to the effects of scale, and invariably will give far more certain indication of scale trouble than any detectable loss of efficiency. The simple shell type boiler without additional heating surfaces of any kind, behaves rather differently. Here, when the main heating surfaces are scaled, a rise in the temperature of the flue gases and consequent loss of efficiency are quite likely to occur.

Calculation of Scale Thickness

With all types of boiler there must obviously be some loss, due to impaired heat transfer, but the point to be emphasized is that the importance of this loss is likely to be quite small as compared with the possibility of serious damage.

From a fundamental formula

$$H = \frac{k(t_1 - t_2)}{L}$$

where H = heat transfer rate in B.Th.U. per sq ft per hour
k = thermal conductivity in B.Th.U. per sq ft/hour/°F/in. thickness
t_1 = temperature on one side of conducting medium
t_2 = temperature on the other side of the medium
L = thickness of the conducting medium in in.

Consider a modern boiler working at 200 lb per sq in. gauge. In such a boiler the rate of heat transfer might well be 80,000 B.Th.U. per sq ft per hour locally. If the safe limit of metal temperature is taken as 900° F, and the thermal conductivity of the scale is 16 B.Th.U. per sq ft/hour/°F/in. we get

$$80{,}000 = \frac{16(900 - 388)}{L}$$

$$L = \frac{16 \times 512}{80{,}000} = 0{\cdot}1025 \text{ in.}$$

so that for the conditions given, for scale of the conductivity cited, a thickness of $\frac{1}{10}$ in. would be sufficient to introduce dangerous conditions.

In point of fact, the conductivity of scales varies greatly, from as high as 18 B.Th.U./sq ft/hour/°F/in. thickness, to as low as 0·6. Contrary to common belief, some of the porous scales fall into the lowest category, this being due to the fact that bubbles of steam are entrapped in the pores, forming a high resistance belt, immediately adjacent to the boiler plate or tube wall. The effect of scale formations on smoke tube boilers can be particularly distressing. Even with clean surfaces, the tube expansions into the back tube plate are working under extremely severe conditions, and the formation of scale fillets adhering to the tubes and plate at this point will inevitably cause rapid overheating and distortion, leading to leakage. Once leakage has started, only the cutting out and replacement of the tube or tubes concerned is an effective remedy. It is absolutely useless to attempt to re-expand the tube after scale carrying water has been forced by the boiler pressure between the tube and the tube plate.

A satisfactory job can only be made by thoroughly cleaning the seating in the tube plate before expanding in the new tube. Even with the conservatively rated Lancashire boiler, dangerous conditions can be set up by localized scale. Such a boiler designed for 10,000 lb of steam per hour would have a heating surface of about 1235 sq ft, giving an average rating of 8·1 lb of steam per sq ft of heating surface per hour.

But it has already been established in previous chapters, that about 60 per cent of the evaporation is carried out by about 15 per cent of the heating surface, i.e. the area immediately over the fires. Then 6000 lb per hour will be evaporated by 185 sq ft, a rating of 32·4 lb per sq ft per hour. Assume that the heat required by the steam is 1150 B.Th.U. per lb, then the heat transfer per sq ft per hour is $1150 \times 32{\cdot}4 = 37{,}000$ B.Th.U. per hour.

If the temperature of the water in the boiler is 355, and we assume as before that the limiting temperature of the metal is 900° F, and the conductivity of the scale 16 B.Th.U./sq ft/hour/°F/in. thickness,

$$L = \frac{16(900 - 355)}{37{,}000}$$

$$= \frac{16 \times 545}{37{,}000} = 0{\cdot}236 \text{ in.}$$

which is the thickness of scale sufficient to introduce dangerous conditions if situated over the furnace crowns. This means to say that a thickness of less than $\frac{1}{4}$ in. of certain types of scale may so increase the temperature of the metal of the furnace tubes that danger of collapse may be imminent.

Water treatment should therefore be considered from the safety standpoint, even if a case cannot be made out on grounds of increased efficiency.

Factors Governing Choice of System

Certainly, no modern boiler should be installed without due investigation of all the aspects of water treatment as they apply to the particular case. The type of plant or treatment to be used can be decided only after detailed scrutiny of all the factors involved.

These factors include—

1. Complete analysis of the water to be used.
2. Source of supply, and whether this is likely to be constant or variable, as affecting the quality of the water.
3. Proportion of returned condensate when the plant is working.
4. Type of boiler, whether fitted with economizer, etc.
5. Personnel available for the supervision and working of the plant.
6. Personnel available for chemical control if necessary, and facilities for routine testing.

Impurities of Water

When water falls to the earth in the form of rain it is affected by the atmosphere through which it passes. It may fall through the comparatively clean air of the open country, or through the smoke and sulphur-laden air of the industrial regions.

In any case, carbon dioxide and oxygen will be dissolved by the water in some degree, so that it will contain both carbonic acid and free oxygen, but in addition, the industrial areas may yield ammonia (NH_3), hydrochloric acid (HCl), sulphur dioxide (SO_2), and other forms of atmospheric pollution. Besides dissolving some soluble substances, the weak acids which the water has absorbed in its journey through the atmosphere dissolve various substances when the earth is reached, which would otherwise be insoluble. Limestone, for instance, insoluble in water, is dissolved by the carbonic acid, forming calcium bicarbonate ($Ca(HCO_3)_2$).

Upon heating the water, some of the dissolved substances may be thrown out of solution when boiling point is reached, or even before, while others may remain in solution at much higher temperatures and pressures, or until higher concentrations are obtained due to evaporation of the water. Thus, it is readily seen that certain constituents may be thrown down as deposits or precipitates in the hot well, feed lines, or economizer tubes, before even entering the boiler proper.

The probable behaviour of a particular water can only be predicted after analysis. It may be said that water contains both scale-forming constituents, and highly soluble substances which do not normally form scale. Of the latter, common salt (sodium chloride, NaCl) is the main component. During normal working and evaporation of the boiler water, the concentration of these soluble substances is increased appreciably, causing priming, foaming, and kindred troubles.

The chief scale-forming constituents are compounds of calcium, magnesium, and silica. Compounds of the first two are generally known as "hardness" of the water. Most waters possess both temporary and permanent hardness, the temporary being attributed to the carbonates, and the permanent to the sulphates.

Temporary and Permanent Hardness

The temporary hardness is formed by the bicarbonates of calcium and magnesium, which are relatively soluble, and which, upon heating of the water, decompose, forming insoluble carbonates, with release of carbon dioxide. This decomposition takes place at temperatures approximating to the atmospheric boiling point, hence the term "temporary hardness." The insoluble carbonates thus formed are thrown out of solution and are deposited on the metal surfaces as scale.

Of the permanent hardness, salty scale is formed by calcium sulphate only, magnesium sulphate being highly soluble. Calcium sulphate ($CaSO_4$) has the peculiar property of decreasing in solubility with increase of temperature. Owing to the higher temperature, and the concentrating effect of the evaporation of the water, calcium sulphate will normally be precipitated in the boiler only, where it forms a most troublesome, hard, close-grained scale, which is difficult to remove either by mechanical or chemical methods.

Finally, any silica in the feed will also be precipitated under the conditions met with in the boiler; silica scales are particularly dangerous, as their thermal conductivity is extremely low, very thin layers being sufficient to cause serious overheating and damage to the boiler.

Methods of Treatment

Broadly speaking, there are four general methods of water treatment—

1. The lime-soda method, with or without subsequent conditioning by phosphates, aluminates, etc.

2. The base exchange or zeolite method.
3. The demineralization method, which gives a product in every respect like distilled water.
4. Various forms of internal treatment.

Each of these processes has its own field of application.

Lime Soda

The lime soda is, properly speaking, a precipitation process, in which such chemicals are added in their correct quantities to the water as will throw the calcium and magnesium out, as compounds of low solubility. When precipitation is complete, the water is passed through settling tanks and filters. The precipitation is brought about by the addition, in precise quantities, of hydrated lime and sodium carbonate. An additional reagent, sodium aluminate, may be used, which is especially useful where magnesium compounds and silica are present. In this case, the magnesium hydroxide, which normally tends to remain in suspension, and the silica are coagulated by the addition of the aluminate, into a floc which settles rapidly. The floc has an additional useful function, in that it adsorbs other suspended matter during its formation, and so accelerates the clearing process of the water.

The temperature at which the process is carried out has some influence both on the settling time and the degree of final hardness which may be attained. If the water can be heated to a temperature of 160° F it is possible to produce a water whose final hardness is of the order of $1\frac{1}{2}$ to 2 parts of $CaCO_2$ per 100,000. The settling time is about two hours for the hot process, longer for the cold, and in the latter case the final hardness will not be less than 4 to 6 parts per 100,000.

The quantities of reagents to be added are calculated from the analysis of the water to be treated, and it is of the greatest importance to ensure the greatest possible precision in this respect. Over-dosage can produce in the boiler and auxiliaries an effect far worse than if no softening agents were used at all. Mechanical means are provided automatically to proportion the chemicals added to the quantities of incoming raw water. Fig. 131 illustrates a typical measuring and mixing apparatus.

The necessary chemicals, in this form of commercial plant, are dissolved in a measured quantity of water, in a tank at ground level. The proportions of the chemicals, hydrated lime and soda ash, and the water in which they are first mixed, are carefully determined primarily from analysis of the water but influenced in some degree by the particular plant, capacity of chemical valve, size of tippers, etc.

The lower tank is provided with a pump, and hand- or power-operated paddle gear. Usually, the pump and stirrers are interconnected, so that after the preliminary mixing, it is only necessary to open a valve for the pump to elevate the

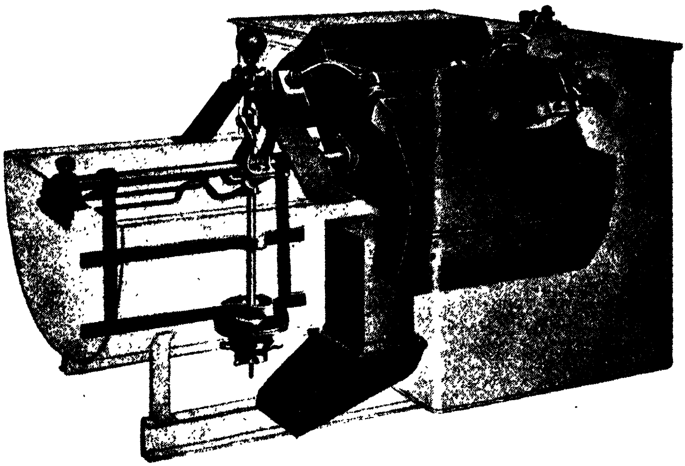


FIG. 131. MIXING GEAR OF LIME-SODA SOFTENING PLANT
(*Permutit Co. Ltd.*)

mixture to the chemical tank proper, while the stirring continues.

The chemical tank proper consists of a semi-circular section tank, provided with a set of paddles and a chemical valve, both operated by the tippler arrangement shown. The incoming water pours into the pivoted buckets, each of which is provided with a float, directly attached to a vertical moving plunger. As each bucket fills, the float moves higher and higher, moving the plunger with it until the top of the plunger pushes up the retaining trip lever, so that the bucket is free to fall by its own weight. The buckets, as they fall, turn the spindle, which actuates both the chemical valve and the paddle. The rate of tipping, that is to say, the number of times the buckets move in a given time, therefore depends on the rate of filling, and as the chemical valve is operated for each swing of the buckets, it follows that definite quantities of chemical are injected for known quantities of water.

The chemical valve is a specially designed double-beat device, so arranged that the amounts of chemical injected per stroke can be adjusted. Two variations are therefore possible. The valve, as explained above, may be used to increase or decrease the amount of chemical for a given water flow, that is to say, the total addition of mixture, while the proportions of the

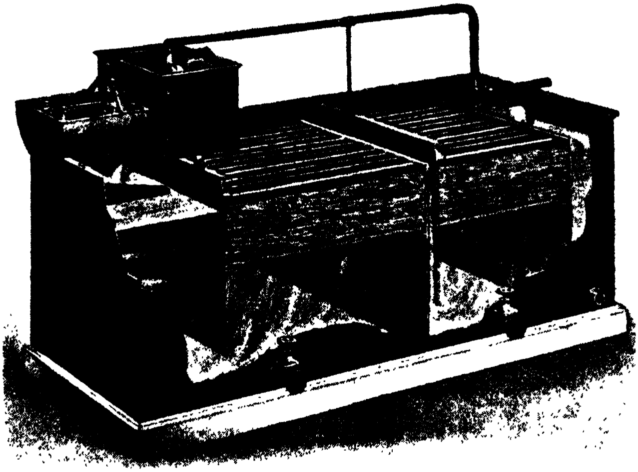


FIG. 132. COMPLETE LIME-SODA SOFTENING PLANT
(*Permutit Co. Ltd.*)

reagents forming the mixture may be altered to compensate for changes of water quality, which cannot be met by simple adjustment of quantity alone. The complete plant is shown in Fig. 132.

The water, after the addition of the chemicals, moves slowly through the tanks, settling in the early stages, and any suspended matter is afterwards removed in the large area filter beds, usually formed of wood, wool, or sand. Provision is made to clear the filter beds as required, and for the disposal of the sludge deposited in the tanks.

A more modern conception of the lime-soda process is the two-stage plant, in which there are two reaction tanks. This method is particularly valuable where *Mg* hardness forms an appreciable proportion of the total.

The water is first of all treated with a small excess of lime, so that the temporary hardness is decomposed, and the magnesium precipitated as hydroxide. The water is then passed to a second, quite separate, tank for treatment with soda for the removal of the calcium hardness.

The chemicals are fed to the reaction tanks through separate proportioning pumps, capable of individual control.

Corrections may, if necessary, be carried out between the stages, and very exact and reliable control is possible.

This method is much more scientific in conception than the older procedure in which the lime and soda are mixed together in the one tank, and many of the uncertainties and irregularities associated with the single-stage plant are eliminated.

The "Spiractor" Plant

A somewhat different interpretation of the lime-soda method is shown in Fig. 133. This is the "Spiractor" plant, produced by the Permutit Company. The principle involved is known as catalytic precipitation. A specially designed conical chamber is filled with a catalyst in granular form. At the bottom of the chamber, i.e. at the apex of the cone, are three inlet pipes, arranged to give tangential flow to the raw water and chemicals passed through them. The chemicals and water flow through separate pipes, so that mixing does not take place until the swirling motion produced by the tangential disposition of the pipes ensures that the most intimate contact is effected between the chemicals, the water, and the catalyst.

The effect of the catalyst is that it very much shortens the time required for the softening process, which, it is claimed, is completed in five minutes. As the water and chemicals flow upward through the conical chamber, the velocity of flow is progressively decreased by reason of the gradual increase in area. Approaching the top, the velocity is no longer sufficient to hold the granules of catalyst in suspension so that after this point the water is completely clarified.

In the passage through the catalyst, the precipitated compounds of calcium and magnesium adhere to the original granules, whose size is thereby increased. Because of the rotative motion within the cone, these heavier particles tend to gravitate to the bottom, where they may be removed through a special cock provided for the purpose. No sludge is produced.

For boiler-feed purposes it is not usual to install the Spiractor equipment alone unless the water to be treated contains only carbonate hardness. For waters possessing both

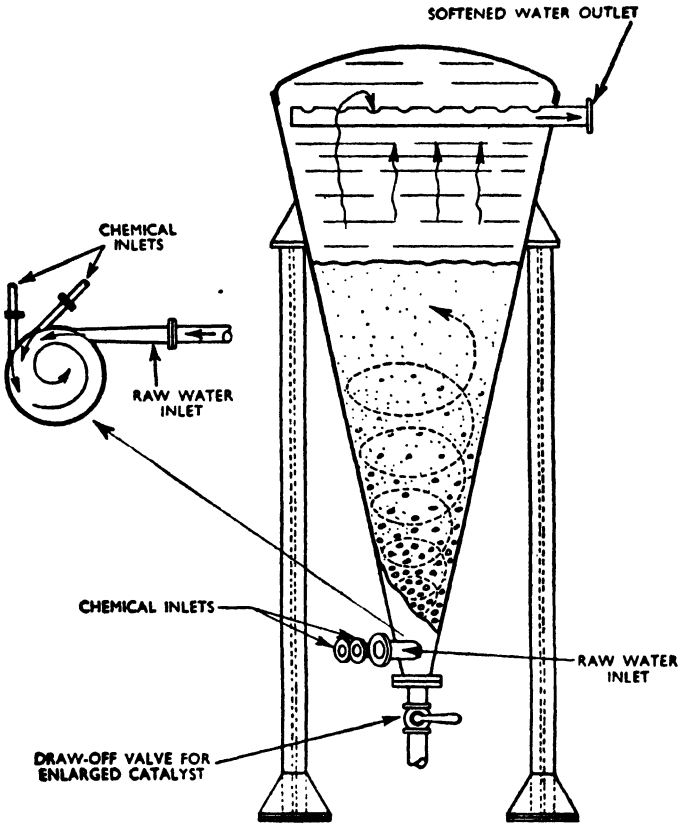


FIG. 133. DIAGRAM OF "SPIRACTOR" PLANT
(*Permutit Co. Ltd.*)

temporary and permanent hardness it is recommended that the plant should be used in conjunction with a base exchange softener, by which means a very suitable water of practically zero hardness can be produced.

Base Exchange

In the base exchange or zeolite process, the water is passed through a specially prepared zeolite bed, which has the property of exchanging its sodium for the calcium and magnesium constituents of the water, thus rendering the scale-forming constituents soluble.

FIG. 134. BASE EXCHANGE PLANT
(*Permutit Co. Ltd.*)

The calcium and magnesium are thus retained, as it were, by the zeolite, and the remaining water contains only soluble, non-scale-forming salts.

The process does not, however, continue indefinitely, and the zeolite gradually loses the property of exchange. It can, however, be restored by flushing through with a strong solution of brine, which removes and carries away with it the calcium and magnesium. Thus, in the working of a base-exchange plant, a definite cycle of operation is necessary, and it is absolutely essential that the strictest routine should be observed with regard to the regeneration and subsequent flushing. Usually, two reaction vessels are provided, so that the supply of softened water to the boiler may be continuous.

It will be observed that since the calcium and magnesium salts are converted into sodium salts, the temporary hardness will be converted into sodium bi-carbonate, which, on being heated, will decompose with release of CO_2 and the formation of sodium carbonate.

The possible dangers of corrosion, if not in the boiler itself, then in the condense lines of the system, must not be neglected. Also, the further decomposition of sodium carbonate into CO_2 and caustic soda, with the attendant possibility of caustic embrittlement, should be realized. The conversion of the permanent hardness into soluble salts, and the resulting tendency for the density of the boiler water to increase, make a definite and controlled blow-down routine a necessity.

Base-exchange plant should not as a rule be installed where the water to be treated has high temporary hardness, for the reasons already given. A complete plant is shown in Fig. 134. It is particularly necessary to study every aspect of the case, both from the standpoint of the chemical characteristics of the water and its freedom from purely physical impurities. The zeolite bed may be ruined by the presence of suspended matter and certain metallic compounds which may be contained in the water. Some form of filter or settling tank may be necessary in order to remove injurious matter.

Some form of lime-soda treatment may be installed if desired, to remove temporary hardness before the base-exchange plant. The Spiractor plant is particularly suitable for this purpose because of the rapidity with which it performs the necessary function of removing the carbonate hardness.

Demineralization

A comparatively recent development of the base-exchange principle is shown in Fig. 135. It produces a water which is comparable with distilled water. The principle, which can only be explained briefly, is as follows. Certain artificial base-exchange materials, when regenerated with acid instead of brine, will convert calcium, magnesium, and sodium salts into the appropriate acids. The passing of the water through a bed of this material forms the first stage of the process.

Certain other materials of the resin type have the property of retaining acids. By-passing the "converted" water through

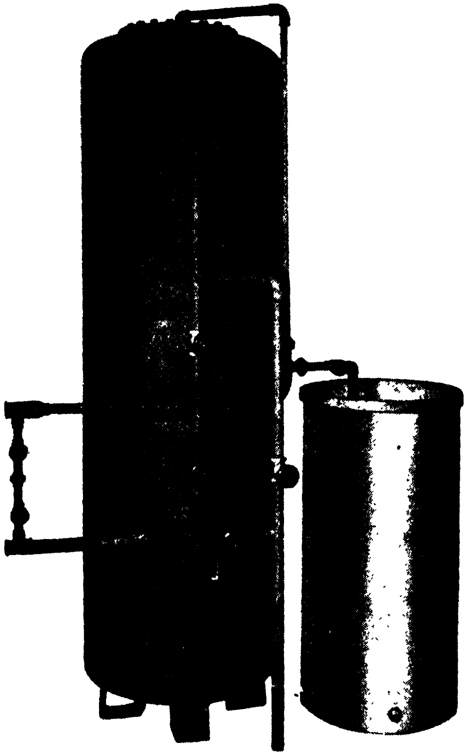


FIG. 135. PART OF DEMINERALIZATION PLANT
(*Permutit Co. Ltd.*)

a bed of the second material, the acids, other than carbonic acids formed by the first stage of the process are removed, and pure water is the result. The carbonic acid may then be removed by degassification.

If silica is present in the raw water it is not affected by the process, and passes into the boiler, some conditioning being possibly necessary to prevent the formation of silica scales. Practically all the dissolved salts are removed. The base-exchange materials are regenerated in the first stage by acids, and in the second, the acid-retaining stage, with suitable alkalis such as caustic soda, and sodium carbonate. A schematic diagram of the complete plant is shown in Fig. 136.

Where the size of the plant will not justify the costs of external treatment plant, or, for other reasons not the least of which may be the lack of suitable supervision or control, it may be found desirable to use some type of internal treatment. There are very many forms of varying efficacy to be obtained, and although all may be effective on a particular water in some

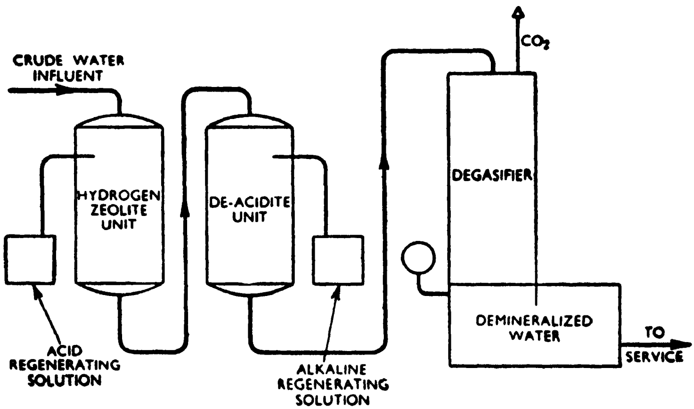


FIG. 136. SCHEMATIC DIAGRAM OF DEMINERALIZATION PLANT (*Permutit Co. Ltd.*)

degree, it is generally found that for any one case, some trial may be necessary before the best can be decided upon.

Broadly speaking, there are two main classes, to which all internal treatments belong: chemical or colloidal.

Chemical or Colloidal

The chemical forms may consist of sodium carbonate or phosphate, which is added to the water before its entry to the boiler, in amounts calculated to precipitate the scale-forming salts. The sludges so formed are thrown down inside the boiler, and unless precautions are taken it may bake on to the heated surfaces in considerable quantities. The sludges must therefore be moved as soon as formed, by means of the boiler blow-down. Tannins are sometimes added to the phosphates in order to keep the sludges in a suspended form until they can be dealt with in this way. The difficulty of effectively disposing of the sludges has hitherto been a major disadvantage to the use of internal treatments, and has put definite limits to the type of boiler for which they would normally be selected.

The Purifying Plant

There has, however, recently appeared a form of sludge-removal plant, which has very greatly enlarged the field of

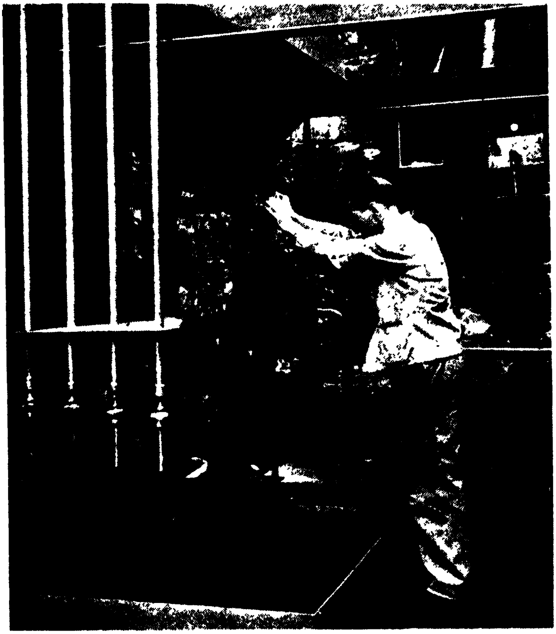


FIG. 137. SLUDGE DRAWING FROM HOUSEMAN-GROOME PLANT
(Houseman & Thompson, Newcastle upon Tyne)

usefulness of the internal treatment for which it was designed. This is the Houseman-Groome Purifying Plant, supplied by Messrs. Houseman & Thompson, of Newcastle upon Tyne. The treatment with which it is used is known as DM Boiler enamel, and its action is partly colloidal. A special blend of vegetable extracts is added to the water, whereby a condition is produced in the boiler in which the formation of adhesive scale cannot occur. Continued treatment causes the formation of a protective polish on the internal surfaces. The great difference between this and other treatments is that the quantities to be added are not critical, and provided that the concentration of the DM solution is kept above a given point, ascertainable by a simple test, the treatment is effective. No harm can be done to the boiler or the auxiliaries by overdosing.

The action of the solution keeps the solid particles in suspension while the boiler is at work, and where a purifying plant is not fitted, the resulting sludges can usually be hosed out on opening up. It is necessary to keep the suspended matter in the water to certain limits, depending on the type of boiler, conditions of working, and other factors, by using the blow-down. With this type of treatment it is particularly necessary to control the blow-down in order to prevent not only excessive loss of heat, but loss of DM.

FIG. 138. PICK-UP FITTING FOR HOUSEMAN-GROOME PLANT. BABCOCK WIF TYPE BOILER

(Houseman & Thompson, Newcastle upon Tyne)

The purifying plant constitutes a simple means of controlling this suspended matter, with little loss of heat and no loss of treatment.

Two settling tanks are installed, to which continuous blow-down pipes from the boiler are brought. The size of the tanks is calculated to give the required time for sludge settlement, and the "decanted" water, free from sludge, is returned to the hot well, containing practically all its unused DM, and some proportion of its sensible heat. The tanks are provided with sludge draw-off cocks, which are operated at periods determined by the working of the boiler, and the sludge content of the water. The operation is shown in Fig. 137.

The connections between the boiler and the settling tanks call for special comment.

A pick-up point is chosen, the position of which depends on the type of boiler. This may necessitate internal pipes in horizontal shell boilers, but in quick circulation units a perfectly satisfactory arrangement may be made, without piercing the boiler, by the external fitting shown in Fig. 138, which illustrates the design adopted for a WIF type Babcock boiler. A simple pad of sufficient thickness to take the $\frac{3}{4}$ in. bore pick-up valves is bored to suit the standard blow-down flanges, and inserted behind the normal blow-down valve. Thus, the functioning of the blow-down valve is not interfered with in any way, and may be used if desired as well as the pick-up valves. The latter must be of the blow-off type, capable of locking in the open or closed positions, and must be of the sturdiest possible design.

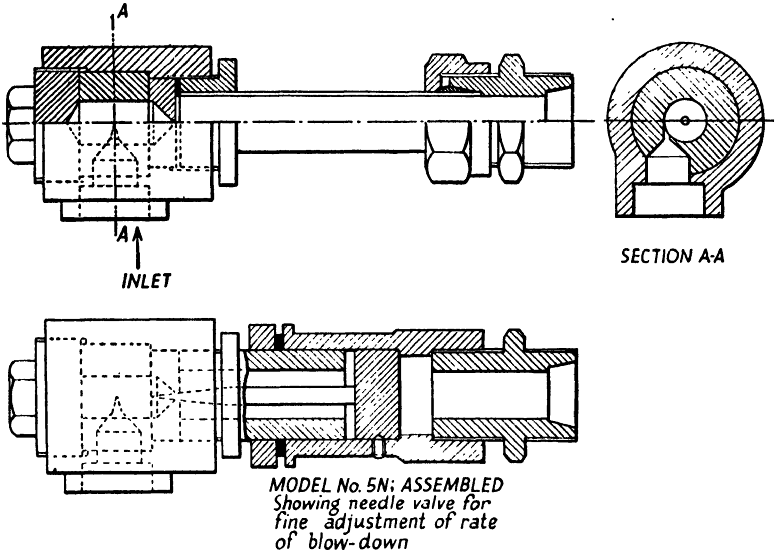


FIG. 139. HOUSEMAN-GROOME ORIFICE VALVE
(Houseman & Thompson, Newcastle upon Tyne)

For vertical boilers a satisfactory arrangement for the pick-up points is in the space between the firebox and outer shell, just above the ogee ring.

Whatever the type of boiler, the principle to be observed is that the pick-up point shall be situated at the lowest point in the natural circulation, and below the main rising area. The position already discussed is therefore convenient, and the attachment easily fitted. From the pick-up valves, $\frac{3}{4}$ in. bore pipe is taken vertically upwards, to some point from which a gravity flow may be obtained to the settling tanks. Here, specially designed orifice valves are fitted. The principle on which these orifice valves operate is made clear in Fig. 139. The flow through them is tangential, and the rapid swirling motion so produced keeps the orifice clear at all times. The size of the outlet orifice determines the quantity of continuous blow-down, and is carefully selected to suit the particular conditions.

From the orifice valves, copper pipes are taken to the settling tanks, to which the sludge-carrying water flows by gravity, aided by the slight pressure at the orifice outlet. It is important that the interior of these pipes should offer as little obstruction to the flow as possible, and to this end they are made continuous if feasible. In the case of long runs, where jointing cannot be avoided, the joints must be carefully butted and bronze-welded, or alternatively jointed with "Ermeto" or similar couplings, which give a perfectly smooth interior. No ridging can be tolerated.

Calculation of Blow-down

The amount of continuous blow-down required must depend to some extent on the characteristics of the water and conditions of working. The feature to be emphasized is that with the purifying plant, which returns a considerable proportion of the heat in the blow-down to the hot well, far higher rates of blow-down can be used than would be economically possible if the heat were all blown away to waste. Under certain conditions, where hot condensate is returned to the feed tank from other sources, the limiting factor may well be the highest temperature the pumps can handle, which again must depend on the position of the feed tank relative to the pumps.

It is important to remember, when assessing the economies possible with this type of plant, that although the suspended matter is effectively disposed of, the dissolved solids are returned to the feed tank, and therefore pass again to the boiler. These dissolved solids, not dealt with by the settling tanks, therefore tend to concentrate in the boiler, and must be controlled by additional blow-down to drain. It should however be appreciated that such dissolved solids will not be formed to anything like the extent that is usual with a chemical, as distinct from an organic treatment. This additional blow-down may be accomplished either through the medium of the normal blow-down cock or by continuous blow-down.

Where the purifying plant is installed, a small leak-off connection may be fitted to the outlet pipe, but to this there is an objection, as will be seen later.

Although certain arguments could be advanced in favour of periodical blow-down (and this applies to all forms of treatment where blow-down is necessary) continuous blow-down from a connection at the boiler is preferred for two reasons.

First, since the amount of blow-down necessary can be calculated, it is possible to fit either an orifice or a form of control valve, which can be locked in the position to give the desired results. Both these devices are likely to give far more certain results than a valve which can be manipulated, and is thus susceptible to forgetfulness on the one hand, and over-enthusiasm on the other.

But the second aspect is the more important because it is involved with the necessity of heat recovery, particularly with the utilization of flash steam. It is perfectly simple to arrange effective heat conservation apparatus, to deal with a small but continuous efflux, but little could be done with the same total quantity discharged over a period of perhaps only a few seconds.

The heat recovery may be twofold.

Assume a boiler working at 150 lb per sq in. gauge. The temperature of the water at this pressure is 366° F, and contains 334 B.Th.U. of sensible heat per lb. Since it must be run to waste because of the dissolved salts it contains, heat can only be recovered from it through some simple form of heat exchanger. But at atmospheric pressure, the water cannot have a temperature higher than 212° F, and so can only hold 180 B.Th.U. of sensible heat. Only part of this can be recovered in the heat exchanger, but even if for the moment it may be assumed that all can be usefully employed, there are still 154 B.Th.U. flashed off as steam. In other words, although a very great saving has been made as compared with complete neglect of heat conservation, there are still nearly half the recoverable B.Th.U. thrown away.

If instead of the simple heat exchanger used alone, the blow-down can first of all be passed into a flash pot at say 2 lb per sq in., and the steam so generated be passed to some other point of use, the sensible heat of the condensate can still be recovered in the heat exchanger.

Blow-down at 150 lb per sq in. . .	334 B.Th.U./lb
Sensible heat in condensate at 2 lb .	187 ,,

(This is available for part recovery in the heat exchanger.)

The remainder passed to feed tank as flash steam is 147 B.Th.U. per lb.

It is from the standpoint of flash steam recovery that the fitting of the leak-off connection to the settling tanks is objectionable, for the reason that it would mean the construction of the tanks as pressure vessels. The recovery of the flash steam from the sludge-carrying water passing through the plant is, on the other hand a very simple matter, involving only the leading of a flash pipe from the highest point of the plant to the feed tank, which usually is directly underneath, or at all events very close. This can be arranged so that what little pressure there is in the tanks is so small as to be negligible.

It should be clear that whatever the form of treatment, and even where no treatment at all is used, blow-down, in greater or less amounts, is always necessary. The point which is not always realized is that it should always be controlled, either by one or other of the devices mentioned, or by strict supervision. The haphazard or indiscriminate manipulation of a blow-down cock should not be allowed under any circumstances.

Routine Testing

The testing of the boiler water to ascertain its condition, both with regard to suspended matter and dissolved solids, should therefore be a matter of routine. Regular samples should be drawn at the same time each day. The taking of samples is in itself a matter to which far too little attention has been given in the past, but a little thought will show that the usual method of drawing the sample from the boiler under pressure into an open pot, besides being troublesome and messy, is also extremely inaccurate. Some considerable portion of the sample so drawn must flash into steam before ever the pot is reached, with the result that the remainder can bear little relation to the actual contents of the boiler shell.

To obviate this difficulty, various devices are employed, some fitted with cooling coils to condense the flash as formed. A very simple and easily fitted apparatus is shown in Fig. 140. This device, patented by Messrs. Houseman & Thompson, combines an effective sampler with means of injecting DM boiler solution into the boiler.

Manipulation of the sleeve valve either connects the boiler to the body of the apparatus, so that a sample of the water is forced into the body under pressure, or alternatively provides a straight through connection to the boiler through the external check valve shown. By a very simple pressure vessel, not shown, the power of the boiler feed pump is utilized to force the treatment solution into the boiler.

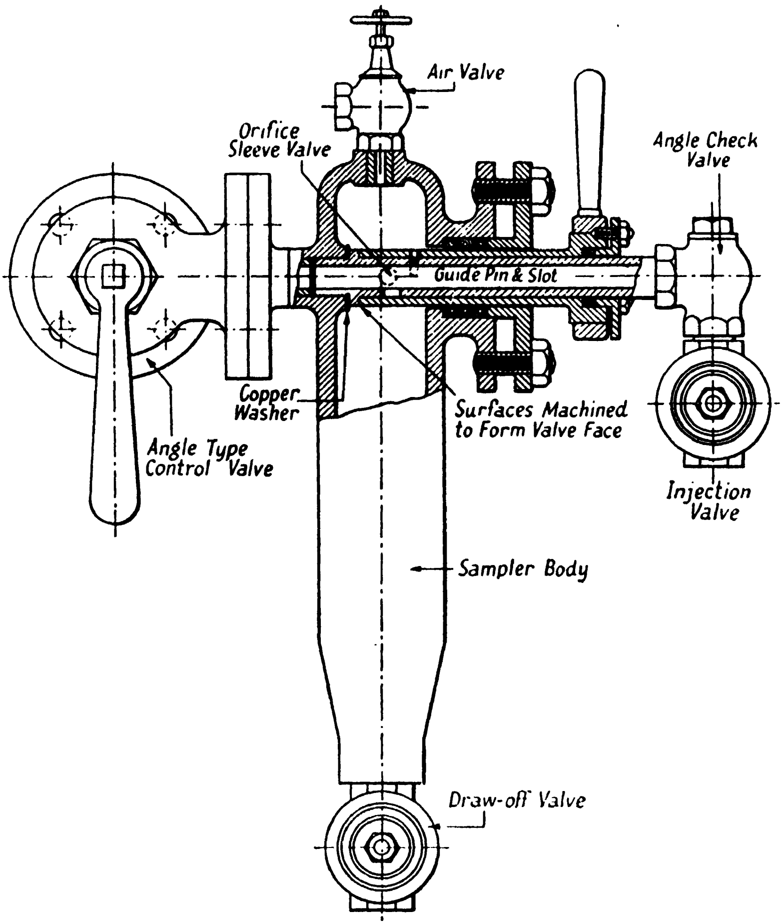


FIG. 140. HOUSEMAN SAMPLER
(Houseman & Thompson, Newcastle upon Tyne)

When the sample has been correctly taken, it should be allowed to cool and filter. The suspended matter is ascertained by filtering a known volume, say 1000 ml of the water, and calcining and weighing the residue. Total dissolved solids may be found either by hydrometer, or by evaporating filtered sample to dryness, and weighing. Where the hydrometer is used, it should be of a special pattern, sensitive and designed for the purpose.

Normally, for every 10 grains of sodium salts dissolved in one gallon of water, the density is increased by roughly 0·0001. For practical purposes, therefore, it is sufficient to ignore the unit figure, and multiply the first four figures after the point by 10. Thus, a density reading of 1·0100 will be 1000 grains per gallon.

Temperature corrections are necessary, and a table of these is given in Table XII.

TABLE XII

TEMPERATURE CORRECTIONS FOR DENSITY READINGS

Each division on the hydrometer = 100 grains per gallon = 0·23 oz. Therefore after translating the read divisions into grains per gallon, allow additional grains according to temperature as follows—

Temperature of Sample, Fahrenheit	Correction to be added to the Hydrometer reading to obtain Grains per Gallon at 60° F	Temperature of Sample, Fahrenheit	Correction to be added to the Hydrometer reading to obtain Grains per Gallon at 60° F
60°	Nil	81°	257
61°	10	82°	272
62°	19	83°	288
63°	29	84°	304
64°	39	85°	320
65°	49	86°	336
66°	60	87°	354
67°	71	88°	372
68°	82	89°	390
69°	94	90°	408
70°	106	91°	426
71°	120	92°	444
72°	134	93°	462
73°	146	94°	480
74°	158	95°	499
75°	170	96°	519
76°	184	97°	539
77°	198	98°	559
78°	212	99°	579
79°	227	100°	599
80°	242		

Thus, assuming the reading at 60° F were *nil*, there would be a minus reading of 96 grains per gallon (approximately 1 division) at 70° F and it would therefore be necessary to add 106 grains per gallon to correct the reading.*

* If the specific gravity hydrometer is marked 990–1040 the minus readings will show on the stem of the hydrometer.

The hydrometer is *not* a reliable or accurate method for determining TDS, and its findings should be regularly checked by evaporating and weighing. The maximum permissible TDS in a boiler water is a figure which can only be ascertained by experience with the particular plant. So much depends on the type of boiler, its working pressure, and upon the conditions under which it works. A boiler subjected to steady loads can safely work at much higher figures for TDS than one which is called upon to sustain heavy and sudden peaks.

Firing methods, too, affect the tendency of the boiler to prime, one of the most fruitful causes being the habit of carrying too high a level in the gauge glasses. Boilers with an inherently rapid circulation will usually prime more easily than the more sluggish shell boiler, so that every case must be studied on its merits.

The safe figure for the plant must, however, be known before the economical amount for the blow-down can be calculated. The Ministry of Fuel and Power give some purely arbitrary figures, which are quoted here.

Type of Boiler	*TDS Grains/Gallon*
Lancashire and Cornish . . .	1000 to 1200
Economic	250 to 300
Vertical	150 to 300
Water tube (low pressure). . .	300 to 700
Locomotive	150 to 250

With knowledge of the maximum permissible TDS, and also of the dissolved solids in the make-up water (obtained by evaporating to dryness), the correct amount of blow-down can be calculated.

Let The make-up water per hour be Mu gallons
Soluble salts in the make-up Sm grains per gallon
Desired concentration C grains per gallon

Then $Mu \times Sm =$ grains of salts added per hour.

But if one gallon of water at C grains per gallon is replaced by one gallon of feed water at Sm grains per gallon, it follows that $C - Sm$ grains are evacuated from the boiler by the removal of one gallon through the blow-down. Then the amount of blow-down per hour must be $\dfrac{Mu \times Sm}{C - Sm}$

As an example, take the case of a Babcock W1F type boiler, evaporating 5000 lb of steam per hour. Most of the feed is returned condensate, and the make-up of raw water is 10 per cent.

The desired maximum concentration in the boiler is 300 grains per gallon, and it has been ascertained that the raw make-up water contains 20 grains per gallon of soluble salts.

Then
$$Mu = 50 \text{ gallons per hour}$$
$$Sm = 20 \text{ grains per gallon}$$
$$C = 300 \text{ grains per gallon}$$
$$\frac{50 \times 20}{300 - 20} = 3{\cdot}56 \text{ gallons per hour.}$$

Finally, it may be said that whatever the type of plant, and whatever the form of treatment used, success can only be achieved by strict attention and routine testing.

Feed water is one of the essential points at which the engineer and the chemist combine, and here it is again repeated that correct treatment, and all the problems entailed are the business of the specialist, whose advice should be sought without hesitation. Moreover, it is important to add that, in the case of new plant, advice should be sought at the outset, in the earliest possible stages, as soon as the size and type of plant can be predicted.

In operation, it is little use calling in the feed water expert when water is already pouring out of the back tube plate. Every maker supplies equipment for routine tests, which can be carried out with a minimum of trouble, if not by the engineer then by the works laboratory staff. Regularity in this respect will give due warning of possible trouble in time to prevent such dismal happenings, apart from the fact that visual and certain evidence of boiler condition do much to ease at least one harassment from the shoulders of the works engineer.

APPENDIX I

PROPERTIES OF SATURATED STEAM

Compiled from Marks and Davis' *Tables and Diagrams of the Thermal Properties of Saturated and Superheated Steam*, by permission of the authors and Messrs. Longmans, Green & Co.

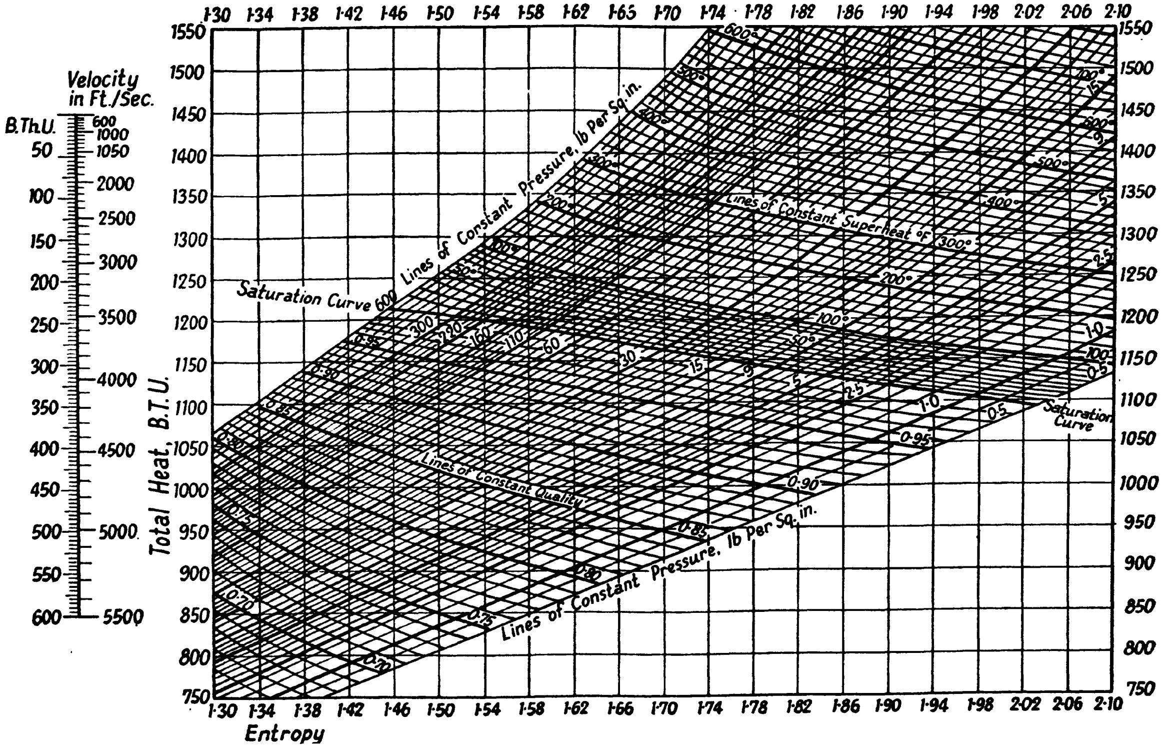
Pressure, lb/in.² abs. p	Temperature, degrees F t	Sp. vol. ft³/lb v	Heat in B.Th.U.			Entropy		
			Sensible h	Latent L	Total H	Water ϕw	Evaporation ϕe	Total ϕ
0·5	79·5	643·0	49·7	1046·8	1094·5	0·0926	1·9413	2·0339
0·75	92·4	436·5	60·4	1039·8	1100·2	0·1157	1·8839	1·9996
1·0	101·83	333·0	69·8	1034·6	1104·4	0·1327	1·8427	1·9754
1·25	109·42	269·7	77·36	1030·35	1107·81	0·1461	1·8100	1·9561
1·5	115·79	227·0	83·7	1026·8	1110·5	0·1573	1·7845	1·9418
1·75	121·3	196·4	89·2	1023·7	1112·9	0·1670	1·7623	1·9293
2·0	126·15	173·5	94·0	1021·0	1115·0	0·1749	1·7431	1·9180
2·25	130·54	155·0	98·39	1018·5	1116·89	0·1824	1·7260	1·9084
2·5	134·5	140·4	102·36	1016·24	1118·6	0·1891	1·7106	1·8997
3·0	141·52	118·5	109·4	1012·3	1121·6	0·2008	1·6840	1·8848
3·5	147·62	102·4	115·48	1008·8	1124·28	0·2110	1·6612	1·8722
4·0	153·01	90·5	120·9	1005·7	1126·5	0·2198	1·6416	1·8614
4·5	157·84	81·0	125·7	1002·88	1128·58	0·2277	1·6241	1·8518
5	162·28	73·33	130·1	1000·3	1130·5	0·2348	1·6084	1·8432
6	170·06	61·89	137·9	995·8	1133·7	0·2471	1·5814	1·8285
7	176·85	53·56	144·7	991·8	1136·5	0·2579	1·5582	1·8161
8	182·86	47·27	150·8	988·2	1139·0	0·2673	1·5380	1·8053.
9	188·27	42·36	156·2	985·0	1141·1	0·2756	1·5202	1·7958
10	193·22	38·38	161·1	982·0	1143·1	0·2832	1·5042	1·7874
12	201·96	32·36	169·9	976·6	1146·5	0·2967	1·4760	1·7727
14	209·55	28·02	177·5	971·9	1149·4	0·3081	1·4523	1·7604
16	216·3	24·79	184·4	967·6	1152·0	0·3183	1·4311	1·7494
18	222·4	22·16	190·5	963·7	1154·2	0·3273	1·4127	1·7400
20	228·0	20·08	196·1	960·0	1156·2	0·3355	1·3965	1·7320
22	233·1	18·37	201·3	956·7	1158·0	0·3430	1·3811	1·7241
24	237·8	16·93	206·1	953·5	1159·6	0·3499	1·3670	1·7169
26	242·2	15·72	210·6	950·6	1161·2	0·3564	1·3542	1·7106
28	246·4	14·67	214·8	947·8	1162·6	0·3623	1·3425	1·7048
30	250·3	13·74	218·8	945·1	1163·9	0·3680	1·3311	1·6991
32	254·1	12·93	222·6	942·5	1165·1	0·3733	1·3205	1·6938
34	257·6	12·22	226·2	940·1	1166·3	0·3784	1·3107	1·6891
36	261·0	11·58	229·6	937·7	1167·3	0·3832	1·3014	1·6846
38	264·2	11·01	232·9	935·5	1168·4	0·3877	1·2925	1·6802
40	267·3	10·49	236·1	933·3	1169·4	0·3920	1·2841	1·6761
42	270·2	10·02	239·1	931·2	1170·3	0·3962	1·2759	1·6721
44	273·1	9·59	242·0	929·2	1171·2	0·4002	1·2681	1·6683
46	275·8	9·20	244·8	927·2	1172·0	0·4040	1·2607	1·6647
48	278·5	8·84	247·5	925·3	1172·8	0·4077	1·2536	1·6613
50	281·0	8·51	250·1	923·5	1173·6	0·4113	1·2468	1·6581
52	283·5	8·20	252·6	921·7	1174·3	0·4147	1·2402	1·6549
54	285·9	7·91	255·1	919·9	1175·0	0·4180	1·2339	1·6519
56	288·2	7·65	257·5	918·2	1175·7	0·4212	1·2278	1·6490
58	290·5	7·40	259·8	916·5	1176·4	0·4242	1·2218	1·6460
60	292·7	7·17	262·1	914·9	1177·0	0·4272	1·2160	1·6432

PROPERTIES OF SATURATED STEAM—*continued*

Pressure, lb/in.² abs. p	Temperature, degrees F t	Sp. vol. ft³/lb v	Heat in B.Th.U. Sensible h	Latent L	Total H	Entropy Water ϕw	Evaporation ϕe	Total ϕ
62	294·9	6·95	264·3	913·3	1177·6	0·4302	1·2104	1·6406
64	297·0	6·75	266·4	911·8	1178·2	0·4330	1·2050	1·6380
66	299·0	6·56	268·5	910·2	1178·8	0·4358	1·1998	1·6355
68	301·0	6·38	270·6	908·7	1179·3	0·4385	1·1946	1·6331
70	302·9	6·20	272·6	907·2	1179·8	0·4411	1·1896	1·6307
72	304·8	6·04	274·5	905·8	1180·4	0·4437	1·1848	1·6285
74	306·7	5·89	276·5	904·4	1180·9	0·4462	1·1801	1·6263
76	308·5	5·74	278·3	903·0	1181·4	0·4487	1·1755	1·6242
78	310·3	5·60	280·2	901·7	1181·8	0·4511	1·1710	1·6221
80	312·0	5·47	282·0	900·3	1182·3	0·4535	1·1665	1·6200
82	313·8	5·34	283·8	899·0	1182·8	0·4557	1·1623	1·6180
84	315·4	5·22	285·5	897·7	1183·2	0·4579	1·1581	1·6160
86	317·1	5·10	287·2	896·4	1183·6	0·4601	1·1540	1·6141
88	318·7	5·00	288·9	895·2	1184·0	0·4623	1·1500	1·6123
90	320·3	4·89	290·5	893·9	1184·4	0·4644	1·1461	1·6105
92	321·8	4·79	292·1	892·7	1184·8	0·4664	1·1423	1·6087
94	323·4	4·69	293·7	891·5	1185·2	0·4684	1·1385	1·6069
96	324·9	4·60	295·3	890·3	1185·6	0·4704	1·1348	1·6052
98	326·4	4·51	296·8	889·2	1186·0	0·4724	1·1312	1·6036
100	327·8	4·429	298·3	888·0	1186·3	0·4743	1·1277	1·6020
105	331·4	4·23	302·0	885·2	1187·2	0·4789	1·1191	1·5980
110	334·8	4·047	305·5	882·5	1188·0	0·4834	1·1108	1·5942
115	338·1	3·880	309·0	879·8	1188·8	0·4877	1·1030	1·5907
120	341·3	3·726	312·3	877·2	1189·6	0·4919	1·0954	1·5873
125	344·4	3·583	315·5	874·7	1190·3	0·4959	1·0880	1·5839
130	347·4	3·452	318·6	872·3	1191·0	0·4998	1·0809	1·5807
135	350·3	3·331	321·7	869·9	1191·6	0·5035	1·0742	1·5777
140	353·1	3·219	324·6	867·6	1192·2	0·5072	1·0675	1·5747
145	355·8	3·112	327·4	865·4	1192·8	0·5107	1·0612	1·5719
150	358·5	3·012	330·2	863·2	1193·4	0·5142	1·0550	1·5692
155	361·0	2·92	332·9	861·0	1194·0	0·5175	1·0489	1·5664
160	363·6	2·834	335·6	858·8	1194·5	0·5208	1·0431	1·5639
165	366·0	2·753	338·2	856·8	1195·0	0·5239	1·0376	1·5615
170	368·5	2·675	340·7	854·7	1195·4	0·5269	1·0321	1·5590
175	370·8	2·602	343·2	852·7	1195·9	0·5299	1·0268	1·5567
180	373·1	2·533	345·6	850·8	1196·4	0·5328	1·0215	1·5543
185	375·4	2·468	348·0	848·8	1196·8	0·5356	1·0164	1·5520
190	377·6	2·406	350·4	846·9	1197·3	0·5384	1·0114	1·5498
195	379·8	2·346	352·7	845·0	1197·7	0·5410	1·0066	1·5476
200	381·9	2·29	354·9	843·2	1198·1	0·5437	1·0019	1·5456
210	386·0	2·187	359·2	839·6	1198·8	0·5488	0·9928	1·5416
220	389·9	2·091	363·4	836·2	1199·6	0·5538	0·9841	1·5379
230	393·8	2·004	367·5	832·8	1200·2	0·5586	0·9758	1·5344
240	397·4	1·924	371·4	829·5	1200·9	0·5633	0·9676	1·5309
250	401·1	1·85	375·2	826·3	1201·5	0·5676	0·9600	1·5276
260	404·5	1·782	378·9	823·1	1202·1	0·5719	0·9525	1·5244
270	407·9	1·718	382·5	820·1	1202·6	0·5760	0·9454	1·5214
280	411·2	1·658	386·0	817·1	1203·1	0·5800	0·9385	1·5185
290	414·4	1·602	389·4	814·2	1203·6	0·5840	0·9316	1·5156
300	417·5	1·551	392·7	811·3	1204·1	0·5878	0·9251	1·5129

APPENDIX II

TOTAL HEAT/ENTROPY DIAGRAM FOR SATURATED AND SUPERHEATED STEAM (ABSOLUTE PRESSURE)



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